PRACTICAL EXAMPLES IN AIR CONDITIONING & MECHANICAL VENTILATION SYSTEM DESIGN

(This short course qualifies for 12 PDUs)

by

MR PETER LIM

Attendee's Name

GOH WHYE LIANG



APPLIED TECHNOLOGY
GROUP OF COMPANIES
Australia • New Zealand • Hong Kong • Malaysia • Singapore

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SINGAPORE

7th & 8th April 2008
JW MARRIOT HOTEL,
KUALA LUMPUR

Practical Examples in Mechanical Ventilation and Air-conditioning system design

Programme

Day One

9:00 am	Introduction, outdoor air required for comfort air-conditioning, Outdoor air required for mechanical ventilation, Design of smoking lounge for hotel, shopping complex/airport
10:30 am	Morning Tea/Coffee
10:50 am	Mechanical ventilation of carpark
12:30 pm	Lunch
2:00 pm	Mechanical ventilation of Plant room, Staircase pressurization
3:30 pm	Afternoon Tea/Coffee
3:50 pm to 5:00 pm	Lobby pressurization , lift motor room ventilation

Day Two

9:00 am	Examples in U –value and cooling load estimation, zoom smoke control
10:30 am	Morning Tea/Coffee
10:50 am	Atrium smoke control , Design of smoke ventilation system for Single Storey Industrial Building
12:30 pm	Lunch
2:00 pm	Examples in smoke control system design , design of ventilation system for flammable and explosive substances
3:30 pm	Afternoon Tea/Coffee
3:50 pm to 5:00 pm	Dilution ventilation design for IAQ

ASHRAE
CIBSE
AS 1668-1,2

Ventilation System

By

Lim Eng Seng

MCIBSE, C.ENG

MASHARE

B.ENG (Hons), Msc(Building Science)

Ventilation System

- Introduction- ACH, Fresh air req
- Basic Industrial Ventilation
- Industry Exhaust Stacks
- Natural Ventilation Methods
- Mechanical Ventilation Methods
- Kitchen Exhaust Hood
- Carpark Ventilation
- Ventilation of Internal Bathrooms and WC

Need for Ventilaiton

1. Oxygen for breathing
2. Dilution of $C0_2$ respiration
3. Control body odours, water vapour given off by people
4. Reduce air temperature
5. Maintain air movement, 0.25 m/s

Five Method to calculate Air Volume Required

1. Air change per hour (ACH)
2. Cubic meter per minute or (l/s) per m^2 of floor area
3. Number of occupants in the space
4. Minimum air velocity through the space
5. Removal of heat in the space

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Air Change Method

$N = Q \times 3600/Volume$

N = number of air changes per hour

Q = volume flow rate in m^3/s

Volume = space volume in m^3.

Example 1

What fan flow rate is required for an office, 8 x 16 x 3 m height, that needs an air change every 5 minute.

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Solution for Example 1

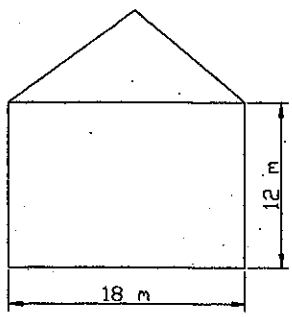
$V = 8 \times 16 \times 3 = 384\ m^3$

Air change every 5 minute = 60/5
= 12 ACH
Flow rate for 12 ACH
$= 12 \times 384/3600\ m^3/s = 1.28\ m^3/s$

Example 2

A warehouse building with dimension as shown is to be ventilated. The building is 14 m long.

Calculate the required fan flow rate in m^3 /hr.



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Solution for Example 2

From table 2 , ACH = 6

Flow rate = 6 ACH

= 6 x 14 x 18 x 12/ 3600

= 5.04 m^3/s
= 18144 m^3/hr

Example 3

A small factory accommodates 20 workers and has a floor area of length 20 m and width 15 m. The ceiling height is 2.5 m.

Calculate the outdoor air required for mechanical ventilation.

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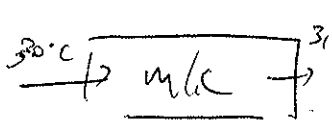

$Q_s = 1.23 \times \frac{m^3}{s} \times \Delta t$ (kW)

$\frac{m^3}{s} = \frac{Q_s}{1.23 \times \Delta t \text{ (1 or 2°C)}}$

= 20 ACH (Don't follow code @ 6 ACH)

Solution for Example 3

The floor area = 20 m x 15 m = 300 m^2

The room volume = 300 x 2.5 = 750 m^3

From table, 6 ACH is required

Air flow required = 750 x 6 = 4500 m^3/hr

Example 4

Calculate the fresh air requirements for air-conditioning.

In an air-conditioned restaurant, the floor area is 20 m wide by 35 m long and the ceiling height is 2.5 m , it can accommodate 450 occupants.

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Solution for Example 4

The floor area = 20 m x 35 m = 700 m^2

From table CP13

The required ventilation rate is 10 m^2 /hr/m^2 of floor area

Outdoor air required = 700 x 10

= 7000 m^3/hr

Example 5

An auditorium has a seating capacity of 500 persons. On most occasions all seats are taken, plus an additional 10% of people are standing. How much ventilation is required ?

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Solution to Example 5

Required cfm = 550 x 15 = 8250 cfm

or

8 l/s x 550 = 4400 l/s

(Note : 15 cfm per peron for ASHRAE 62-1999)

Table 1 Outdoor Air Supply Requirement for Comfort Air-conditioning

Type of building/Occupancy	CP13-1999 Min OA m^3/h per m^2
Restaurants	10.0
Dance Halls	19.0
Offices	2.3
Shops, supermarkets and departmental stores	3.6
Lobbies, concourses and corridors	0.9
Classrooms	9.2
Theatres and cinemas	7.3
Factories and workshops : Light Medium (Heavy) High	4.3 2.6 1.3
Hotel guest room	54 m^3/h per room

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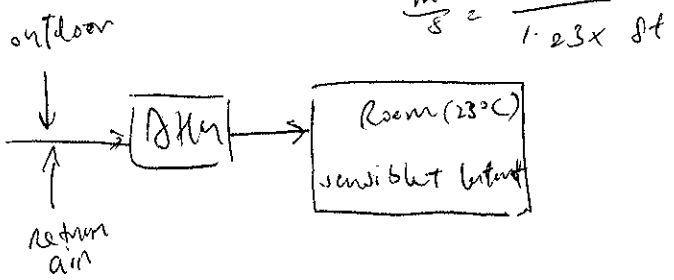

Table 2 Outdoor Air Supply for Mechanical Ventilation in Non-Air-conditioned Buildings

Type of Building	CP13 – 1999 (ACH)	CP13-1980 (ACH)	% Increased
Offices	6	6	0
Restaurants, Canteens	10	6	67
Shops, supermarkets and department stores	6	6	0
Workshops, factories	6	6	0
Classrooms	8	8	0
Theatres and cinemas	8	8	0
Lobbies,concourse, corridors, staircase and exits	4	4	0
Toilets, bathrooms	10	10	0
Kitchen (commercial, institutional and industrial)	20	20	0
Carparks	6	6	0
Fire Command Centers	6	-	-
Civil Defence Shelters	16	-	-
Wastewater facilities/pumping station	8	-	-

Note :

- For carparks and civil defense shelters, air change rate shall be based on 2.5 m ceiling height, if the ceiling exceeds 2.5 m. For such situation it is good engineering practice to locate the high level air inlets and outlets at 2.5 m height as shown in sketch.
- For heavily used public toilet, refer to Code of Practice on Environmental Health.
- With reference to Code of practice on Environmental health, minimum 15 ACH is required for toilets where general public is expected to walk in ,

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Outdoor Air Supply for Comfort Air-conditioning

Type of Building Occupancy	Minimum ODA Supply (L/s per m^2 Floor Area)	m3 /h per m^2 of Floor Area
Restaurants	2.80	10.0
Dance Halls	5.30	19.0
Offices	0.65	2.3
Shops, supermarkets & department stores	1.00	3.6
Lobbies, concourses & corridors	0.25	0.9
Classrooms	2.60	9.2
Theatres & cinemas	2.00	7.3
Factories & workshops :		
Light	1.20	4.3
Medium	0.70	2.6
High	0.35	1.3
Hotel guest rooms	15 L/s per room	54 m^3/ h per room

Outdoor Air Supply for Mechanical Ventilation in Non Air-conditioned Buildings

Type of Building Occupancy	Minimum ODA Supply Air-change per hour (ACH)
Offices	6
Restaurants, canteens	10
Shops, supermarkets and department stores	6
Workshops, factories	6
Classrooms	8
Theatres and cinemas	8
Lobbies, concourse, corridors, staircases and exits	4
Toilets**, bathrooms (Heavily used public toilets, use 15 ACH)	10
Kitchens (commercial, institutional and industrial)	20
Car Parks	6
Fire Command Centres	6
Civil Defence Shelters	16
Wastewater facilities/pumping installation	8

Table 3 **Outdoor Air Requirement for Ventilation**
ASHRAE 62-1999

Applications	Estimated Maximum Occupancy P/1000 ft^2 Or 100 m^2	Outdoor air requirements	
		l/s per person	l/s per m^2
Offices space	7	10	-
Conference room	50	10	-
Corridors and Utilities	-	-	0.25
Smoking lounge	70	30	-

TABLE 2
OUTDOOR AIR REQUIREMENTS FOR VENTILATION* (*Continued*)
2.1 COMMERCIAL FACILITIES (offices, stores, shops, hotels, sports facilities)

Application	Estimated Maximum** Occupancy P/1000 ft^2 or 100 m^2	Outdoor Air Requirements				Comments
		cfm/ person	L/s· person	cfm/ft^2	$L/s \cdot m^2$	
Retail Stores, Sales Floors, and Show Room Floors						
Basement and street	30			0.30	1.50	
Upper floors	20			0.20	1.00	
Storage rooms	15			0.15	0.75	
Dressing rooms				0.20	1.00	
Malls and arcades	20			0.20	1.00	
Shipping and receiving	10			0.15	0.75	
Warehouses	5			0.05	0.25	
Smoking lounge	70	60	30			Normally supplied by transfer air, local mechanical exhaust; exhaust with no recirculation recommended.
Specialty Shops						
Barber	25	15	8			
Beauty	25	25	13			
Reducing salons	20	15	8			
Florists	8	15	8			Ventilation to optimize plant growth may dictate requirements.
Clothiers, furniture				0.30	1.50	
Hardware, drugs, fabric	8	15	8			
Supermarkets	8	15	8			
Pet shops				1.00	5.00	
Sports and Amusement						
Spectator areas	150	15	8			When internal combustion engines are operated for maintenance of playing surfaces, increased ventilation rates may be required.
Game rooms	70	25	13			
Ice arenas (playing areas)				0.50	2.50	
Swimming pools (pool and deck area)				0.50	2.50	Higher values may be required for humidity control.
Playing floors (gymnasium)	30	20	10			
Ballrooms and discos	100	25	13			
Bowling alleys (seating areas)	70	25	13			
Theaters						Special ventilation will be needed to eliminate special stage effects (e.g., dry ice vapors, mists, etc.)
Ticket booths	60	20	10			
Lobbies	150	20	10			
Auditorium	150	15	8			
Stages, studios	70	15	8			
Transportation						Ventilation within vehicles may require special considerations.
Waiting rooms	100	15	8			
Platforms	100	15	8			
Vehicles	150	15	8			
Workrooms						
Meat processing	10	15	8			Spaces maintained at low temperatures (−10°F to + 50°F, or −23°C to + 10°C) are not covered by these requirements unless the occupancy is continuous. Ventilation from adjoining spaces is permissible. When the occupancy is intermittent, infiltration will normally exceed the ventilation requirement. (See Reference 18).

* Table 2 prescribes supply rates of acceptable outdoor air required for acceptable indoor air quality. These values have been chosen to dilute human bioeffluents and other contaminants with an adequate margin of safety and to account for health variations among people and varied activity levels.
** Net occupiable space.

TABLE 2
OUTDOOR AIR REQUIREMENTS FOR VENTILATION* (*Continued*)
2.1 COMMERCIAL FACILITIES (offices, stores, shops, hotels, sports facilities)

Application	Estimated Maximum** Occupancy P/1000 ft^2 or 100 m^2	Outdoor Air Requirements				Comments
		cfm/ person	L/s· person	cfm/ft^2	$L/s \cdot m^2$	
Photo studios	10	15	8			
Darkrooms	10			0.50	2.50	
Pharmacy	20	15	8			
Bank vaults	5	15	8			
Duplicating, printing				0.50	2.50	Installed equipment must incorporate positive exhaust and control (as required) of undesirable contaminants (toxic or otherwise).
2.2 INSTITUTIONAL FACILITIES						
Education						
Classroom	50	15	8			
Laboratories	30	20	10			Special contaminant control systems may be required for processes or functions including laboratory animal occupancy.
Training shop	30	20	10			
Music rooms	50	15	8			
Libraries	20	15	8			
Locker rooms				0.50	2.50	
Corridors				0.10	0.50	
Auditoriums	150	15	8			
Smoking lounges	70	60	30			Normally supplied by transfer air. Local mechanical exhaust with no recirculation recommended.
Hospitals, Nursing and Convalescent Homes						
Patient rooms	10	25	13			Special requirements or codes and pressure relationships may determine minimum ventilation rates and filter efficiency. Procedures generating contaminants may require higher rates.
Medical procedure	20	15	8			
Operating rooms	20	30	15			
Recovery and ICU	20	15	8			
Autopsy rooms				0.50	2.50	Air shall not be recirculated into other spaces.
Physical therapy	20	15	8			
Correctional Facilities						
Cells	20	20	10			
Dining halls	100	15	8			
Guard stations	40	15	8			

* Table 2 prescribes supply rates of acceptable outdoor air required for acceptable indoor air quality. These values have been chosen to dilute human bioeffluents and other contaminants with an adequate margin of safety and to account for health variations among people and varied activity levels.

** Net occupiable space.

TABLE 2
OUTDOOR AIR REQUIREMENTS FOR VENTILATION*
2.1 COMMERCIAL FACILITIES (offices, stores, shops, hotels, sports facilities)

Application	Estimated Maximum** Occupancy P/1000 ft^2 or 100 m^2	Outdoor Air Requirements				Comments
		cfm/ person	L/s· person	cfm/ft^2	L/s·m^2	
Dry Cleaners, Laundries						Dry-cleaning processes may require more air.
Commercial laundry	10	25	13			
Commercial dry cleaner	30	30	15			
Storage, pick up	30	35	18			
Coin-operated laundries	20	15	8			
Coin-operated dry cleaner	20	15	8			
Food and Beverage Service						
Dining rooms	70	20	10			
Cafeteria, fast food	100	20	10			
Bars, cocktail lounges	100	30	15			Supplementary smoke-removal equipment may be required.
Kitchens (cooking)	20	15	8			Makeup air for hood exhaust may require more ventilating air. The sum of the outdoor air and transfer air of acceptable quality from adjacent spaces shall be sufficient to provide an exhaust rate of not less than 1.5 cfm/ft^2 (7.5 L/s·m^2).
Garages, Repair, Service Stations						
Enclosed parking garage				1.50	7.5	Distribution among people must consider worker location and concentration of running engines; stands where engines are run must incorporate systems for positive engines exhaust withdrawal. Contaminant sensors may be used to control ventilation.
Auto repair rooms				1.50	7.5	
Hotels, Motels, Resorts, Dormitories				cfm/room	L/s·room	Independent of room size.
Bedrooms				30	15	
Living rooms				30	15	
Baths				35	18	Installed capacity for intermittent use.
Lobbies	30	15	8			
Conference rooms	50	20	10			
Assembly rooms	120	15	8			
Dormitory sleeping areas	20	15	8			See also food and beverage services, merchandising, barber and beauty shops, garages.
Gambling casinos	120	30	15			Supplementary smoke-removal equipment may be required.
Offices						
Office space	7	20	10			Some office equipment may require local exhaust.
Reception areas	60	15	8			
Telecommunication centers and data entry areas	60	20	10			
Conference rooms	50	20	10			
Public Spaces				cfm/ft^2	L/s·m^2	
Corridors and utilities				0.05	0.25	
Public restrooms, cfm/wc or cfm/urinal		50	25			Normally supplied by transfer air.
Locker and dressing rooms				0.5	2.5	Local mechanical exhaust with no recirculation recommended.
Smoking lounge	70	60	30			
Elevators				1.00	5.0	Normally supplied by transfer air.

* Table 2 prescribes supply rates of acceptable outdoor air required for acceptable indoor air quality. These values have been chosen to dilute human bioeffluents and other contaminants with an adequate margin of safety and to account for health variations among people and varied activity levels.

** Net occupiable space.

TABLE C-1
STANDARDS APPLICABLE IN THE UNITED STATES FOR COMMON INDOOR AIR POLLUTANTS[a]

Pollutant	Indoor Standards	Outdoor Standards	Industrial Workplace Standards
Asbestos	Consumer Product Safety Commission has banned use of asbestos in artificial logs for fireplaces, in patching compounds, and in certain garments (16 CFR 1304, 1305, 1500.17(a)(7)); voluntary ban on use in hairdryers (C-10) EPA regulates use in schools (40 CFR 763.80) and removal projects (40 CFR 763.120); bans installation of friable asbestos for facility insulation (40 CFR 61.150) (C-11) (C-7)	National Emissions Standard: no visible emissions; may also comply by cleaning emissions as specified before particulate asbestos material escapes to air (EPA, 40 CFR 61.140 et seq.) (C-7) State air quality limits: CT 0.0010 µg/m^3 8 hr. MA 0.0001 fb/cm^3 24 hr. NC 0.0100 µg/m^3 24 hr. NY 5.0000 µg/m^3 1 yr. VA 2.0000 µg/m^3 24 hr. (all TLV based-fibers longer than 5 microns)* (NATICH Data Base, 1986) (C-8)	0.2 fiber/cm^3 8 hr. TWA (optically measured longer than 5 microns) (OSHA, 29 CFR 1910.1001(c) (C-12) 2.0 fiber/ml, 8 hr. TWA (Mine Safety and Health Admin., 30 CFR (56.5001(b), 57.5001(b)) (C-13)
Carbon Monoxide (See Table 1)		National Ambient Air Quality Primary Standard: 10 mg/m^3 (9 ppm) 8 hr. avg. 40 mg/m^3 (35 ppm) 1 hr. avg. (EPA, 40 CFR 50.8) (C9) State air quality limits: CT 10000. µg/m^3 8 hr. NV 1.3100 mg/m^3 8 hr. (NATICH Data Base, 1986) (C-8)	55 mg/m^3 (50 ppm) 8 hr. TWA (OSHA, 29 CFR 1910.1000, Table Z-1) (C-12) Mine Safety and Health Admin. uses ACGIH TLV* (30 CFR 57.5001(a)) (C-13)
Formaldehyde	Federal: 0.4 ppm target ambient level, HUD standard for manufactured homes, achieved through product emissions standards of .2 and .3 ppm (HUD, 24 CFR 3280.308, 1984) (C-14) State: 0.4 ppm standard for indoor exposure (MN statute 144.495, 1985) (C-15)	No federal standard. State air quality limits: CT 12.00 µg/m^3 8 hr. IL 0.0150 µg/m^3 1 yr. IN 18.00 µg/m^3 8 hr. MA 0.2000 µg/m^3 24 hr. NC 300.00 µg/m^3 15 min. NV 0.0710 mg/m^3 8 hr. NY 2.0000 µg/m^3 1 yr. VA 12.000 µg/m^3 24 hr. (NATICH Data Base, 1986) (C-8)	1 ppm 8 hr. TWA-PEL[b] 2 ppm 15-min. STEL[c] (OSHA, 29 CFR 1910.1000, Table Z-2; OSHA issued a final rule Dec. 4, 1987 (52 FR 46168) lowering a previous standard to the above levels, which was effective on Feb. 2, 1988). Mine Safety and Health Admin. uses ACGIH TLVs (30 CFR 57.5001(a)) (C-13)
Lead (See Table 1)	CPSC has banned in paint for consumer use or uses on consumer products (16 CFR 1303) (C-10)	National Ambient Air Quality Primary and Secondary Standard: 15 µg/m^3 max. arithmetic mean over calendar qtr. (EPA 40 CFR 50.12) (C-9) State air quality limits: CT 1.500 µg/m^3 8 hr. IL 0.500 µg/m^3 24 hr. MA 0.680 µg/m^3 24 hr. NV 0.004 mg/m^3 8 hr. VA 2.500 µg/m^3 24 hr. (NATICH Data Base for lead powder, 1986) (C-8)	50 µg/m^3 8 hr. TWA (OSHA, 29 CFR 1910. 1025(c)) (C-12) Mine Safety and Health Admin. uses ACGIH TLV (30 CFR 57.5001(a)) (C13)
Nitrogen Dioxide (See Table 1)		National Ambient Air Quality Primary and Secondary Standards: 100 µg/m^3 (0.053) ppm annual arithmetic mean (EPA, 40 CFR 50.11) (C-9) State air quality limits: CT 120.0 µg/m^3 8 hr. NV 0.143 mg/m^3 8 hr. (NATICH Data Base, 1986) (C-8)	(5 ppm) ceiling 9 mg/m^3 (OSHA, 29 CFR 1910.1000, Table Z-1) (C12) Mine Safety and Health Admin. uses ACGIH TLV (30 CFR 57.5001(a)) (C-12)

[a] Most ACGIH TLVs are referenced in western nations' standards, including Canada, Western Europe, and Australia.
[b] PEL—Permissible exposure limit.
[c] STEL—Short-term exposure limit.

TABLE C-1
STANDARDS APPLICABLE IN THE UNITED STATES FOR COMMON INDOOR AIR POLLUTANTS[a] (Continued)

Pollutant	Indoor Standards	Outdoor Standards	Industrial Workplace Standards
Ozone (See Table 1 and 3)	FDA prohibits devices (e.g., germicides, deodorizers) that result in more than 0.05 ppm in occupied enclosed spaces such as homes, offices, or hospitals, or that result in any releases in places occupied by the ill or infirm (21 CFR 801.415) (C-16)	National Ambient Air Quality Primary and Secondary Standards: 235 $\mu g/m^3$ (0.12 ppm) max. hourly avg. (EPA, 40 CFR 50.9) (C9) State air quality limits: CT 235.0 $\mu g/m^3$ 1 hr. NV 0.005 mg/m^3 8 hr. (NATICH Data Base, 1986) (C-8)	0.2 mg/m^3 (0.1 ppm) 8 hr. TWA (OSHA, 29 CFR 1910.1000, Table Z-1) (C12) Mine and Safety and Health Admin. uses ACGIH TLV (30 CFR and 57.5001(a)) (C-13)
Particulates (See Table 1)		National Ambient Air Quality Primary Standard: 75 $\mu g/m^3$ annual geom. mean 260 $\mu g/m^3$ maximum 24 hr. Secondary Standard: 60 $\mu g/m^3$ annual geom. mean 150 $\mu g/m^3$ maximum 24 hr. (EPA, 40 CFR 50.6, 50.7) (C-9)	
Radon (See Table 3)		National Emission Standard for Radon-222 emissions from underground uranium mines—requires bulkhead construction (EPA, 40 CFR 61.22) (C-7) National Emissions Standard for Radionuclide Emissions (excluding Radon-220, 222) from DOE facilities, other federal facilities, and NRClicensed facilities: 25 mrem/y whole body 75 mrem/7 critical organ (EPA, 40 CFR 61.92, 61.102) (C-7)	1.0 WL radon progeny maximum 4 WLM radon progeny calendar year (Mine Safety and Health Admin., 30 CFR 57.5038, 57.5039) (C-13)
Sulfur Dioxide (See Table 1)		National Ambient Air Quality Primary Standard: 80 $\mu g/m^3$ (0.03 ppm) annual arithmetic mean 365 $\mu g/m^3$ (0.14 ppm) 24 hr. Secondary Standard: 1300 $\mu g/m^3$ (0.5 ppm) 3 hr. (EPA, 40 CFR 50.4, 50.5) (C-9) State air quality limits: CT 860.0 $\mu g/m^3$ 8 hr. NV 0.119 mg/m^3 8 hr. TN 1.200 $\mu g/m^3$ 1 yr. (NATICH Data Base, 1986) (C-8)	13 mg/m^3 (5 ppm) 8 hr. TWA (OSHA, 29 CFR 1910.1000, Table Z-1) (C12) Mine Safety and Health Admin. uses ACGIH TLV (30 CFR 57.5001(a)) (C-13)

[a] Most ACGIH TLVs are referenced in western nations' standards, including Canada, Western Europe, and Australia.

TABLE C-2
GUIDELINES USED IN THE UNITED STATES FOR COMMON INDOOR AIR POLLUTANTS

Pollutant	Indoor Guidelines	Outdoor Guidelines	Industrial Workplace Guidelines
Asbestos			0.2-2.0 fibers/cm^3 8 hr. TLV-TWA (depending on type of fiber) (fibers longer than 5 microns) (ACGIH, 1986-87) (C-1)

TABLE C-2
GUIDELINES USED IN THE UNITED STATES FOR COMMON INDOOR AIR POLLUTANTS (Continued)

Pollutant	Indoor Guidelines	Outdoor Guidelines	Industrial Workplace Guidelines
Carbon Monoxide			55 mg/m^3 (50 ppm) 8 hr. TLV-TWA 440 mg/m^3 (400 ppm) 15 min. STEL (ACGIH, 1986-87) (C-1)
Chlordane	NAS recommendation for military housing: (C-17) 5 μg/m^3 maximum		
Formaldehyde			1.5 mg/m^3 (1 ppm) 8 hr. TLV-TWA 3 mg/m^3 (2 ppm) 15 min. STEL (ACGIH, 1986-87) (C-1) 1.2 mg/m^3 (1 ppm) 8 hr. TWA 2.5 mg/m^3 (2 ppm) 15 min. STEL (American Industrial Hygiene Assn., 1986) (C20) NAS recommendations for manned spacecraft: (C-18) 1.0 mg/m^3 (1.0 ppm) 60 min. 0.1 mg/m^3 (0.1 ppm) 90 days 0.1 mg/m^3 (0.1 ppm) 6 mo. Navy Submarine Atmospheric Control Manual, levels set by Naval Research Laboratory: (C-19) 3.0 ppm 1 hour 1.0 ppm 24 hour 0.5 ppm 90 days
Lead Dust and Fumes			0.15 mg/m^3 8 hr. TLV-TWA (ACGIH, 1986-87) (C-1)
Nitrogen Dioxide			6 mg/m^3 (3 ppm) 8 hr. TLV-TWA 10 mg/m^3 (5 ppm) 15 min. STEL (ACGIH, 1986-87) (C-1) NAS recommendation for manned spacecraft: (C-18) 4 mg/m^3 (2.0 ppm) 60 min. 1.0 mg/m^3 (0.5 ppm) 90 days 1.0 mg/m^3 (0.5 ppm) 6 mo.
Ozone			0.2 mg/m^3 (0.1 ppm) 8 hr. TLV-TWA 0.6 mg/m^3 (0.3 ppm) 15 min. STEL (ACGIH, 1986-87) (C-1)
Radon (See Table 3)	EPA 1986 recommendation for homes: 4 pCi/l or less-can be reached in most homes At 4-20 pCi/l-take action to reduce within a few years At 20-200 pCi/l-reduce within several months At 200 pCi/l or above, reduce within several weeks or relocate until levels are reduced (EPA, "A Citizen's Guide to Radon," August 1986) (C-21) (EPA, "Radon Reduction Methods, A Homeowner's Guide," August 1986) (C22)		
Sulfur Dioxide			5 mg/m^3 (2 ppm) 8 hr. TLV-TWA 10 mg/m^3 (5 ppm) 15 min. STEL (ACGIH, 1986-87) (C-1) NAS recommendation for manned spacecraft (C-18) 13 mg/m^3 (5.0 ppm) 60 min. 3 mg/m^3 (1.0 ppm) 90 days 3 mg/m^3 (1.0 ppm) 6 mo.

TABLE C-3*
SUMMARY OF CANADIAN EXPOSURE GUIDELINES FOR RESIDENTIAL INDOOR AIR QUALITY

Contaminant	Acceptable Exposure Ranges		Page
	ASTER[g]	ALTER[h]	
Aldehydes (total)	$\Sigma c_i/C_i<1$[a]	—	16
Carbon dioxide	—	× 6 300 mg/m³ (<3 500 ppm)	17
Carbon monoxide	<11 ppm—8 h[b] <25 ppm—1 h[b]	—	18
Formaldehyde	[c]	[d]Action Level 120 µg/m³ (0.10 ppm) Target Level 60 µg/m³ (0.05 ppm)	26
Nitrogen dioxide	<480 µg/m³ (<0.25 ppm)—1 h	<100 µg/m³ (<0.05 ppm)	19
Ozone	<240 µg/m³ (<0.12 ppm)—1 h	—	21
Particulate matter[e]	<100 µg/m³—1 h	<40 µg/m³	21
Sulphur dioxide	<1 000 µg/m³ (<0.38 ppm)—5 m	<50 µg/m³ (<0.019 ppm)	23
Water vapor	30-80% R.H.—summer 30-55% R.H.—winter[f]	—	24
Others See Below[i]	Minimize exposure	Minimize exposure	28-36

[a] C_i = 120 µg/m³ (formaldehyde); 50 µg/m³ (acrolein); 9000 µg/m³ (acetaldehyde), and C_i are respective concentrations measured over a five-minute period.
[b] Units given only in parts per million so that guidelines are independent of ambient pressure.
[c] See Aldehydes (total).
[d] Although the epidemiological studies conducted to date provide little convincing evidence that formaldehyde is carcinogenic in human populations, because of this potential, indoor levels should be reduced as much as possible.
[e] <2.5 µm mass median aerodynamic diameter—MMMD.
[f] Unless constrained by window condensation.
[g] ASTER—Acceptable short-term exposure range.
[h] ALTER—Acceptable long-term exposure range.
[i] Other contaminants of concern; biological agents; chlorinated hydrocarbons; pest control products; product aerosols; fibrous materials; lead; polycyclic aromatic hydrocarbons (PAHs); and tobacco smoke.
* From Reference C-23, p.41.

TABLE C-4
WHO WORKING GROUP CONSENSUS OF CONCERN ABOUT INDOOR AIR POLLUTANTS AT 1984 LEVELS OF KNOWLEDGE*

Pollutant[a]	Concentrations[b] reported	Concentrations[b] of limited or no concern	Concentration[b] of concern	Remarks
Tobacco smoke (passive smoking)				
Respirable particulates	0.05-0.7	<0.1	>0.15	Japanese standard 0.15 mg/m³
CO	1-1.5	<2	>5	indicator for eye irritation (only from passive smoking)
Nitros-dimethylamine	$(1\text{-}50) \times 10^{-6}$	—	—	Mutagens under investigation fro carcinogenicity
NO_2	0.05-1	<0.19	>0.32	
CO		2% COHb	3% COHb	99.9%[c]
	1-100	<11	>30	Continuous exposure
Radon and Progenys	10-3000 Bq/m³	~0	70 Bq/m³	Swedish standard for new houses
Formaldehyde	0.05-2	<0.06	>0.12	Long- and short-term
SO_2	0.02-1	<0.5	>1.35	SO_2 alone short-term
CO_2	600-9000	<1800	>12000	Japanese standard 1800 mg/m³
O_3	0.04-0.4	0.05	0.08	
Asbestos	<10 fibres/m³	~0	10^5 fibre/m	For long-term exposure
Mineral fibers	<10 fibres/m³	—	—	Skin irritation
Organics				
Methylene chloride	0.005-1	—	350	TLV[d]
			260	NIOSH[e] recommendations
Trichlormethene	0.0001-0.02	—	270	TLV
			135	NIOSH recommendations
Tetrachloroethene	0.002-0.05	—	335	TLV
1.4 Dichlorobenzene	0.005-0.1	—	450	TLV
Benzene	0.01-0.04	carcinogen	carcinogen	
Toluene	0.015-0.07	—	375	TLV
m.p.-Xylene	0.01-0.05	—	435	TLV
n-Nonane	0.001-0.03	—	1050	ILO (1980)
n-Decane	0.002-0.04	—	—	
Limonene	0.01-0.1	—	560	TLV turpentine

[a] All gases were considered on their own without other contaminants.
[b] Typical ranges of concentration given in mg/m³ unless otherwise indicated and for short-term exposures.
[c] According to Environmental Health Criteria No. 4, Geneva World Health Organization, 1977.
[d] TLV (threshold limit values) established by the American Conference of Governmental Industrial Hygienists (1983-1984). These values are for occupation exposures and should be considered as extreme upper limits for nonoccupational populations for very short-term exposures.
[e] National Institute for Occupational Safety and Health (NIOSH), USA
[f] International Labor Organization (ILO)
— = no meaningful numbers can be given because of insufficient knowledge
* From Reference C-24, Table 3

ASHRAE 62.1-2004 Ventilation for Acceptable indoor Air Quality

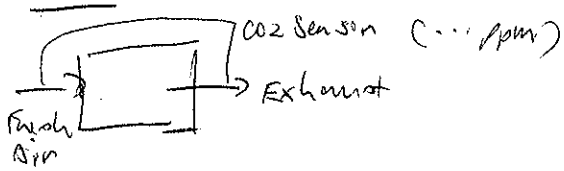
Feb 2008

ASHRAE Standard Table (Based on Ventilation Rate Procedure)

Application	Default occupancy	Default outdoor air cfm/person OA	Required Ventilation Rates	
			Cfm/person OA	Cfm/ft²
Office Space	5	17	5	0.06
Conference Room	50	6	5	0.06
Science Lab (school)	25	17	10	0.018

ASHRAE
1. Ventilation Rate Procedure
2. IAQ (CO2)

IAQ



Air Rates

- Under Paragraph 6.2, in its simplest form, the amount of air to be supplied is estimated by an equation similar to that shown below :

$$Q_{OA} = Q_P + Q_b = R_P P_d + R_b A_b$$

where

Q_{OA} = *design OA ventilation rate*

Q_p = *ventilation required to control "people" sources of air contamination*

R_P = *minimum rate of outdoor air in cfm per person*

P_d = *number of people in the occupied area*

R_b = *minimum rate of outdoor air in cfm per square foot of occupied area*

A_b = *floor area in occupied portion of the building , square feet*

Example 1

- In an office space of A = 1,000 sq feet with 7 occupants, the minimum required ventilation rate to the occupant zone is :

ASHRAE

office ⇒ $R_b = 0.06$ cfm/ft²

$R_p = 5$ cfm/person

$$Q_{OA} = R_P P_d + R_b A_b$$

$$(5)(7) + (0.06)(1000) = 95 \text{ cfm, fresh outdoor air}$$

This results in an OA rate, $Q_{OA} = 95/7 = 13.6$ *cfm / person, slightly less than the older standard minimum* (15 *cfm OA*) *but considerably below the previous office requirement of* 20 *cfm OA / person*

Example 2

- For conference rooms the proposed values are :

$R_p = 5\ cfm/person \quad R_b = 0.06\ cfm/sq\ feet$

For a small conference room of $A = 500\ sq\ feet$ *occupied by 25 occupants, the minimum required ventilation rate is :*

$Q_{OA} = R_P P_d + R_b A_b$

$= (5)(25)(1) + (0.06)(500) = 155\ cfm,$ *fresh outdoor air*

This results in a OA rate,

$Q_{oa} = \frac{155}{25} = 6.2\ cfm$, *considerably less than the previous standard* (15 *cfm OA/ person*)

Example 3

- For example 1, if the ventilation effectiveness factor for air distribution and mixing is estimated to be 80%, or E= 0.8, what is the required total outdoor ventilation rate ?

$Q_{TOA} = \frac{Q_{OA}}{E} = \frac{95}{0.8} = 119\ cfm,$ *fresh outdoor air*

This results in an OA rate of $Q_{oa} = \frac{119}{7} = 17\ cfm\ per\ person$

a little below the former standard for offices of 20 *cfm OA/ person*

Exercise

- A building has four HVAC zones supplied by a single air handling unit (AHU). Each zone contains 10 offices staffed by 2 people per office. The building is non-smoking. At 20 cfm OA per person, how much outdoor air (OA) must supplied to each office and zone, and by the AHU ?

SOLN

- The optimum situation is to supply the correct amount of OA to each person in the building.
- Four zones with 10 offices of 2 persons each
- = 4 x 10 x 2 = 80 people.
- Older versons of the standard called for 20 cfm/person OA in offices.
- Therefore, the AHU must supply a minimum of 80 x 20 = 1600 cfm OA.
- Furthermore, each zone should supply 400 cfm OA , and each office should receive 40 cfm OA.
- Hopefully ,each person would get his/her share of the OA . Whether this is possible will have to be determined on a case by case basis. Obviously, some reasonableness must be applied where exact distribution is not possible.

Exercise 2

- What are the two compliance procedure suggest by the Standard, and which of the two methods do you think is a better approach to providing acceptable IAQ ? Why ?

soln

- The first method is called the Ventilation Rate Procedure and relies on specified amount of OA and maintaining carbon dioxide level below "700" ppm.
- The second is called the Indoor Air Quality Procedure and relies on actually keeping contaminant concentrations below certain specified levels. The second method has been difficult to apply in practice. It is expensive to provide monitoring.
- Building occupants and designers usually emphasize the first method and insist that it be used.

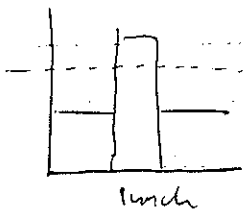
~~Note~~
Casino ~~air~~ air can recycle but not allowed to go to other rooms ∴ smoking.

2/20/2008

Exercise 3

- A smoking lounge is to be installed in an office building. About 3 people will be smoking at any one time, as an average. However, at lunch and during breaks, as many as 30 persons may smoke. If 80 cfm/person is determined to be the desired dilution flowrate, what total OA airflow should be supplied to the room ? What other provisions would be appropriate ?

ASHRAE



Soln

- On the daily average 3 x 80 = 240 cfm OA .
- However, during breaks and lunch, 30 x 80 = 2400 cfm OA.
- The exact "correct" approach is not specified anywhere. However, it seems logical that 240 cfm will not be sufficient for the lunch and break periods. Perhaps a two speed system could be installed.
- Perhaps a time-weighted amount could be provided calculated from use/time relationships.
- For example , 6 hrs x 3 = 18 smoker-hrs;
- 2 hrs x 30 = 60 smoker-hrs; adding the smoker-hrs = 78 smoker-hrs; dividing by 8 = 10 smokers, average = 10 x 80 = 800 cfm OA as the average required.
- Other provisions include (1) isolating the ventilation system from the rest of the building, and (2) providing direct exhaust of the contaminated air.

Designing a dedicated ventilation system to a smoking area

1. Determine the actual boundaries of the proposed smoking area . (This is the total space where air moves, not an area designated on paper). Estimate the total room volume of the smoking area in cubic feet, V_{Space}.
2. Estimate the maximum number of persons in the smoking area at any one time, P_S.
3. Estimate the required fresh outdoor air required ($Q_{OA\text{-}req}$)
4. Determine the design (or actual) recirculation percentage at the air handling system, R (e.g. 0.80) . Alternatively, determine the design (or actual) OA percentage introduced to the system, N (e.g. 0.20).
5. Estimate the total supply air rate in the smoking area, Q_{SA}.
6. Estimate the design (or actual) outdoor air supplied to the area :

   $Q_{OA\text{-}Actual}=Q_{SA}X\ (100\text{-}R)= Q_{SA}\ X\ N$

7. Compare $Q_{OA\text{-}req}$ to $Q_{OA\text{-}Actual}$. The latter should equal or exceed the former.
8. Estimate room air changes per hour, N

   $$N = \frac{Q_{SA}}{V_{Space}} X\ 60$$

   Also,

   $$N = \frac{Q_{OA}}{V_{Space}} \text{x60 for outside air only}$$

9. Check to see that the return air equals or exceed supply air to the smoking area
10. Check to see that air from the smoking area is returned directly (not after traveling through other occupied areas).
11. Check the mixing potential in the smoking area (walls, fans, etc)
12. Recommend or investigate potential alternations if necessary (e.g. moving partitions, installing fans, moving supply/exhaust registers, closing doors and windows).
13. Check to see if smoking area in under negative pressure (smoke tube, air flows).
14. Establish a comprehensive housekeeping program

Sample Calculation

It is determine that up to 5 people will occupy a designated smoking lounge, 10 ft x 10 ft x 8 ft. There is a separate, dedicated air supply and exhaust AHU serving the smoking area.

Q_{SA}=500 acfm , R= 80% (N=20%)

per person

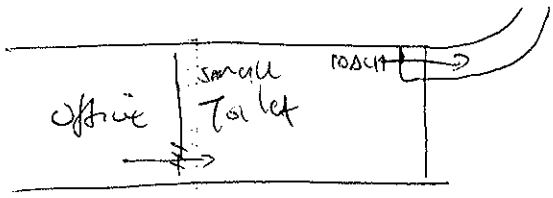
can reduce if grille is lower

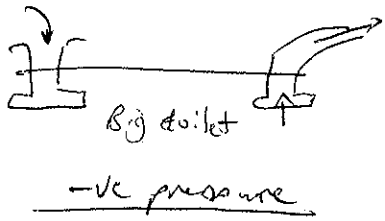
1. Room Volume = 10 x 10 x 8 = 800 cu ft; room area = 100 sq ft
2. P_S= 5 people
3. $Q_{OA\text{-}Req}$=5 x 60 = 300 cfm (ASHRAE 62-1989/99 STD)
4. R= 80% (0.80) or N = 20% (0.20)
5. Q_{SA}=500 cfm
6. $Q_{OA\text{-}actual}$=(500)(1.00 – 0.80)= 100 cfm (or, 500 x 0.20 = 100 cfm)
7. $Q_{OA\text{-}Actual}$ is not sufficient – it should be 300 cfm (alterations to the ventilation system will be required to increase the supply of fresh air to the space).
8. N=(Q_{SA} X 60)/V_{Space}=N=(500 X 60)/800 = 37.5 Air Changes per hour

   Also , N= (300 X 60)/800 = 22.5 OA Air Changes per hour (Okay)

Step (9 -14) As necessary. In this case, the recirculation rate R should be reduced to R = 0.40 , which

Will provide 60 cfm per person of outdoor air.





Smoking Lounge

Apr 2008

Design Criteria

- 1. Air from the smoking room is exhausted directly to the outside by an exhaust fan
- 2. No smoking room air is recirculated to other parts of the building.
- 3. Smoking room are in a non-work area where employees are not required to ne present as part of their work responsibilities other than a custodial or maintenance work when the room is unoccupied.

Example

- A smoking lounge is to be installed in a small room of a large laboratory/office building. Approximately 3 people will be smoking at any one time, as a daily average. However, at lunch and at break times, as many as 10 persons may smoke. What amounts of OA should be supplied to the lounge ? What other provisions are required ?

Soln,1

- The standard calls for 60 cfm/person OA in smoke lounges. On the daily average 3 x 60 = 180 cfm OA
- However, during breaks and lunch
- 10 x 60 = 600 cfm OA
- The exact "correct" approach is not specified in the standard. However, it seems logical that 180 cfm will not be sufficient for the lunch and break periods.
- Perhaps a two speed system could be installed.
- Perhaps a time-weighted amount could be provided, calculated from use/time relationships.

Soln,2

- For example:
- 6 hrs x 3 = 18 smoker-hrs
- 2 hrs x 10 = 20 smoker-hrs
- Adding the smoker-hrs = 38 smoker-hrs
- Dividing by 8 = about 5 smoker-hrs
- Average : 5 x 60 = about 300 cfm OA as the average required.

Existing Design Guidelines for Containment Rooms

- Many organizations have issued guidelines for negatively pressure rooms. These guidelines are based on field experience using smoke test methods:

1. A "Rule of thumb" of designing negatively pressurized rooms has been that a 10% differential between a room's supply and exhaust (or return) airflow is adequate to prevent room air leakage to adjoining spaces. This 10% rule of thumb is a hangover from an old ASHRAE guide.
2. The Francis J. Curry National Tuberculosis Center recommends that the negative pressure differential across Tuberculosis isolation rooms be approximately -7.5 Pa (0.03 in H_2O). The same Center recommends that exhaust should exceed supply by at least 47 l/s (100 cfm)

Existing Design Guidelines for Containment Rooms

- 3. CDC recommends 0.25 Pa (0.001 in.wc) negative pressure for TB isolation rooms and that the exhaust should be 10% or 24 l/s (50 cfm) greater than the supply.
- 4. The California Office of Statewide Health Planning and Development (OSHPD) and the California Mechanical Code recommend for TB isolation rooms the same pressure differential as CDC , but also specify that the exhaust should be 35 l/s (75 cfm) greater than the supply, that the room should be under negative pressure, and that the velocity at the "transfer opening" be 0.51 m/s (100 fpm)

Existing Design Guidelines for Containment Rooms

- 5. For laboratories , the American National Standard for Laboratory Ventilation ,specifies that where air must be contained, the exhaust and supply airflow rates must be maintained through any opening between the controlled space and adjoining areas, including open doorways, so that the following velocities be achieved at the opening :
- - minimum velocity :0.25 m/s (50 fpm); and
- - preferred velocity : 0.51 m/s (100 fpm)

Existing Design Guidelines for Containment Rooms

- 6. For areas undergoing asbestos containment, OSHA recommends that these areas be negatively pressurized at 5 Pa (0.02 in w.c.)
- 7. Wiseman recommends a minimum negative pressure of 2.5 Pa and advices a pressure 12 Pa or higher for "Critical areas".

Some data for smoking room

Room dimension	2.2 m x 4.6 m x 2.4 m
Room floor area	10.2 m^2
Room volume	25 m^3
Door size	2.1 m x 0.89 m
Gap under door	0.64 cm
Supply flow rate	3- 54 l/s
Supply flow per flow area	0.05 – 1.1 cfm/ft^2
Exhaust flow	13- 99 L/s
Percentage of outside air	100%
ACH	0.4 to 7.9
Calculated velocity at door opening with door open (at 99 l/s exhaust)	0.053 m/s
Ceiling Plenum Height	23 cm
Linear feet of ceiling tiles	61 m (200 ft)

PRACTICAL GUIDE

HEAT GAIN/COOLING LOAD FROM ELECTRICAL EQUIPMENT

Electrical distribution equipment (transformers, wiring, panel boards, etc.) is among the most efficient of all equipment in buildings, but it is not 100% efficient. The inefficiencies show up as waste heat, which HVAC&R engineers need to quantify and factor into their load calculations.

Although the electrical resistance of wires, bus ducts, switches, and other electrical equipment is very small, it is not zero. Therefore, all current flow generates waste heat.

The rate of waste heat generation from any piece of electrical equipment is a function of Joule's Law:

$W = I^2 * R$

where

W = power in watts. In the inch-pound system of measurement, heat gain in watts converts to BTU/h by multiplying by 3.413.

I = current in amperes

R = electrical resistance in ohms

If the room is large and the waste heat gain is small, cooling might not be required. Even in a large room, the amount of electrical equipment can require that the HVAC/R designer include waste heat from electrical equipment in the cooling load. A dedicated electric room can easily have enough waste heat gain to require some form of cooling to maintain a safe room temperature for the equipment.

When a room requires cooling for equipment but not for personnel, a space temperature 10°F to 15°F (5°C to 8°C) higher than the adjacent occupied space is often acceptable. That condition typically applies to electric rooms in office buildings. In those cases, ventilation using air from the adjacent conditioned space might provide adequate cooling.

As with mechanical cooling, the amount of airflow required depends on the room heat gain and the allowable temperature rise above the supply air temperature. The following formula[1] quantifies that relationship:

*BTUH = 1.1 * CFM * TD (I-P)*

or

*W = 1.232 * L/s * TD (SI)*

where

BTUH, W = room sensible cooling load

CFM, L/s = airflow

TD = temperature difference (F or C)

1.1, 1.232 = conversion factors

With cooling by ventilation from adjacent conditioned space, the supply air temperature is the room temperature in the adjacent space.

For example, an electric room with a 75 KVA transformer operating at a continuous 40 KVA load has a waste heat gain of approximately 2.5% of transformer full load capacity or 1875 watts (obtained from *Figure I-1*). To keep the electric room no warmer than 90°F (32°C) with a 75°F (24°C) adjacent conditioned space, an exhaust fan would have to transfer approximately 400 CFM (200 L/s) from the conditioned space through the electric room.

The cooling load calculation described here is based on long term, steady state conditions. It does not take into account thermal flywheel effects or conduction losses from the warm electric room through usually uninsulated partitions to the adjacent conditioned space. Considering those factors plus the fact that the load on the transformer varies throughout the day, somewhat less airflow than calculations predict for full load, steady state might do the job.

Although the electrical resistance of wires, bus ducts, switches, and other electrical equipment is very small, it is not zero. Therefore, all current flow generates waste heat.

Motors

The inefficiency in a motor is the difference between the energy input and the amount of energy converted to

mechanical work. That inefficiency becomes waste heat to the space. The inefficiency can be expressed in watts, which the HVAC/R engineer can include in the space cooling load.

For example, a 25 HP (18.65 KW output), premium efficiency motor has a NEMA nominal efficiency of 94.1%. The power input to that motor is 18.65 KW output/94.1% efficiency or 19.82 KWe input. The inefficiency is 19.82 KW input minus 18.65 KW output, or 1.17 KW (3993 BTU/hr in I-P units).

Motor efficiency varies with motor load. Motors are usually designed for peak efficiency at about 75% of nameplate load with only small variations in efficiency between 50% load and 100% load. At light loads, motor efficiency drops off. For example, a 94.1% full load efficient 25 HP motor is only 78% efficient at 25% load. The actual waste heat gain to the space is a function of the actual load on the motor and the motor efficiency at that operating point.

If the driven equipment or process is in the conditioned space, the energy the motor converts to mechanical work also becomes part of the space cooling load. Mixing operations and machine shops are examples where both the inefficiency and the mechanical work of the motor become part of the space cooling load.[2] The mechanical energy from pumps and fans ultimately becomes a waste heat load that HVAC&R engineers must recognize in their system load calculations.[3]

Transformers

Like motors, transformers have no-load losses and load losses. The no-load losses in a transformer derive from the energy necessary to magnetize the core, or hysteresis losses. As the name implies, no-load losses are independent of the load on the transformer. Since transformers usually remain energized all the time and are not switched on and off, no-load losses are constant over 8760 hours per year.

Load losses are Joule heating or "I^2R" losses that result from current flow through the transformer windings. Since losses vary with the square of current, load losses (expressed as a percent of full load) fall off quickly as load decreases.

Figure I-1 shows typical transformer loss data for dry type, 480 volt primary, 208Y/120 volt secondary transformers at loads ranging from 0% to 100% of full load.[4] This type of transformer is the most common in office buildings, laboratories, and light manufacturing plants.

The loss data are typical. Actual losses may vary from one manufacturer to another and within any one manufacturer's product line. The National Electrical Manufacturer's Association (NEMA) is developing a standard for transformer efficiency, with a draft expected toward the end of 1996.

Medium voltage transformers and unit substations used to transform voltage from utility distribution voltage (usually between 4000 volts and 15,000 volts) to building utilization voltage (usually 480 volts or 208 volts) will have slightly different loss characteristics.

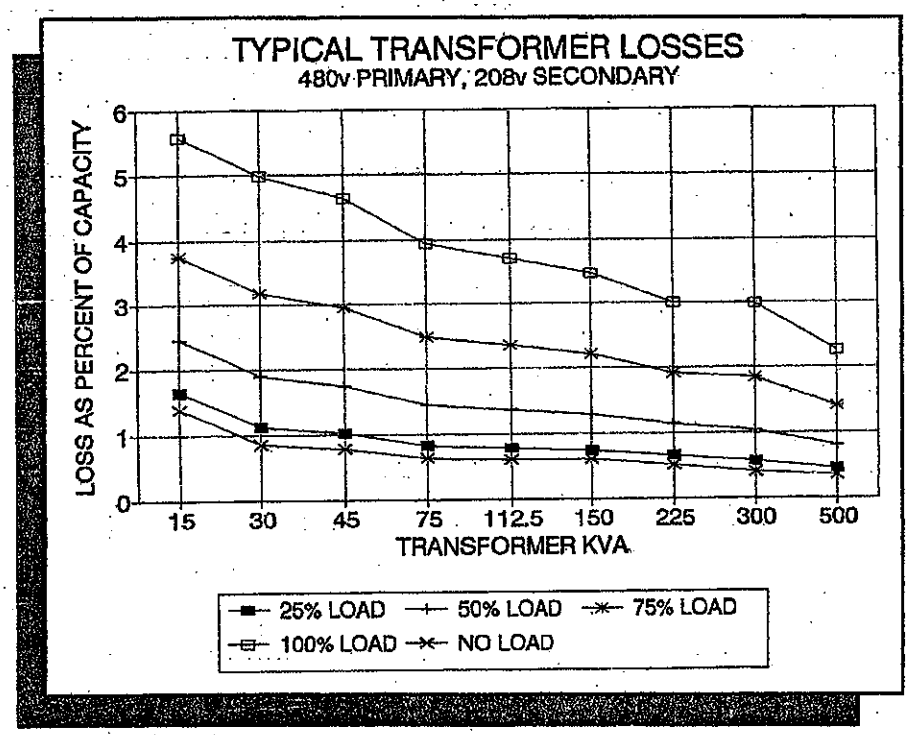


Figure I-1: Typical transformer losses as percent of capacity.

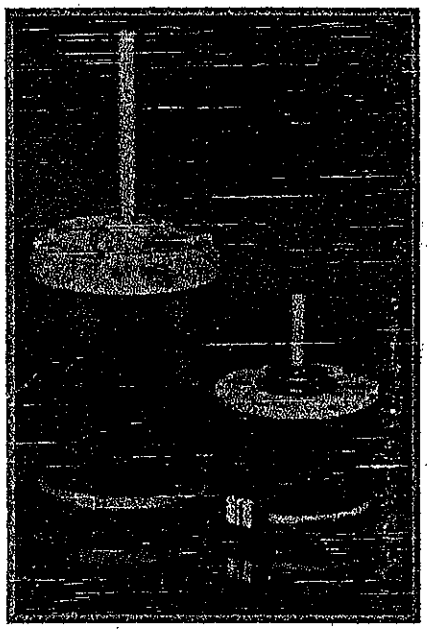
Switchgear

In switchgear, the conductors are large and the lengths are short so the overall resistance is small. However, the current flows can be large. With heat loss varying as the square of current, large switchgear can generate appreciable waste heat. Indeed, conventional (non-electronic) circuit breakers detect long term overloads by using current flow to generate heat inside the circuit breaker.

Heat loss from switchgear and circuit breakers varies with breaker frame size and load. The following table[5] presents typical ranges of losses for circuit breakers loaded to 80% of nominal rating:

Frame Size	Watts Loss
100 amp	8 - 15
225 amp	13 - 40
400 amp	19 - 50
800 amp	19 - 50
1200 amp	38 - 150

Like transformers, individual circuit breakers are rarely loaded to capacity. For example, common design practice puts only 8 to 12 amps of connected load on a 20 amp lighting circuit. Receptacle circuits might have a higher connected load, but the actual load at any time is usually quite low. Because waste heat load is a function of current squared, a breaker with a continuous



load of 50% of breaker rating rather than the 80% referenced in the table has a waste heat load of only 40% of the table values [(50%/80%)^2 = 40%].

Motor Control Centers

Motor control centers are like switchgear in that they have high current flows concentrated in a small area. In addition, motor control centers have relays and control power transformers that generate waste heat.

On average, 250 watts per lineal foot of lineup is a reasonable estimate for waste heat losses from a motor control center.[6] That allowance includes the starters, control transformers, and bus. It recognizes that motor control centers often include standby equipment, so not all starters are energized at once.

Starters located near the motors they control also have waste heat losses. The following table presents typical losses for individual combination starters, including circuit breakers and control power transformer:[7]

NEMA Size	Watts Loss
1	27
2	57
3	130
4	200
5	300
6	650

1. Adapted from Handbook, Fundamentals, ASHRAE 1993, page 26.15, equation 25.

2. See ASHRAE Handbook--Fundamentals, 1993, pages 26.9-10 and Table 4.

3. For a thorough analysis of fan and pump heat including a discussion of why fan heat results in a temperature rise across the fan only while pump heat results in a temperature rise distributed along the system, see Williams, GJ, "Fan Heat: Its Source and Significance", Heating/Piping/Air Conditioning, January 1989, pp. 101-112.

4. Data provided by Square D Company.

5. *Data from* Field Testing Industrial Molded Case Circuit Breakers, Square D Company, 1988, and Application and Selection Guide - GE 8000 Line Motor Control Centers, General Electric Company, Publication GET-6728C, page J-15, 1992.

6. Application and Selection Guide - GE 8000 Line Motor Control Centers, General Electric Company, Publication GET-6728C, page J-15, 1992.

7. Application and Selection Guide - GE 8000 Line Motor Control Centers, General Electric Company, Publication GET-6728C, page J-15, 1992. ❖

Carparks

Objective of Ventilation of Carpark

- To remove CO and other combustion products
- Full carpark mechanical ventilation – 6 ACH
- Can be operated at a lower ACH rate during low occupancy period but CO level <25 ppm averaged over 8 hrs period

Residential Carpark (condo min)

- The mechanical ventilation system may be switched off whenever the CO< 9 ppm or the ambient level
- CO sensor monitoring system with comparator could be used to on/off fan if required to save energy and eliminate noise generated by fans

Carpark Ventilation

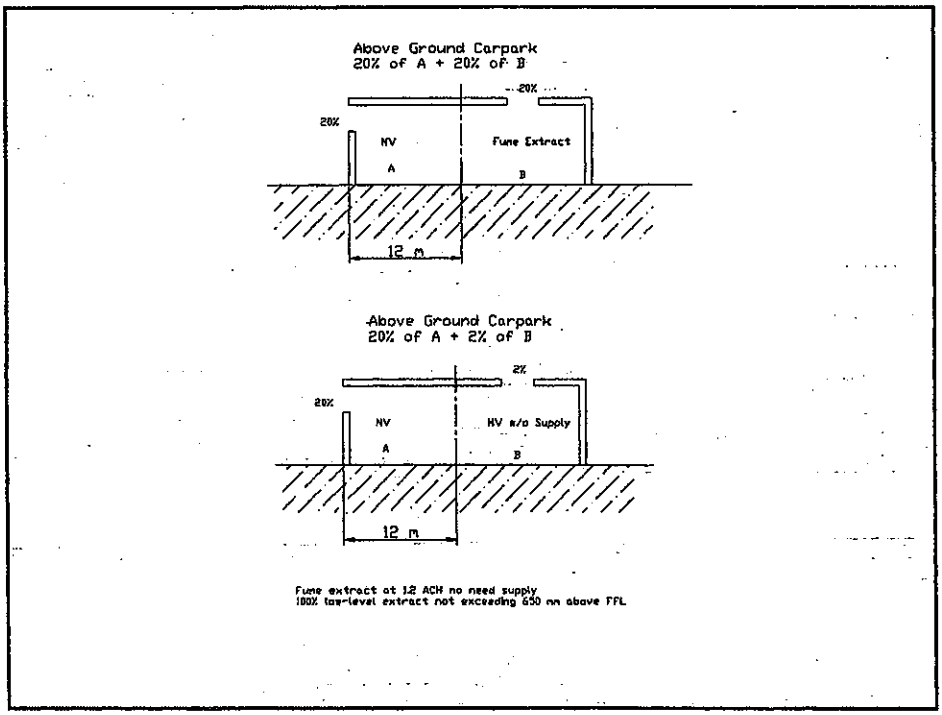
- Carpark area > 1800 m^2 , ventilation is required to lower the limits of petrol vapor inflammability
- Limit the concentration of CO to100 ppm
- 6 to 10 ACH
- Large carpark can be treated as having 2 Zones, Zone A and B.

Types of Carparks

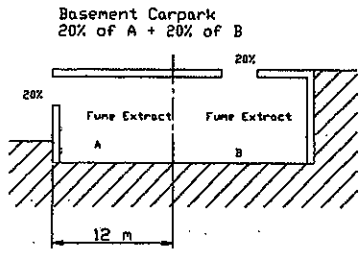
- 1. Above ground Carpark
- 2. Basement Carpark

Table : Mode of Ventilation for aboveground carpark

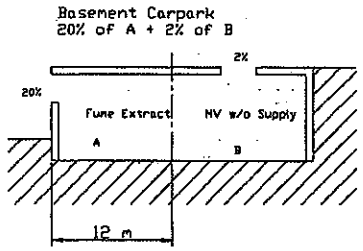
Size of Vent Opening (% of floor area)	Mode of Ventilation to be provided (NV, MV , or Fume Extract)	
	Zone "A"	Zone 'B"
20% of A + 20% of B	NV	Fume Extract
20% of A + 2% of B	NV	MV w/o Supply
20% of A	NV	MV
2% of A 2% of B	MV w/o Supply	MV w/o Supply
2% of A	MV w/o Supply	MV



Carpark must be under -ve pressure



1.2 ACH



Fume extract at 12 ACH no need supply
100% low-level extract not exceeding 650 mm above FFL

Table : Mode pf ventilation for basement carpark

Size of Vent Opening (% of floor area)	**Mode of Vent to be provided (MV or Fume Extract)**	
	Zone "A"	Zone "B"
20% of A + 20% of B	Fume extract	Fume extract
20% of A + 2% of B	Fume extract	MV w/o Supply
20% of A	Fume extract	MV
2% of A + 2% of B	MV w/o Supply	MV w/o Supply
2% of A	**MV w/o Supply**	MV

Carpark Smoke Purging System

- 9 ACH in basement car parks with floor area > 1900 m^2
- The duct should be of heavy gauge sheet (1.2 mm thick) and can withstand 250°C for 2 hours
- The exhaust fan shall be capable of operating effectively at 250°C for two hours.
- Supply fans: Normal duty
- The system shall be independent of any other supply and exhaust systems serving other parts of the building

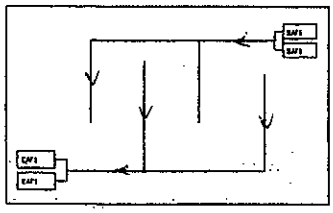
with standby genset

supply not necessary, exhaust need to 1.2 mm thick to ~~provide~~ last 2 hrs during fire

Carpark Ventilation System Design

Consist of a supply and an exhaust part, each part to operate in two sections conforming to the following requirements:

Carpark design



Using two speed fan.

If volume of carpark $= V\ m^3$

Supply air flowrate $= 9xV\ m^3 / hr$

Exhaust air $= 1.15\ x9V\ m^3 / hr$

Each Supply air fan capacity $= \frac{1}{2} x\ 9V\ m^3 / hr$

Each Exhaust air fan $= \frac{1}{2} x1.15x9V\ m^3 / hr$

Carpark Ventilation system design

- The capacity of each section shall be sufficient to provide half the required air change of outdoor air
- The two sections may operate through a common duct
- In the event of failure of one section, the other shall continue operating
- The ventilation fan shall continue to run even if the principal source of electrical supply fails. This is done by connecting the system to the standby generator

- The failure of an exhaust section shall cause the corresponding supply section to be shut down. This is to prevent noxious gases being transported to adjacent areas of the building.
- The quantity of the replacement air shall not exceed the exhaust air
- Extract points should be positioned so those fumes cannot accumulate in any one place. The air should be exhaust from both high and low levels.
- At least 50% of the extraction points takes place at low level, not above 650 mm above finish floor level, measured from the top of the grilled to the finished floor.

- The exhaust points of air discharge shall be sufficient far away, at least 5 m from any doorways and intake openings to prevent re-entry of objectionable odours of inflammable vapors.
- For natural ventilation, openings in the outside walls should have an aggregate area equal to at least 20% of the floor area at each level ,and at least half of that area should be at opposite sides.

Good Design

- Fresh air supply should be by natural ventilation if possible
- The time of travel from the supply to the exhaust duct intake should be uniformly at all paths lower than 10 minutes
- No stagnant spot
- More extraction rates at the carpark entrance (because people stay longer for payment, etc)

US and International standards for ventilation requirements of enclosed parking garages

	Times (hours)	PPM	Ventilation
ASHRAE	8 1	9 35	7.61 l/sm^2 (1.5 cfm/ft^2)
ICBO	8 1	50 200	7.6 l/sm^2 (1.5 cfm/ft^2)
NIOSH/OSHA	8 Ceiling	35 200	---
BOCA	--	--	6 ACH
SBCCI	--	--	6-7 ACH

NFPA	--	--	6 ACH
ACGIH	8	25	--
CANADA	8 1	11/13 25/30	--
Finland	8 15 minutes	30 75	2.7 l/sm² (o.53 cfm/ft²)
France	Ceiling 20 Minutes	200 100	165 l/s car (350 cfm/car)
U.K.	8 15 minutes	50 100	6 -10 ACH

Case study 1
The air flow in carpark

the current design equation for unflanged rectangular hoods is :

$$Q = v(10X^2 + A)$$

$where\ Q = exhaust\ volume\ flow\ rate, m^3/s$

$v = control\ velocity, m/s$

$X = distance\ of\ hood\ from\ source, m$

$A = area\ of\ hood\ opening, m^2$

Figure 1A Carpark Ventilation Design

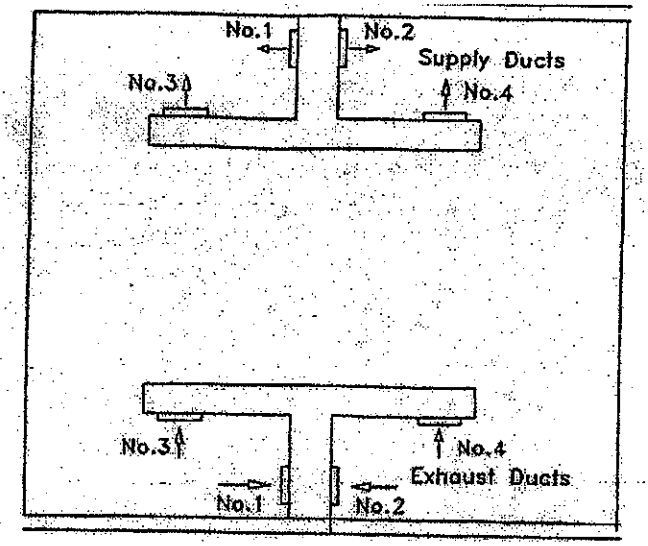
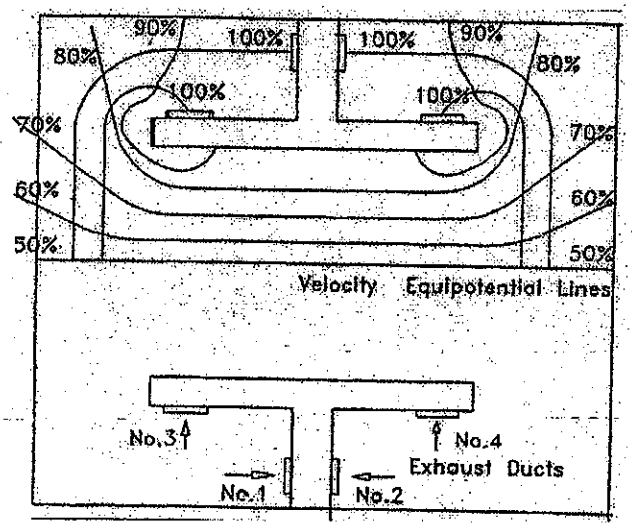


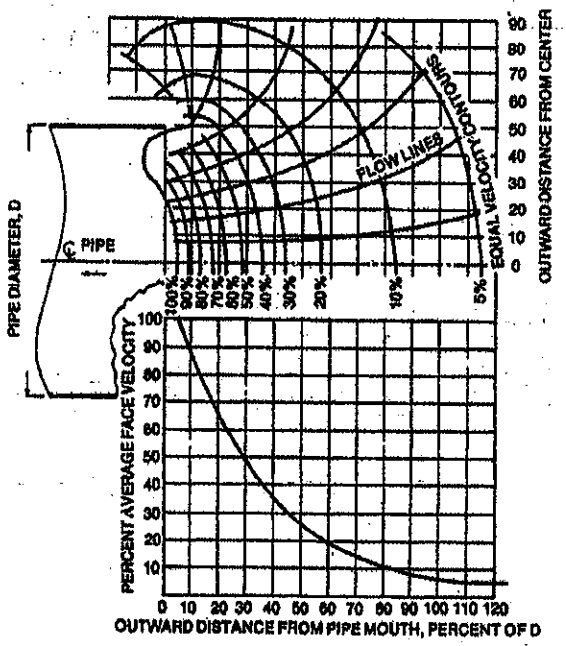
Figure 2 Field Mapping of carpark ventilation system



The two supply outlets No.1 and No.2 were arranged to blow out at the low level, an the two supply outlets No.3 and No.4 were arranged at the high level, similar, for the exhaust outlets.

(This is to comply with the Building Control Regulations that 50% of the carpark exhaust should be at the floor level.)

Please take note that the direction of the supply No.3 and No.4 was designed to be away from the exhaust ducts, and that the direction of the exhaust outlets No.3 and No.4 was similarly designed to be away from the supply ducts. This was done in order to ensure maximum turbulence and mixing of carpark pollutants with the fresh air supply and a lower minimum stay of pollutants in the carpark.



Hood

$$Q = V(10X^2 + A)$$

$Q = Flow\ rate, m^3/s$

$V = Capture\ velocity, m/s$

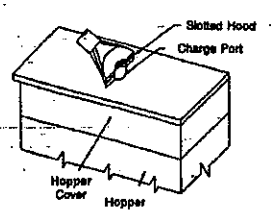
$X = Distance\ from\ hood\ face, m$

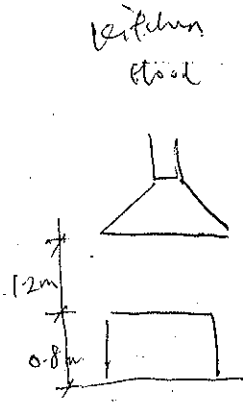
$A = Hood\ face\ area, m^2$

Capture hood example

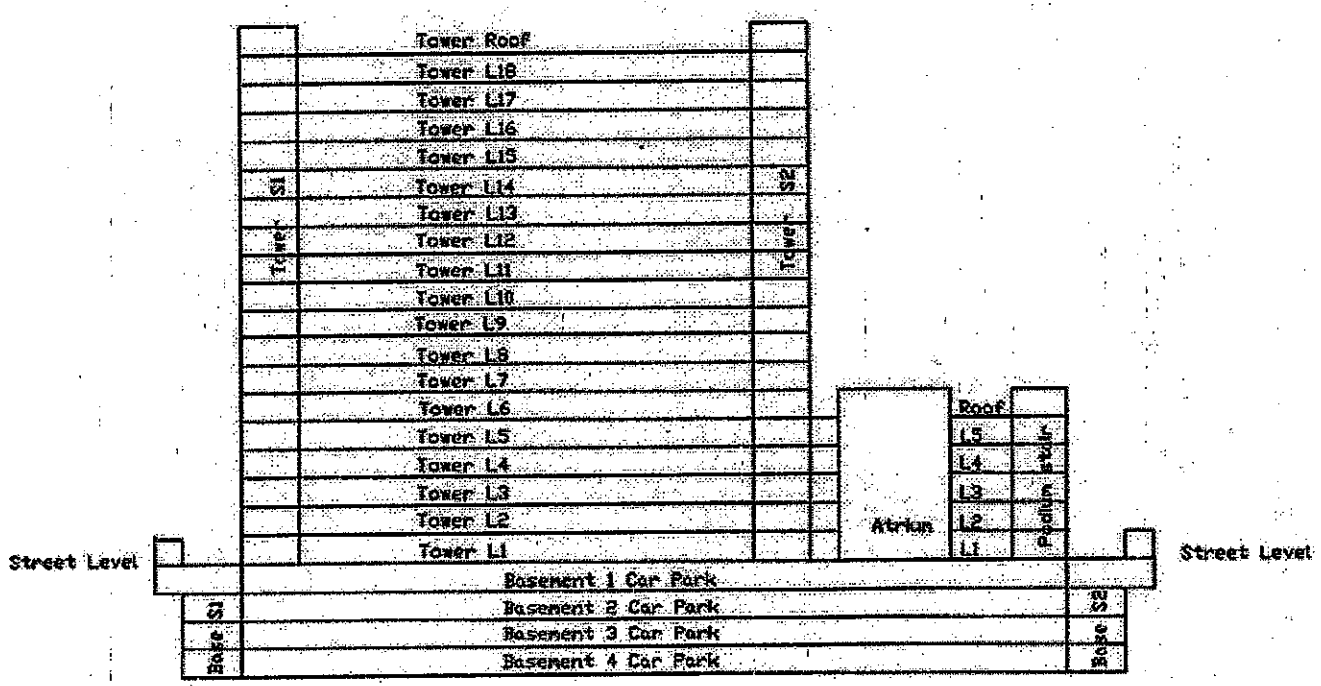
- As an example as shown in figure , if the point of generation is 0.4 m from a 0.13 m by 0.20 m hood, the required volumetric flow is given by :

$$Q = V\,x\left(10X^2 + A\right)$$
$$= 0.75\,x\left[10(0.4)^2 + (0.13x0.20)\right]$$
$$= 1.2\ Actual\ m^3/s$$





Building on Practical Examples of ACMV



Practical Examples on ACMV Based on Practices and Codes
(Introduction and Objective)

All Air-conditioning & Mechanical Ventilation systems & installations must meet the minimum stringent requirements (fire, safety and function ability) in accordance to codes' requirements and practices.

The objective of this short course is to interpret & explain the requirements of the codes & show by examples & illustrations their requirements & practices.

An 18-storey high rise multi-functional building with 4 basements is used as an example to show & demonstrate the various requirements & practices of the codes & how they can be met by calculations & design.

This building consists of :

(a) Hotel bedrooms from L7 to L18 (12 floors).

(b) Offices from L5 to L6 (2 floors).

(c) Shops from L1 to L4 (4 floors).

(d) Car parks from basement 1 to basement 4 (4 floors).

Mechanical Ventilation of Basement Car Park

- **Objective**
  The Objective of Ventilating Car Parks is to remove carbon monoxide & other combustion products exhausted by car engines. These products are harmful to people. CO level to be maintained below 25 ppm averaged over an 8 hour period.

- **Basement Car Park Mechanical Ventilation System**
  As all the 4 basements have less than 2 % ventilation opening, full mechanical ventilation is required. (Note : For 2 % to 20% ventilation opening only exhaust systems are required at 6 ACH. More than 20% opening, then only fume extract systems are required at 1.2 ACH).

- **Design Requirement & Criteria of Full Mechanical Ventilation**
  (1) Minimum 6 ACH.
  (2) Supply & Exhaust Fans & Systems.
  (3) Minimum 2 supply fans & 2 exhaust fans.
  (4) Supply air inlets shall be a minimum of 5.0 m away from any exhaust outlets.
  (5) Car park areas are to be at negative pressure.
  (6) Supply air fans are to be interlocked with corresponding exhaust air fans.
  (7) For exhaust, minimum 50% low level with top of exhaust grille at 650 mm max. from FFL.
  (8) Fans are to be connected additionally to emergency power supply.

Mechanical Ventilation of Basement Car Park
(Smoke Purging)

As the total basement (1,2,3 & 4) car park areas exceeds 1900 sq.m, smoke purging is required.

- **Design Requirements & Criteria on Smoke Purging**

(1) Minimum 9 ACH.
(2) Supply & Exhaust Fans & Systems.
(3) Capacity of each section shall be in the event of breakdown, the remaining section can be able to provide ½ the total air for the storey (minimum 2 supply air fans & 2 exhaust air fans).
(4) Supply air inlets shall be a minimum of 5.0 m away from any exhaust outlets.
(5) Car park areas are to be at negative pressure.
(6) Supply air fans are to be electrically interlocked with corresponding exhaust air fans.
(7) For exhaust, minimum 50% low level with top of exhaust grille at 650 mm max. from FFL.
(8) Fans are to be connected additionally to emergency power supply.
(9) Electric power & control cables of to be fire-rated (or fire-protected).
(10) Ducts outside the basement car-park compartment but inside the building shall be fire-rated.
(11) A remote start-stop panel with status indicating lights to be located inside the FCC. If no FCC, this panel to be located next to where the main fire alarm panel is.

Mechanical Ventilation of Basement Car Park

(Design Calculations)

- (1) 1st Basement Car Park Area (after deducting ram areas, staircase areas, and plant room areas if any, etc.)
  = 852 sq.m
  Height = 3.0 m
  Therefore, Volume = 852 * 3.0
  = 2556 cu.m

(2) As this basement has less than 2% natural ventilation opening, full mechanical ventilation is required.

(3) Using 6 ACH, minimum S.A. required = 2556 * 6 cmh
= 4.26 cms

(4) As minimum 2 S.A. fans are required, therefore, capacity of each fan = 2.13 cms.

- (5) Also, a minimum of 2 'corresponding' exhaust fans is required. To maintain a negative pressure in the basement car parks, a 12 % more E.A. capacity is used than S.A.

Hence, E.A. required = 4.26 * 1.12 cms
= 4.77 cms

& E.A. capacity of each fan
= $\frac{4.77}{2}$ = 2.38 cms

Mechanical Ventilation of Basement Car Park
(Design Layout)

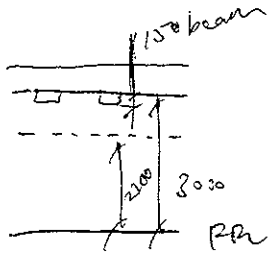
- **Fans & Ducts Layout**
  (1) Common problems encounter to mechanically ventilate (& smoke purging) of basement car parks are :
  (a) Insufficient head room for installation of overhead fans & main ducts.
  (b) To ensure ducts do not cross each other.
  (c) Space not given for installation of floor fans.
  (d) To find sufficient columns or walls to space out the low level exhaust grilles not more than 650 mm above FFL.
  (e) Ensure all areas are ventilated.
  (f) To locate the external S.A. inlet & E.A. outlet.
  (g) Noise.
  (h) Aesthetic.

- **Proposed Solutions**
  (a) Locate fans at corners of floor. S.A. fans & E.A. fans diagonally opposite.
  (b) At fans inlets & outlets to create plenum.
  (c) Install silencers to reduce noise.
  (d) E.A. ducts to run along or near sides of walls. S.A. ducts to run along centers.
  (e) Locate exhaust air grilles & sub-ducts at walls columns. (Low-level grilles and sub-ducts to be protected against damage by vehicles).
  (f) Locate all S.A. grilles at high level.
  (g) Run S.A. ducts along both sides of vehicles' pathways.
  (h) Ensure duct A.R. (say, 4 : 1 or maximum 4.5 : 1) is met.
  (i) Check fan noise and grille velocity.

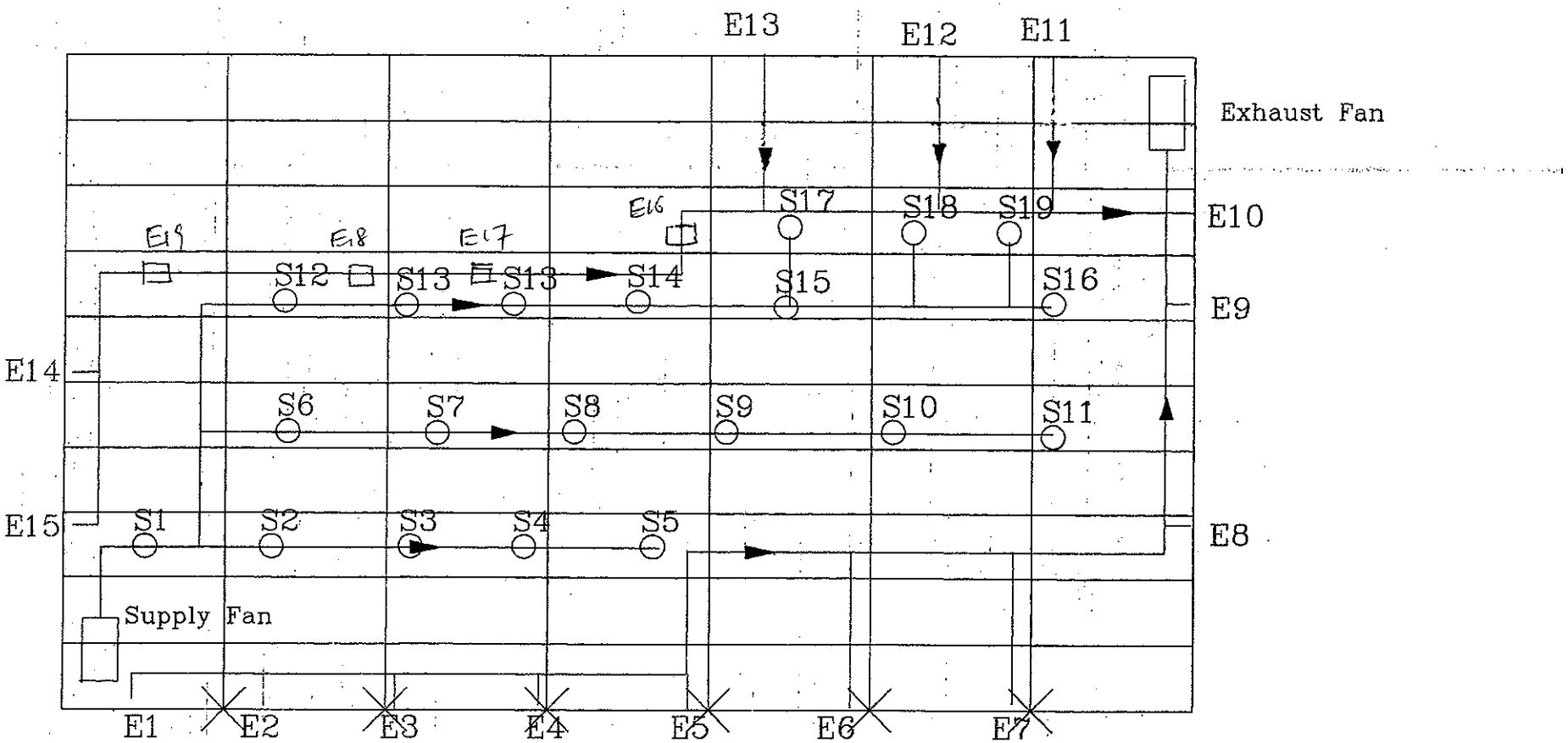
Mechanical Ventilation of Basement Car Park
(Exhaust Fan Selection)

(1) At this stage, to ensure sufficient headroom, draw single-line diagram with estimated overall fan dimension & main duct sizes & air inlets/ outlets.
(2) Estimate generously the total pressure requirement of the exhaust/ supply air systems, taking into account all fitting losses & pressure requirements including silencers, grilles, external louvers, etc.
(a) **Example for the MV Exhaust System** (exhaust air of each fan = 2.38 cms)
(i) Estimated system pressure = 265 Pa.
(ii) From fan manufacturer's performance characteristic curve, a 24 inch (610 mm) axial fan at a speed of 1440 RPM is selected.
From fan curve, at 2.38 cms air flow, fan velocity pressure = 28 Pa.
Hence, Fan Total Pressure
= 265 + 28 = 293 Pa

System operating point of 2.38 cms & 293 Pa total pressure falls within 610mm diameter fan at 1440 RPM.
(iii) Other data of this fan are :
Noise level = 77 dB
Absorbed power = 1.19 kW
Fan efficiency = 76% (select fan with best efficiency)
Pitch angle = 16 degree
Note : Silencers required to attenuate noise level to say, 60 to 65 dB.
(iv) To install the axial fan suspended at ceiling slab, a ceiling space of [610 mm + (2 x 50 mm)] = 710 mm is required. (2 x 50 mm) is required for the flanges & supporting brackets.
FFL to ceiling slab = 3.000 m.
Beam depth = 0.150 m.
Min. head room required in car parks is 2.200 m.
Hence, ceiling space available
= 3.000 – (0.150 + 2.200) = 0.650



Basement carpark , 6ACH



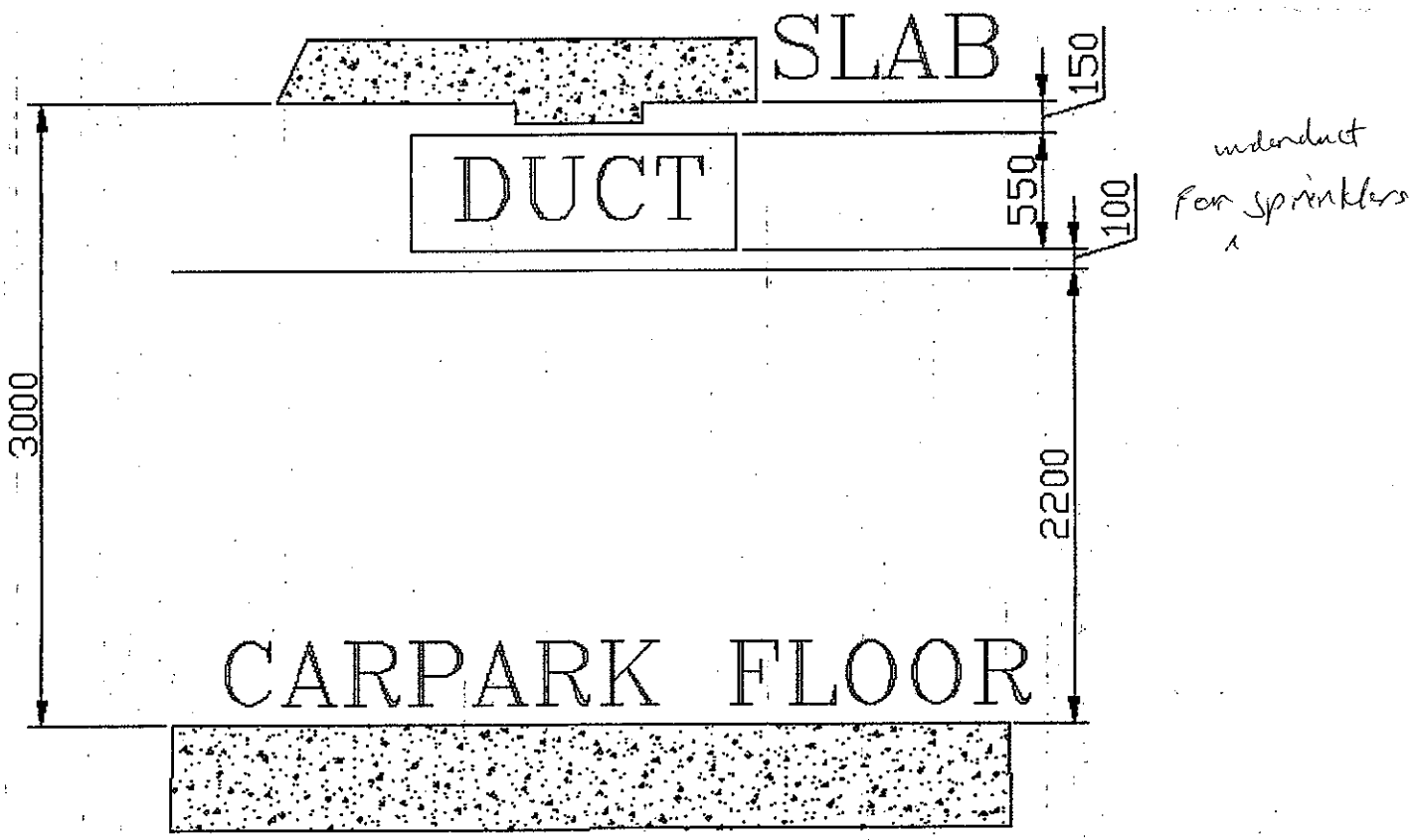
BASEMENT CARPARK
No natural ventilated opening or natural ventilated opening less than 2%, use 6 ACH Supply & Exhaust

E Exhaust Air Grille (19 NO)
S Supply Air Grille (19 No)

Exhaust Grille E1 to E15 are low-level (Not exceeding 650mm)
Exhaust Grille E16 to E19 are high-level exhaust grille.

note: > 50% Exhaust grille at low level

Section of Carpark



Mechanical Ventilation of Basement Car Park
(Duct Space/Exhaust Air Grilles)

- Hence, for fan installation, there is a shortage of 710mm – 650mm = 60mm. If possible, install the fan between beams or at a place without beams.
- (b) **Duct Installation** : Allow 0.100 m for sprinkler installation below ducts; therefore maximum duct depth allowable = 0.650 – 0.100 = 0.550m. From ductulator, with 2 x 2.38 cms = 4.77 cms (which is the largest air flow at the exhaust fan inlets), a duct size of 550mm x 1000mm (or 4500mm x 1200mm) could be selected.

- (c) **Exhaust Air Grilles**
  (i) Total air to be exhausted = 4.77 cms
  (ii) Proposed no. of grilles, say is 19.

Therefore, each grille to exhaust
$= \frac{4.77}{19} = 0.2511$ cms of air.
(Note : More grilles, better air distribution but higher cost)
(iii) From grille manufacturer's technical performance data sheet (model SLG) propose size 0.126 sq.m.
Using 0.500m as the depth of the grille, then width of grille $= \frac{0.126}{0.500} = 0.252$m

Grille static pressure = 12 Pa (this need to be added to duct pressure when sizing fan total pressure).
Noise rating = 22 dB (acceptable).
Hence, exhaust grille size of 250mm x 500 mm acceptable.

Mechanical Ventilation of Basement Car Park
(Grille Size Estimation)

- (iv) **Quick Estimation of Grille Size Area**
  Each grille to exhaust 0.2511 cms of air. Using an air velocity of 3 m/s through the grille, then
  Net Grille Face Area = $\frac{0.2511}{3}$
  = 0.0837 sq.m.
  Using a factor of 1.6 (which means that 60% of gross grille area are occupied by louver blades or deflectors), then
  Gross Grille Area = 1.6 x 0.0837 = 0.134 sq.m.

  (Note : Factor depends on louver angles to the horizontal. The bigger the angle, the bigger the factor).

- (d) **Supply Air System**
  Similarly, the supply air fans, supply air ducts & grilles are sized & selected except that all the supply air grilles are at high level which is permissible by the codes of practice.

Mechanical Ventilation of Basement Car Park
(Smoke Purging)

- **Smoke Purging System**
  The car park area of 1 basement level = 852 sq.m.
  Therefore, total area of 4 basements = 3408 sq.m >1900 sq.m.
  Hence, smoke purging is required to remove the smoke during fire mode (in addition to mechanical ventilation during normal mode).
  Volume of 1 basement = 2556 cu.m.
  For smoke purging, 9 ACH is required.
  Therefore,
  Minimum S.A. required = 2556 * 9 = 23004 cmh = 6.39 cms.
  With 2 S.A. fans, S.A. of each fan = $\frac{6.39}{2}$ = 3.2 cms.
  With an additional of 12% for the exhaust system, then E.A. of each fan = 1.12 * 3.2 = 3.58 cms.

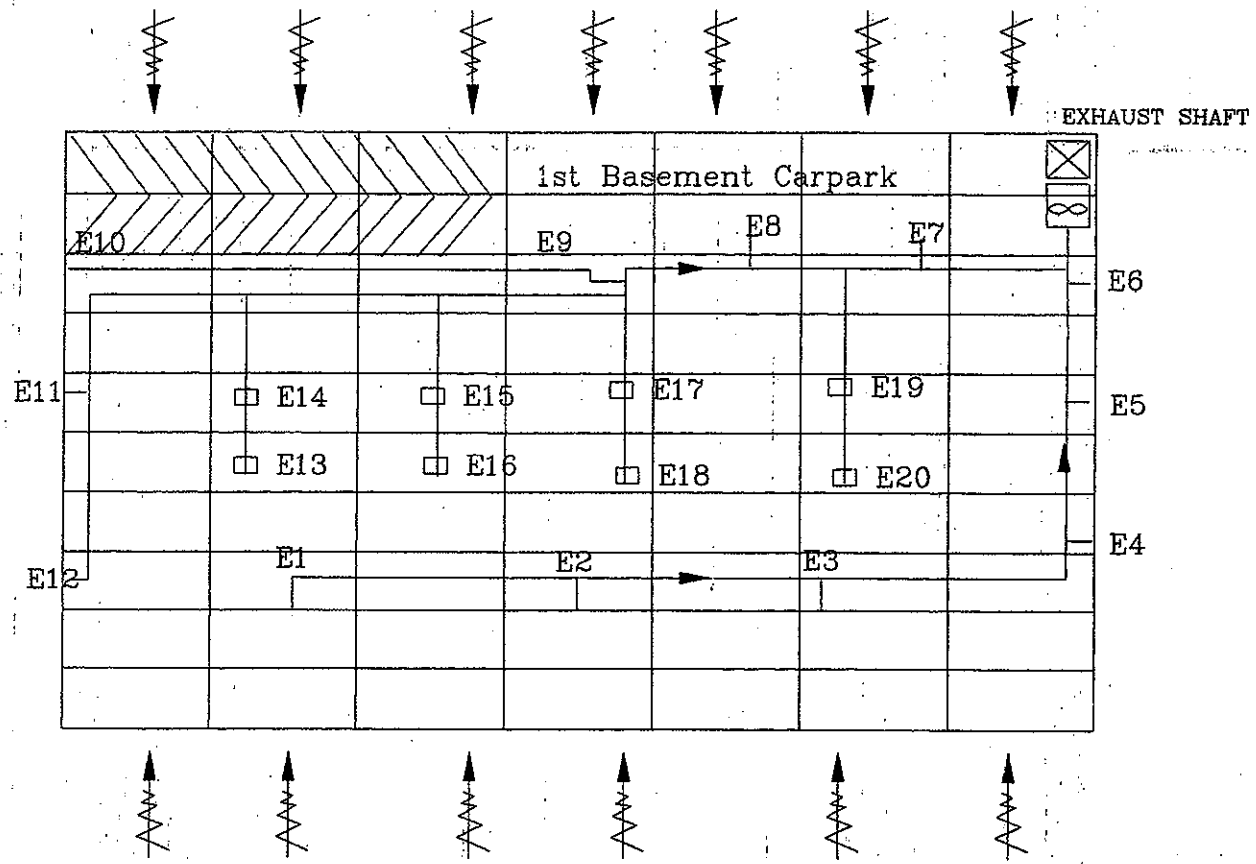
- This also could be obtained as follows :
  For 6 ACH, E.A. fan capacity = 2.38 cms. For 9 ACH, E.A. fan capacity = $\frac{9}{6}$ * 2.38 = 3.57 cms.
  Using Fan Laws,
  $\frac{RPM2}{RPM1} = \frac{3.57}{2.38} = 1.5$ (or 9/6)
  But RPM1 = 1440
  Hence, RPM2 = 1440 * 1.5
  = 2160
  For 6 ACH, fan total pressure = 293 Pa.
  For 9 ACH,
  $\frac{\text{Fan total pressure}}{293} = \frac{Sq(3.57)}{Sq(2.38)} = Sq(1.5)$
  Hence,
  Fan total pressure at 9 ACH = 293 * 659.25 Pa.
  And
  Absorbed Power = 1.19 kW *Cube(1.5)
  = 4.016 kW

Mechanical Ventilation of Basement Car Park

(Air Shaft for the 4 Car Park Basements)

- **Exhaust Air Shaft**
  (i) A common air shaft could be used for all the 4 car park basements.
  (ii) Total amount of air to be exhausted for the 4 basements = 4 * 2.77
  = 19.08 m/s.
  From ductulator, using an air velocity of 12 m/s, size of air shaft = 2.4m x 0.8m, say.
  The air outlet louvres could also be sized similar to the exhaust grilles.
- **Supply Air Shaft**
  Similarly, the supply air shaft is sized.
- **5.0m Distance**
  The external E.A. outlet and the S.A. inlet shall be a min. of 5.0m distance away, measured from the nearest edges.
- **Minimize Air Intermixing**
  S.A. should be designed to minimize intermixing from one car park level to another.
  Similarly for E.A.

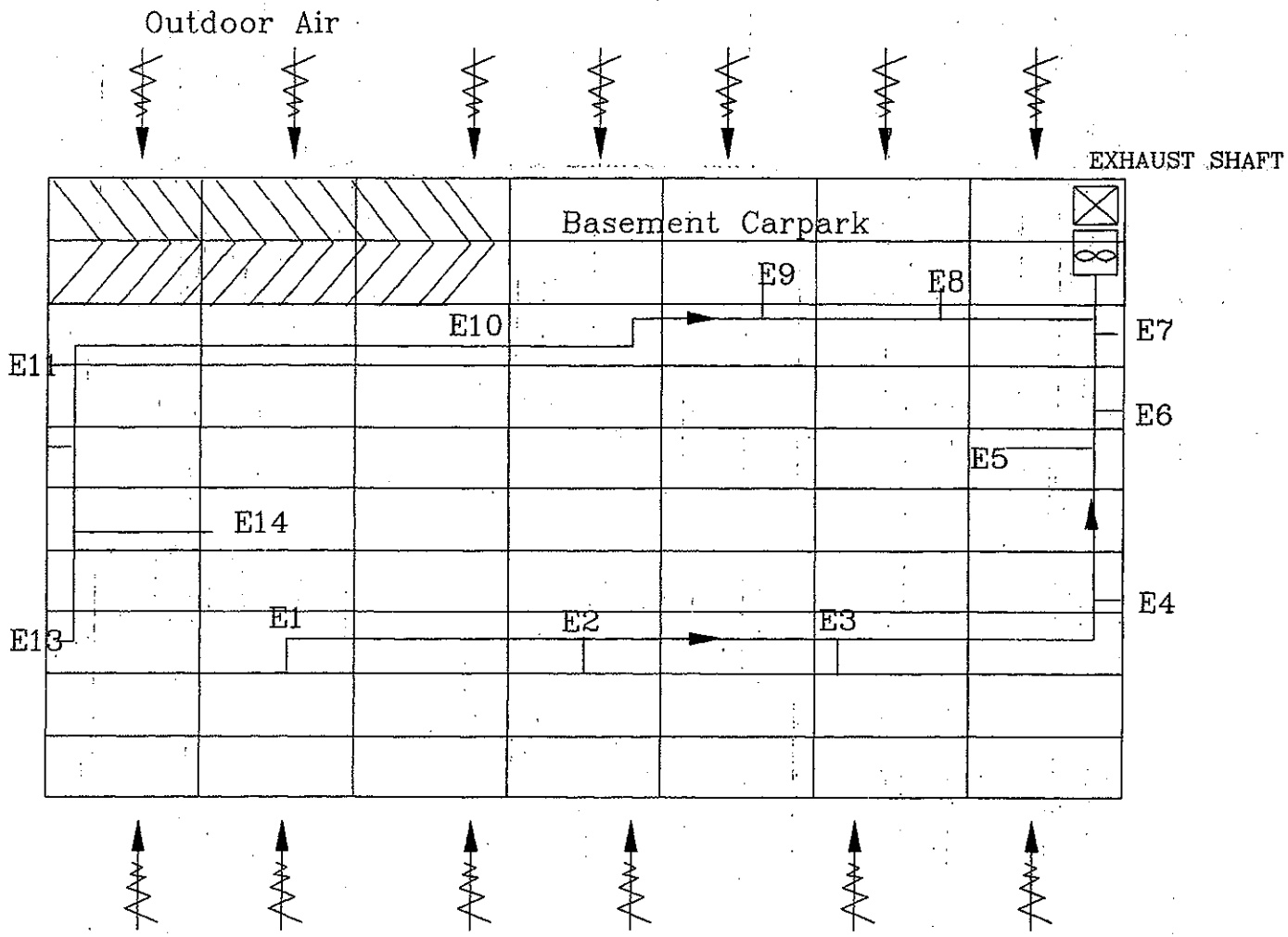
1st Basement Carpark: 2% Opening , 6 ACH Exhaust only (supply omitted)



Natural ventilation opening not less than 2% preferably cross ventilation

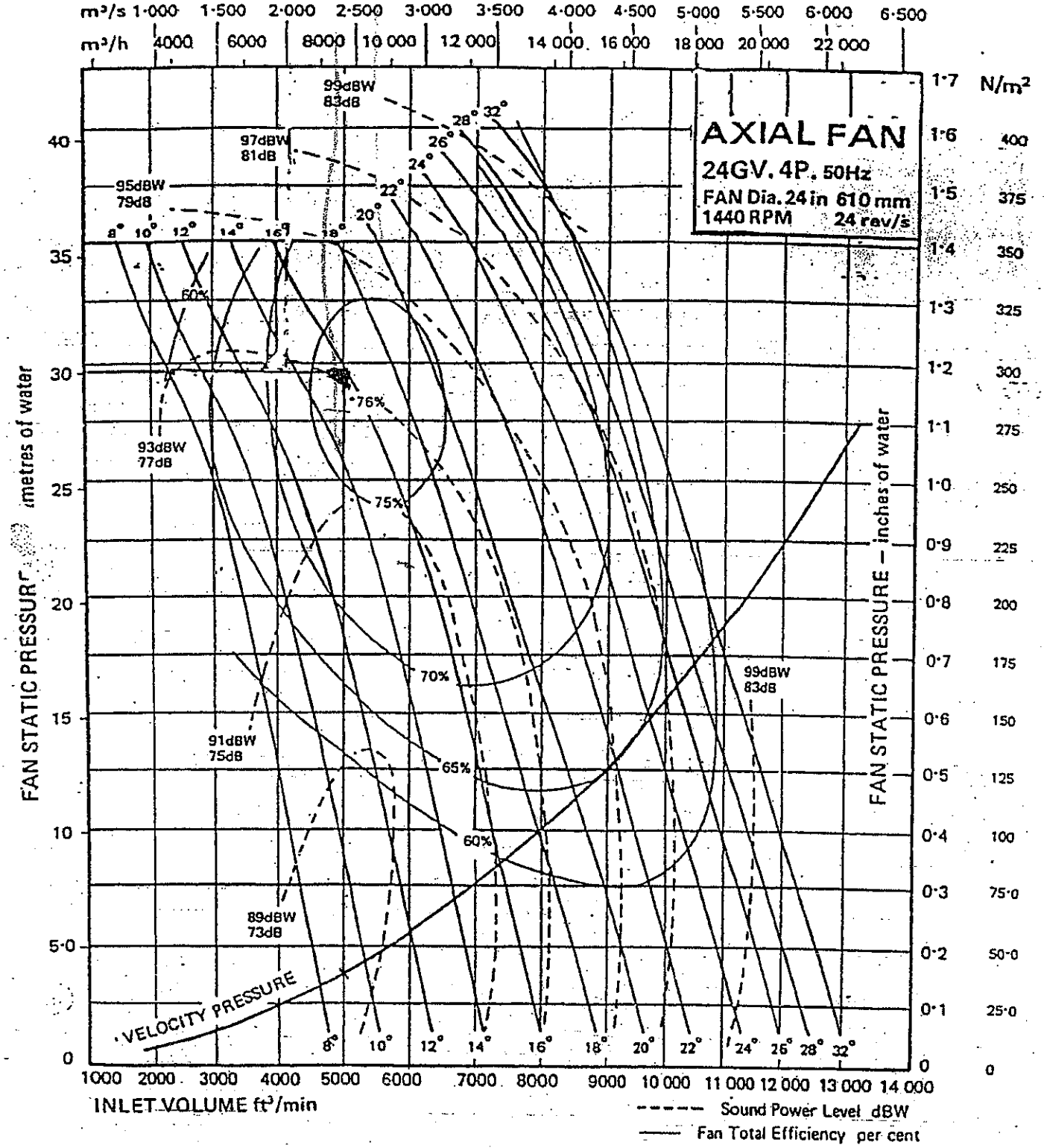
Exhaust Grille E1 to E12 are low level (not exceeding 650 mm)
Exhaust Grille E13 to E20 are high-level exhaust.

20% Opening Criteria, 1.2 ACH Fume Extract Only



All exhaust low level .

Basement Carpark Exhaust Air Fan
(For Natural Ventilation Less than 2%
Full Mechanical Ventilation)



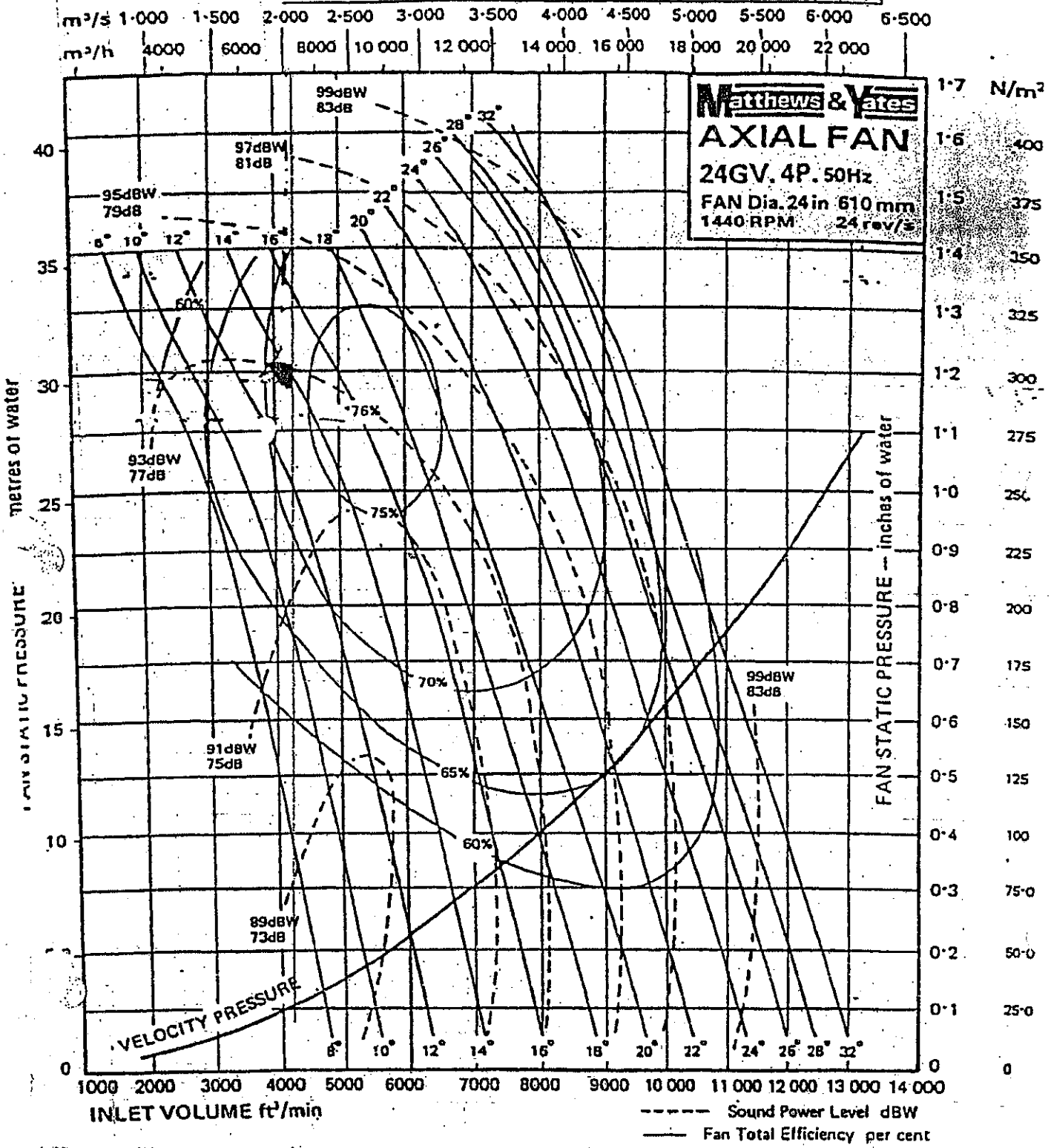
THREE PHASE 415v

ANGLE	8°	10°	12°	14°	16°	18°	20°	22°	24°	26°	28°	30°	32°
PHASE	3	3	3	3	3	3	3	3	3	3	3	3	3
kW Peak absorbed Per Stage	0.48	0.60	0.82	1.01	1.19	1.30	1.49	1.79	2.16	2.46	2.98	3.21	3.47
hp Peak absorbed Per Stage	0.65	0.80	1.10	1.35	1.60	1.75	2.00	2.40	2.90	3.30	4.00	4.30	4.65

ACOUSTIC ANALYSIS

IN DUCT SPECTRUM	OCTAVE BAND MID-FREQUENCY Hz					
	125	250	500	1000	2000	4000
CONSTANTS	13	4	12	13	17	22
To obtain Spectrum subtract above constants from Sound Power level of the fan selected						

BASEMENT CAR PARK SUPPLY AIR FAN



THREE PHASE 415v

ANGLE	8°	10°	12°	14°	16°	18°	20°	22°	24°	26°	28°	30°	32°
PHASE	3	3	3	3	3	3	3	3	3	3	3	3	3
kW Peak absorbed Per Stage	0.48	0.60	0.82	1.01	1.19	1.30	1.49	1.79	2.16	2.46	2.98	3.21	3.47
hp Peak absorbed Per Stage	0.65	0.80	1.10	1.35	1.60	1.75	2.00	2.40	2.90	3.30	4.00	4.30	4.65

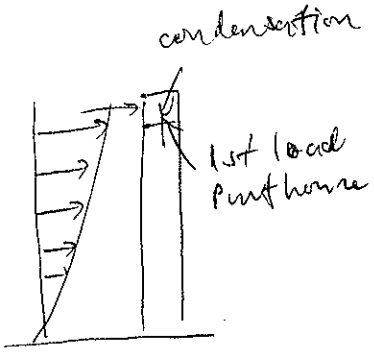
ACOUSTIC ANALYSIS

IN DUCT SPECTRUM	OCTAVE BAND MID-FREQUENCY Hz					
	125	250	500	1000	2000	4000
CONSTANTS	13	4	12	13	17	22
To obtain Spectrum subtract above constants from Sound Power level of the fan selected						

FOR MOTOR DETAILS SEE MOTOR DATA SHEET.

NATURAL VENTILATION LESS THAN 2%. FULL MECHANICAL VENTILATIC

Wind velocity
$= \frac{1}{2} \rho V^2$
$= \frac{1}{2} \times 1.2 \times 10^2$
$= 60 Pa$
$Q = 0.827 A \sqrt{\Delta P}$



Wind velocity = 10 m/s
ASHRAE → Windflow around Bldg.

Mechanical Ventilation for Plant room

17 MAR 2008

Example on Plant room ventilation

137.3 m³

A Transformer room for a factory (8.6 x 5.7 x 2.8 m) .It was designed to house two 2 MVA transformer each with 20 kW full-load losses.

The maximum temperature of the plant room is 42°C, the outside air temperature is 29°C.

$$CMH = \frac{Q}{0.333 \times \Delta t} = \frac{40{,}000}{0.333 \times 13} = 9240 \text{ CMH}$$

ACH = 9240/137.3 = 67.2

or $Q_{(W)} = 1.232 \times \frac{ft^3}{s} \times \delta t$

$Q_{(kW)} = 1.23 \times \frac{m^3}{s} \times \delta t$

Mechanical Ventilation Based on Heat Load (Lift Motor Room : Layout)

- **Mechanical Ventilation of Lift Motor Room.**
  (1) **Hotel Tower Lifts**
  In the Hotel Tower there are 6 passenger lifts located at the hotel reception lobby. These lifts serve 1st storey level to 18th storey level. The loading capacity of these lifts is 11 persons (750 kg) each and the speed is 90 m/min.A separate lift with its own fire-fighting lobbies (and fire-compartmented) is designated as fire lift.
  There is also a separate goods lift serving this block.
  (2) **Basement Lifts of Hotel Tower**
  There are two sets of 2 lifts serving 1st storey level to the 4th basement level of the tower block. One of the lifts is designated as fire lift.
- (3) **Podium (Shopping Center) Lifts**
  In the Podium block there are two sets of 3 passenger lifts and 1 goods lift serving the 1st storey level to the 5th storey level. The lobbies of these passenger lifts are designed as smoke-stop lobbies.One of the 6 passenger lifts is designated as fire lift.
  (4) **Basement Lifts of Hotel Tower**
  There are two sets of 2 lifts serving 1st storey level to the 4th basement level of the tower block. One of the lifts is designated as fire lift.

Mechanical Ventilation Based on Heat Load (Lift Motor Room : Fan Capacity)

- **Hotel Tower Lifts**
  (i) The lift motor room is located at the tower roof.
  (ii) From lift manufacturer's data specification, the heat output of each of the 11 persons (750 kg) lift at 90 m/min. and its accessories in the lift motor room is 1700 kcal/hr.
  Hence,
  Total Heat Output of 6 Lifts
  = 6 * 1700 = 10,200 kcal/hr
  = 10,200 * 1.1621 w = 11.854 kW
  The air flow required to remove this heat is given by the formula
  Q = Heat / (Temp Rise * Air Density)
  where Q = air flow in m^3/s

- Heat = heat gain in kW
  Temp Rise = temperature rise in room against outdoor air temperature in deg.C
  Air Density = 1.2 kg/m^3
  (iii) Using an outdoor air temperature of 33 deg C and the maximum allowable air temperature in the lift motor room of 37 deg C, therefore
  Air Flow, Q = 11.854 / (37-33) * 1.2
  = 2.47 m^3/s
  Two (or more) fans is used instead of one.
  A combination of supply and exhaust air system is used and the supply shall be more than exhaust to positively pressurized the lift motor room.

Mechanical Ventilation Based on Heat Load (Lift Motor Room : Air Change Per Hour)

- **Air Flow Per kW Heat Removed**
  From above calculations, this means that to remove 11.854 kW of heat requires 2.47 m^3/s of air flow in order to maintain a temperature difference of 4 deg C between the outdoor air temperature (33 deg C) & the indoor air temperature (37 deg C). Or for rough estimation $\frac{2.47}{11.854}$ = 0.21 m/\^3/s per kW heat at a temperature difference of 4 deg C.

- **Air Change Per Hour (ACH)**
  Let the volume of the room be V m^3.
  2.47 m^3/s = 2.47 * 3600 = 8892 m^3/hr.
  But, Air Flow = ACH * V
  where air flow = m^3/hr
  Therefore,
  8892 = ACH * V

  or ACH = $\frac{8892}{V}$

  If V, the volume of room to be mechanically ventilated is known, then the air change per hour required can be calculated.

Mechanical Ventilation Based on Heat Load
(Typical Heat Output of Lift)

Load (kg)	Speed (m/min)	Calorific value (kcal/hr)
450	60	700
550	60	850
600	60	900
700	60	1,050
750	60	1,150
900	60	1,350
1,000	60	1,500
1,150	60	1,750
550	90	1,250
600	90	1,350
700	90	1,600
750	90	1,700
900	90	2,050
1,000	90	2,250
1,150	90	2,600
550	105	1,450
600	105	1,600
700	105	1,850
750	105	2,000
900	105	2,350
1,000	105	2,600
1,150	105	3,050

Mechanical ventilation of Lift Motor Room

- In a 14 storey office building, there are six passenger lifts in the lift shaft, and the total heat product in the motor room is 11.85 kW.
- Determine the Outdoor air required to cool the motor room.

Solution

- Using outdoor air temperature = 33°C
- Maximum allowable temperature in lift motor room = 37°C

- Air flow =11.84/(1.2 x (37-33)
- = 2.47 m^3/s

Ventilation of Refrigeration Machinery Rooms

- The dominant reason for ventilation machine rooms is to prevent excessive concentrations of refrigeration, regardless of the refrigerant.
- A further function of the ventilation system is to minimize discomfort to the operators by removing heat from motors and other electrical equipment .
- The flow rate of ventilation air for this purpose is recommended in ANSI/IIAR –2-92 , such that ambient ventilation air brought through the machinery room does not rise in temperature more than 10 ^{0}C.

Ventilation of Refrigeration Machinery Rooms

- For example, compressor motors of 900 kW could be in operation in addition to 20KW of additional energy usage in the machine room, the total kW of heat would be 20 kW + (0.1) (900 kW) , or 110 kW. Only 10% of the compressor motor power is estimated to be released into the machinery room. Based on about 6% due to motor inefficiency and about 4% heat loss to the ambient from the compressor and its auxiliary equipment.
- To limit the temperature rise of the ventilation air to 10° C , the required flow rate Q would be :

$$Q = \frac{110\ kW}{1.2\ X\ 1.0\ X\ 10} = 9.17\ m^3/s$$

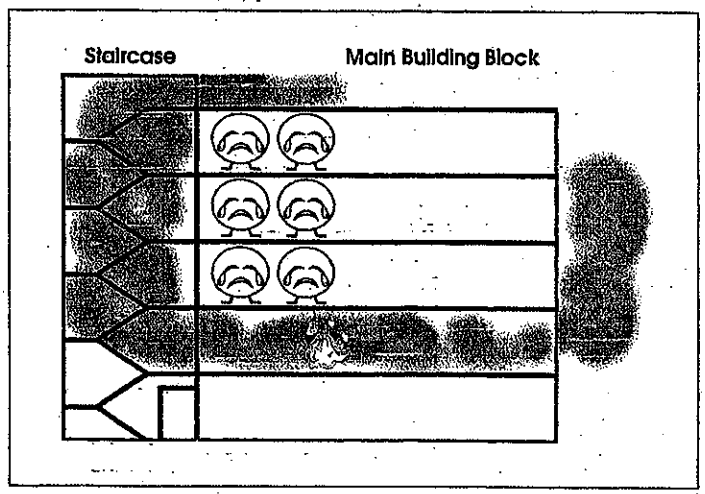
Smoke Control Methods

A fire always produces an enormous quantity of smoke. Smoke contains toxic gases and displaces air. It also reduces the visibility of the escaping building occupants. Indeed, most of the victims in fires are killed by smoke.

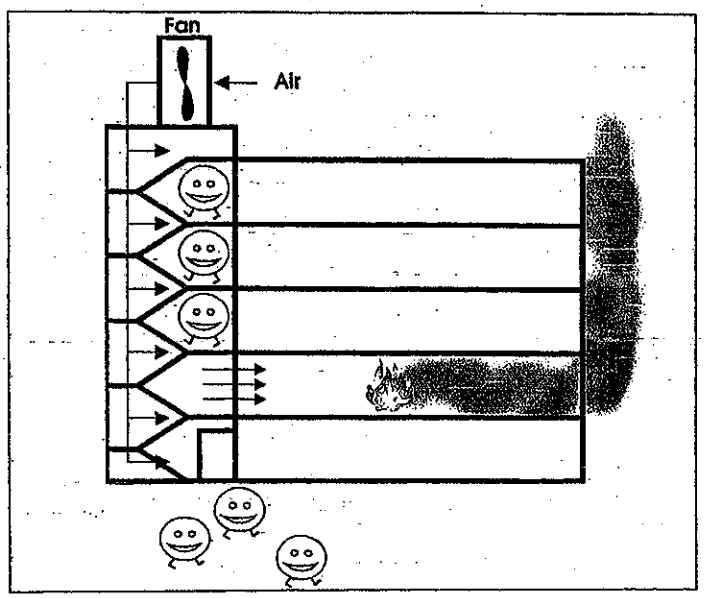
Pressurisation and smoke control systems can be installed to prevent the smoke from spreading into the escape routes and remove it from the building.

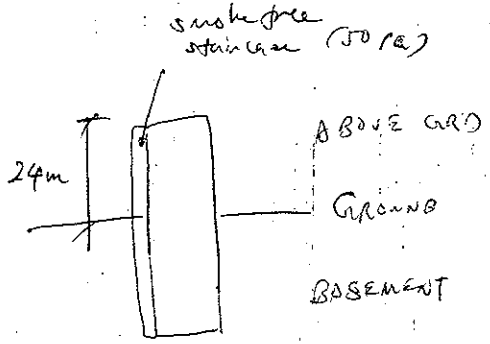
7.2.1 Pressurisation for Exit Staircase

Smoke generated by a fire can spread into the exit staircase, which is designed for building occupants to escape.



If a pressurisation system is installed, however, a fan will supply a large quantity of air into the staircase in the event of a fire. The high pressure air will prevent the smoke from entering the staircase and the occupants will be able to escape from the building.

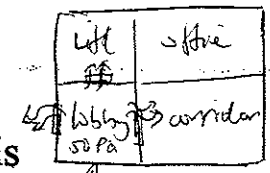




> 24m : need 3 doors opening & 50 Pa

Design
- Performance Based Design
- Code - Based Design

Staircase Pressurization
(Normal & Fire Mode)



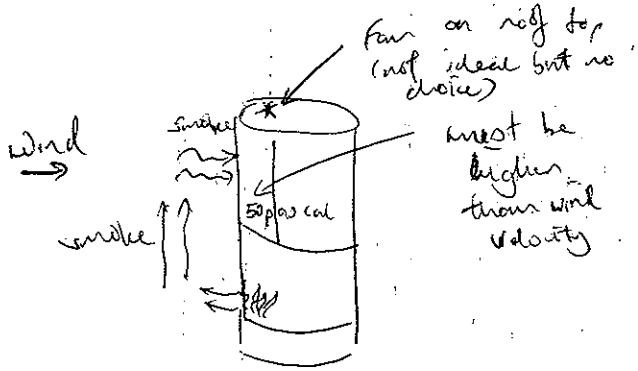
$Q = 0.827 A\sqrt{\Delta P}$

- BS 5588
- Principle of smoke control System (2004)

- Any fire escape staircases if not naturally ventilated require mechanical ventilation during normal mode and pressurization during fire mode.This is common for internal staircases.
- **Code Requirements** :
  (1) Fan to be connected to emergency power supply (power & control cables to be fire-rated or fire-protected when running outside staircase enclosure).
  (2) Start-stop control & visual indication of operation shall be provided in the FCC (if no FCC then at MFIB).
- **Habitable Height Less Than 24m**
  Min. 4 ACH for mechanical ventilation and min. 10 ACH for pressurization (supply mode only).
  Podium Stair
  The podium stair serves 5 floor levels.

Using 3.5 m per level, therefore, this staircase serve the highest floor of habitable height 3.5 * 5 = 17.5 m.

Hence, for podium stair, 4 ACH for MV and 10 ACH with supply mode only for pressurization is applicable.
(i) Area of Staircase = 3m * 4.5m
= 13.5 sq.m
(ii) Volume of Staircase = 13.5 * height
= 13.5 * (3.5 * 6) = 283.5 cu.m
(iii) **Normal Mode** (4 ACH)
Hence, Air Volume Flow Rate =
283.5 * 4 = 1134 cu.m/hr
(iv) **Fire Mode** (10 ACH)
Air Volume Flow Rate = 283.5 * 10
= 2835 cu.m/hr
In practice, a 2 speed fan is used (instead of using 2 fans). However, the fan must be connected to emergency supply.



Door area x velocity
= ± 2m² x 1m/s
= 2m³/s x 2 doors
≥ 4 m³/s

$Q = 0.827 \times A\sqrt{\Delta P}$

$4 = 0.827 \times A\sqrt{\Delta 60}$

Staircase Pressurization (Code Requirements)

If the fan & air ducts is in the staircase enclosure or outside the building, the fan & ducts need not be fire-protected or fire-rated.
Similarly, for the power & control cables.

- **Habitable Height 24m & More (Also, 4 Basement Levels & More).**
Min. 4 ACH for mechanical ventilation & "Full Pressurization" required in accordance to code requirements.
Code Requirements & Criteria
(1) When all staircase doors closed, a min. of 50Pa pressure difference is required between staircase & occupied areas.
(2) Automatic door closures are used to close the doors.
(3) All doors are to open towards the staircase enclosure except the door at street level which is to open towards the street level.
(4) Force required to open any staircase door shall not exceed 110N at the door handle when pressurization system & automatic closing device are in operation.
(5) Door air-flow velocity to be not less than 1.0m/s when main discharge door & any 2 doors of successive stories are fully open.

Staircase Pressurization (Tower Stair)
(Air Flow Estimation)

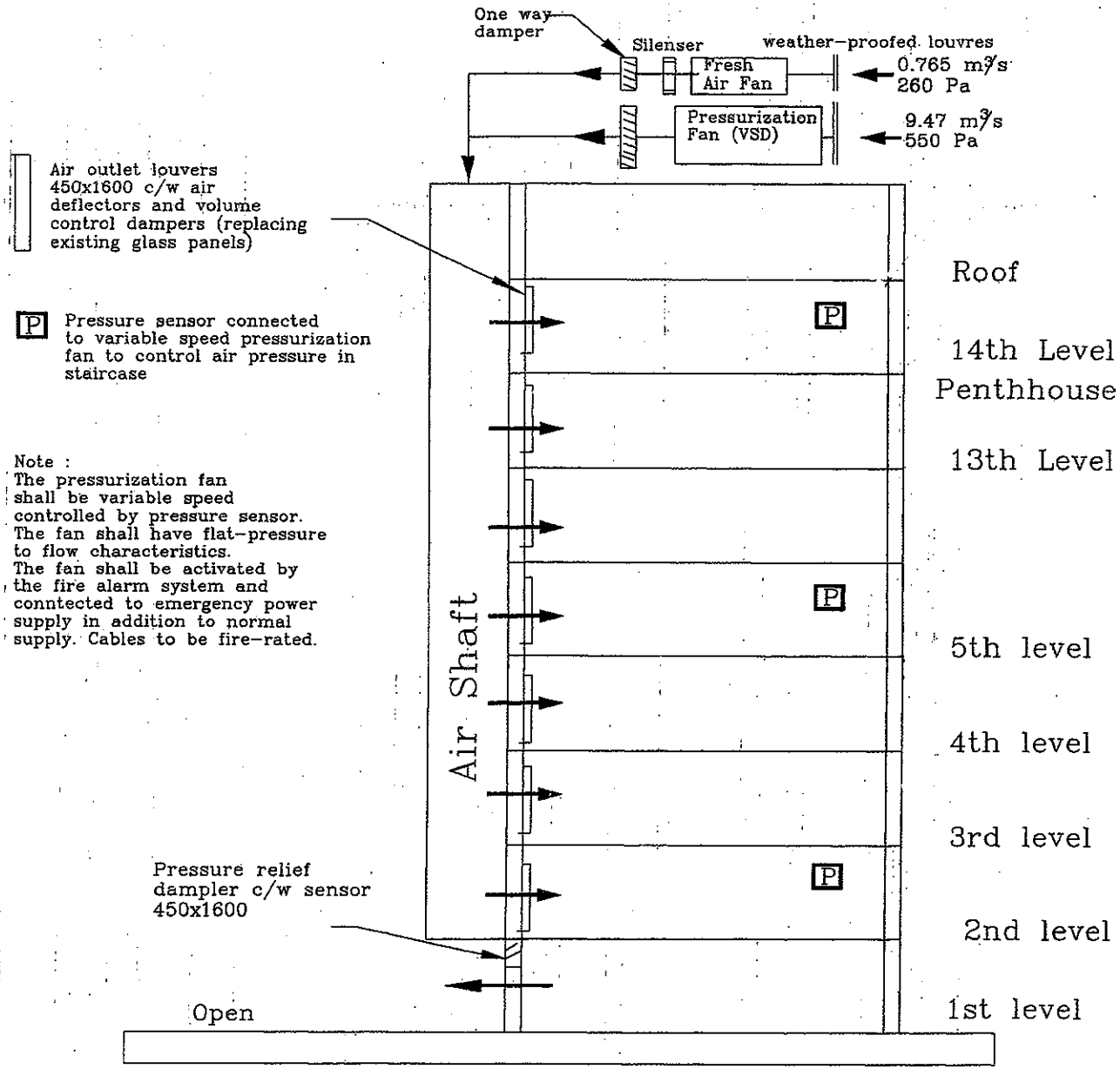
(6) Pressure of exit staircase shall be higher than adjacent smoke stop/fire fighting lobby, if any, or any other adjacent areas (eg. occupied areas, exit passage-ways, or corridors).
(7) A pressure differential of not less than 50 Pa is to be restored 'as soon as practicable', preferably 15 sec. or less but not more than 60 sec., if any pressure fluctuations arise due to opening of doors.
(8) Fan sizing shall include air leakages through door gaps & under-cuts.
(9) To supply large air-flow when doors are opened and yet without causing high pressure fluctuations when doors are closed, fan with flat pressure-to-flow characteristics curve should be selected.

- **Calculations of Staircase (Full) Pressurization System**

(A) **Tower Staircase S1 (>24m height).**

(1) Estimation of Air Flow Rate
(a) Air volume through doors :
(i) For the main discharge door, door width = 1.1m, door height = 2.2m.
Therefore, door area = 1.1 * 2.2
= 2.42 sq. m.
(ii) For the 1st other door (use maximum size), door width = 1.1 m, door height = 2.2m.
Therefore, door area =1.1 * 2.2
= 2.42 sq.m.
(iii) For the 2nd other door (use maximum size), door width = 1.1 m, door height = 2.2m.
Therefore, door area =1.1 * 2.2
= 2.42 sq.m.

Staircase Pressurization System



Staircase Pressurization
(Effective Door Leakage Areas)

BS 5588

Type of Door	Door Size	Door Perimeter (m)	Crack Length (m)	Leakage Area (sq.m)
Single Leaf Opening into Pressurized Space	2m x 0.8m	2 * (2 + 0.8)	5.6	0.01
Single Leaf Opening Outwards	2m x 0.8m	2 * (2 + 0.8)	5.6	0.02
Double Leaf with or without Central Rebate	2m x 1.6m	2 * (2 + 1.6) +2	9.2	0.03
Lift Door	2m High x 2m Wide	2 * (2 + 2)	8.0	0.06

Staircase Pressurization (Tower Stair)
(Air Leakage Thru' Door Gaps)

(iv) Total Door Areas = 2.42 + 2.42 +2.42 = 7.26 sq.m.
But the air flow velocity from the stair shaft outward through the doors must not be less than 1 m/s.
Therefore, through the 3 fully open doors, Air Flow = Total Door Areas * air velocity = 7.26 * 1 = 7.26 cms.

- (B) **Air Leakage through Door Gaps of Closed Doors**

(i) There are a total of (18 + 1 + 1) = 20 doors in Tower Stair S1 (18 levels plus discharge door at L1 & tower roof door). When 3 doors fully open, 17 doors fully closed. Out of these 17 doors, 1 open 'outwards' at the roof & 16 open into 'pressurized space' (pressurized smoke stop lobby, fire fighting lobby or escape corridor. These are single-leaf doors of size 1.1m x 2.2m.

(ii) For the 1 Door that Open 'Outwards'
Door Perimeter (Crack Length)
= 2 * (1.1 + 2.2) = 6.6m.
But Leakage Area per Crack Length = 0.02 sq.m per 5.6m.
Hence, Equivalent Door Leakage Area for this door = $\frac{6.6}{5.6} * 0.02$
= 0.02357 sq.m.

(iii) For the 16 Doors that Open into 'Pressurized Space'
Door Perimeter (Crack Length)
= 2 * (1.1 + 2.2) * 16 = 6.6 * 16 m
= 105.6 m.
But Leakage Area per Crack Length = 0.01 sq.m per 5.6m.
Hence, Equivalent Door Leakage Area for these 16 doorws = $\frac{105.6}{5.6} * 0.01$
= 0.1886 sq.m.

proportionate 5-6 crack length, 2 door leakage area

Staircase Pressurization

(Hobson Stewart Formula/Force Required to Open Door)

(iii) Therefore, Total Equivalent Effective Door Leakage Areas
= 0.02357 + 0.1886 = 0.2121 sq.m.
Using Hobson & Stewart Formula,
Total Air Leakage = 0.827 * Equivalent Air Leakage Area in sq.m * (Pressure Differential in Pa Across Door)^0.5
= 0.827 * 0.2121 * (50)^0.5
= 1.24 cms. where 0.827 is a constant for the formula and a minimum pressure differential of 50 Pa across door is used as per Code.
Therefore, Air Leakages = 1.24 cms.

$Q = 0.827 A\sqrt{\Delta P}$

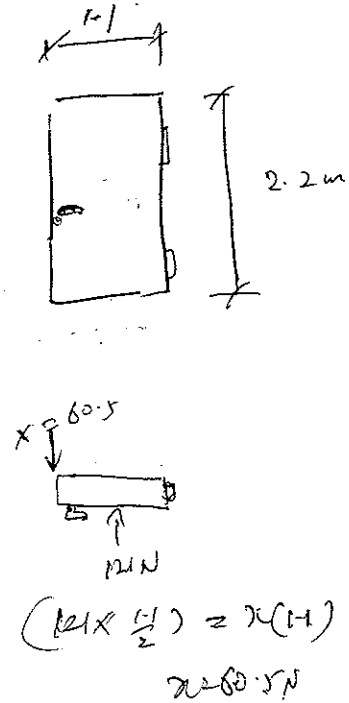
- Hence to pressurize Staircase S1, minimum Air Flow required = 7.26 + 1.24 = 8.5 cms.
Add 10% contingency,
Hence,
Air Flow = 1.1 * 8.5 = 9.35 cms.

- **(C) Force Required to Open Door During Pressurization**

(i) Other than main discharge door, door with maximum width chosen.
Door Size 1.1m x 2.2m.
Area of door = 1.1 * 2.2 = 2.42 sq.m.
But pressure differential across door = 50Pa. Therefore, Force exerted at center of door = 2.42 * 50 = 121N $(P = \frac{F}{A})$
Taking Moment at door hinges,
Therefore, Force at Door Handle

$$= \frac{(0.5 * 1.1 * 121)}{1.1} = 60.5N$$

(ii) The force of 60.5N at door handle is acceptable as this force is less than 110N permissible by Code.



Staircase Mechanical Ventilation
(Tower Staircase S1)

- In addition, the staircase need to be mechanically ventilated at 4 ACH during normal mode.
  (i) Area of staircase = 3m * 4.5m = 13.5 sq.m.
  Hence, Volume of Staircase = [13.5 * 3.5 * (18 + 1)] = 897.75 cu.m where 3.5m is the floor to floor height.
  At 4 ACH, Air Flow Required = 897.75 * 4 = 3591 CMH = 0.9975 cms.
  Add 10% contingency, therefore, Air Flow = 1.1 * 0.9975 = 1.097 cms.
  (ii) As the amount of air flow required for mechanical ventilation is only 11.73% of that for pressurization, a separate fan is used for mechanical ventilation during normal mode.

$\frac{1.097 \text{ cms}}{9.35 \text{ cms}} \times 100\%$

- However, both the fans require a non-return damper each to prevent air escaping from one fan to another when on.

Staircase Pressurization (Basement Stair)
(Air Flow Estimation)

- (A) **Basement Staircase S1 (4 Basements).**
  (1) Estimation of Air Flow Rate
  (a) Air volume through doors :
  (i) For the main discharge door, door width = 1.1m, door height = 2.2m.
  Therefore, door area = 1.1 * 2.2
  = 2.42 sq. m.
  (ii) For the 1st other door (use maximum size), door width = 0.8 m, door height = 2.0m.
  Therefore, door area = 0.8 * 2.0
  = 1.6 sq.m.
  (iii) For the 2nd other door (use maximum size), door width = 0.8 m, door height = 2.0 m.
  Therefore, door area = 0.8 * 2.0
  = 1.6 sq.m.
  (iv) Total Door Areas = 2.42 + 1.6 +1.6 = 5.62 sq.m.
  But the air flow velocity from the stair shaft outward through the doors must not be less than 1 m/s.
  Therefore, through the 3 fully open doors, Air Flow = Total Door Areas * air velocity = 5.62 * 1 = 5.62 cms.
- (B) **Air Leakage through Door Gaps of Closed Doors**
  (i) There are a total of (4 + 1) = 5 doors in Basement Stair S1 (4 levels plus 1 discharge door at L1). When 1discharge door and 2 other doors fully open, 2 doors fully closed. These 2 doors open 'outwards' to the car park basements which are not pressurized. These are single-leaf doors of size 0.8 m x 2.0 m.

Staircase Pressurization (Basement Stair)

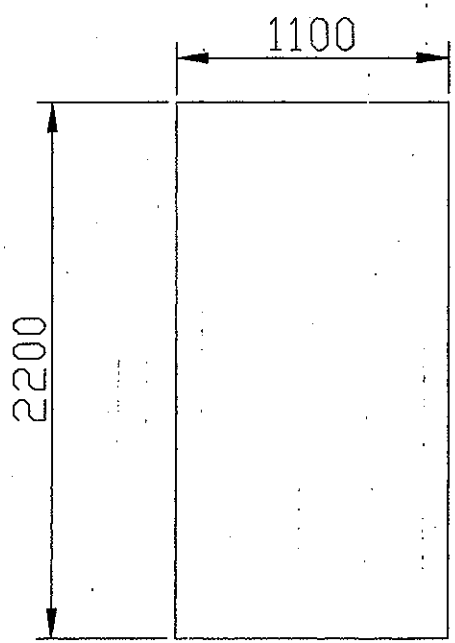
(Air Leakage Thru' Door Gaps)

- (ii) For the 2 closed doors that Open 'Outwards'
  Door Perimeter (Crack Length) of 1 door = 2 * (0.8 + 2.0) = 5.6 m.
  But Leakage Area per Crack Length = 0.02 sq.m per 5.6 m.
  Hence, Equivalent Door Leakage Area for this door = $\frac{5.6}{5.6} * 0.02$
  = 2 * 0.02 sq.m = 0.04 sq.m.
  (iii) Using Hobson & Stewart Formula,
  Total Air Leakage = 0.827 * Equivalent Air Leakage Area in sq.m * (Pressure Differential in Pa Across Door)^0.5
  = 0.827 * 0.04 * (60)^0.5
  = 0.256 cms. where 0.827 is a constant for the formula and a pressure differential of 60 Pa (minimum 50 Pa) across door is used as per Code.
  Therefore, Air Leakages = 0.256 cms.

(iv) Hence to pressurize Staircase S1, minimum Air Flow required = 5.62 + 0.256 = 5.876 cms.
Add 10% contingency,
Hence,
Air Flow = 1.1 * 5.876 = 6.46 cms.

- (C) **Force Required to Open Door During Pressurization**
  (i) Other than main discharge door, door with maximum width chosen.
  Door Size 0.8m x 2.0m.
  Area of door = 0.8 * 2.0 = 1.6 sq.m.
  But pressure differential across door = 60Pa. Therefore, Force exerted at center of door = 1.6 * 60 = 96N
  Taking Moment at door hinges,
  Therefore, Force at Door Handle
  = $\frac{(0.5 * 0.8 * 96)}{0.8}$ = 48N
  (ii) The force of 48N at door handle is acceptable as this force is less than 110N permissible by Code.

Door



$0.0128 m^2/m$ of door crack length

Staircase Pressurization

(Test Results of a Typical Building Using the above Method of Design and Calculations)

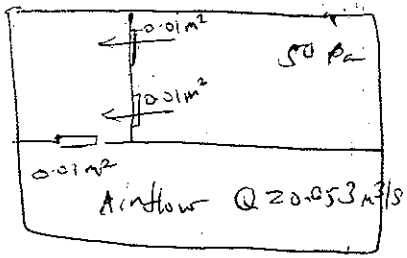
Approved Plan No : BD7/3/8348/96	Description of Project : Staircase Pressurization Test Report For : PROPOSED 11-STOREY OFFICE BUILDING WITH A BASEMENT

Test No.	Storey No.	Air velocity (m/s) with Designated Doors Opened		Pressure (Pa) with all Doors Closed		Force (N) Required to Open Doors	
		Staircase 1	Staircase 2	Staircase 1	Staircase 2	Staircase 1	Staircase 2
1.	1st	1.02	1.15	50	50	105	104
	11	1.15	1.20	50	50	105	110
	10	1.05	1.11	50	50	106	108
2.	1st	1.17	1.11	50	50	107	104
	9	1.13	1.09	50	50	107	110
	8	1.05	1.11	50	50	106	110
3.	1st	1.11	1.07	50	50	102	104
	7	1.19	1.14	50	50	105	107
	8	1.08	1.19	50	50	105	110
4.	1st	1.11	1.09	50	50	102	105
	5	1.19	1.13	50	50	105	110
	4	1.09	1.20	50	50	106	110

Date Tested : 8 March 1996

Correspondence Address and Telephone No.

Stamp and Signature of Professional Engineer



Parallel : $A_e = A_1 + A_2$

Series : $A_e = \frac{A_1 * A_2}{\sqrt{A_1^2 + A_2^2}}$

$Q = 0.827 A_e \sqrt{\Delta P}$

$A_e = \frac{(0.01 + 0.01) \times 0.01}{\sqrt{0.02^2 + 0.01^2}}$

$= 0.009 m^2$

$Q = 0.827 \times 0.009 \sqrt{50}$

$= 0.053 m^3/s$

Staircase Pressurization
(Comparison of Codes)

COMPARISON OF VARIOUS CODES OF PRACTICE

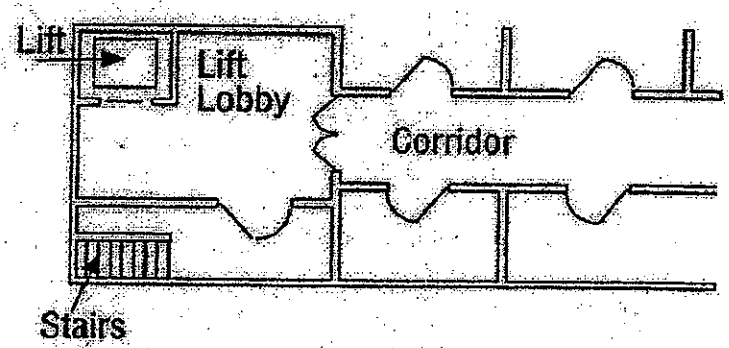
Country	Code	Pressure (Pa) Min.	Max.	Door Velocity	No. Of Effective Open Doors
U.K.	BS5588 Part 4	50	60	0.75 m/sec	ONE (2 Doors on Fire Floor)
	BS5588 Part 5	Not Relevant		2.00 m/sec	THREE (2 Doors on Fire Floor) (Exit and Lift Door)
Australia	AS1668 P.1	50	110	1.00 m/sec	THREE (2 Doors on 2 Floors) (Exit Door)
Singapore	CP13	50	110	1.00 m/sec	THREE (2 Doors on 2 Floors) (Exit Door)
Canada	N.B.C.C. 1990	No Mention		4.72 m³/sec + 0.094 m³/sec For every door	FOUR (2 Doors on 3 Floors) (Exit Door)
U.S.A.	U.S.C. 1988	37		No Mention	No Mention
	N.F.P.A. (92A) 1988	up to 45	133	No Mention	No Mention

Smoke Stop/Fire Fighting Lobby Pressurization
(Code Requirements)

- **Smoke Stop/Fire Fighting Lobby**
  (A) Code Requirements
  (1) Supply air only.
  (2) Normal mode : Mechanical Ventilation at 4ACH.
  (3) Fire mode : Minimum 10 ACH. (2-speed fan normally used.)
  (4) Supply air to be conveyed via vertical shaft or horizontal duct, with air outlets at every lobby.
  (5) Duct outside lobby but inside building to be fire-rated.
  (6) Fire damper to be fitted at air outlet when system is shared with more than 1 lobby.
  (7) Connect to normal & emergency power supply.
  (8) Connect to fire alarm system.
  (9) Electrical (& control) cables to be fire-rated.
  (10) Remote manual start/stop control & indication panel required.
  (11) Independent from other systems.

Smoke Stop/Fire Fighting Lobby Pressurization
(Design & Calculations)

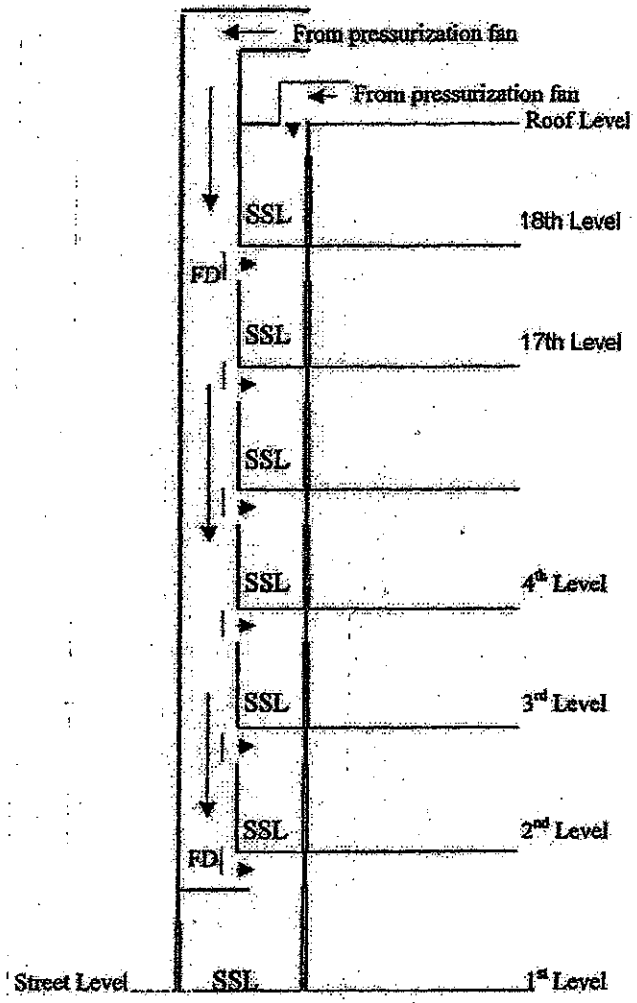
- For the simulated building, just next to the tower staircases are the smoke stop/fire fighting lobbies.
  (i) Using a lobby of area 12 sq.m & a height of 3 m, the volume of each lobby = 12 * 3 = 36 cu.m.
  (ii) For Normal Mode, **MV** with **4 ACH**, S.A. required for each lobby = 36 * 4 = 144 cmh = 0.04 cms.
  Hence, for 18 lobbies, S.A. required = 0.04 * 18 = 0.72 cms.
  (iii) For Fire Mode, 10 ACH required.
  Therefore, S.A. required for each lobby = 36 * 10 = 360 cmh = 0.1 cms.
  Hence, for 18 lobbies, S.A. required = 0.1 * 18 = 1.8 cms.



Stairwell With Lift Lobby & Corridor

Smoke Stop/Fire Fighting Lobby Pressurization
(Design Layout)

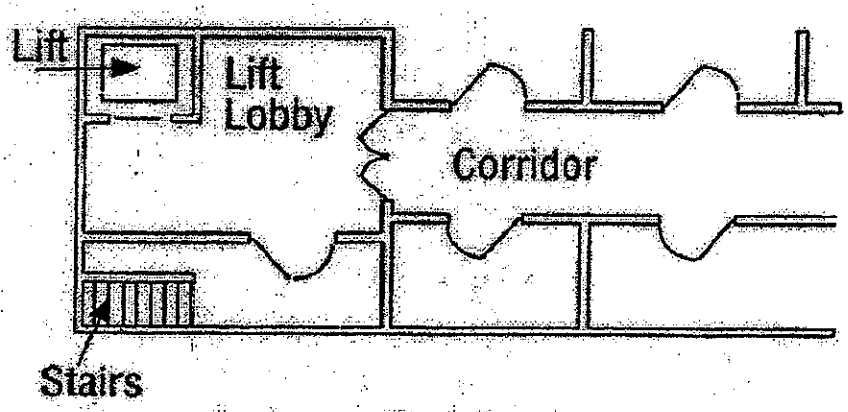
Illustration (smoke-stop and fire-fighting lobbies) :



Smoke-stop and fire-fighting lobbies : Fire dampers (FD) required when system is shared with more than one compartment.

Pressure Gradient : Staircase-Smoke Stop Lobby-Corridor

- **Code Requirements**
  For hotels, during Fire Mode, the pressure at the staircase shall be higher than that at the smoke stop/fire fighting lobby. The pressure at the smoke stop/fire fighting lobby shall be higher than that of the corridor (and the pressure at the corridor shall be higher than the rooms)..
  If the corridor is mechanically ventilated, 4 ACH is required during normal mode and 10 ACH (supply mode only) is used for fire mode. If the corridor is air-conditioned then all requirements during fire mode have to be complied.

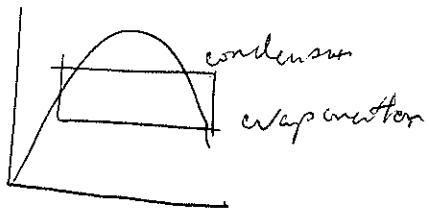


Stairwell With Lift Lobby & Corridor

chiller performance

$Q = 1.23 \times \frac{m^3}{s} \times \delta t$

Power = Pressure × Q

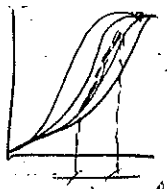


The central air-conditioning system consists of two parts: the air side and the water-side.

A typical air-side arrangement is illustrated in Fig. 6.3.a below.

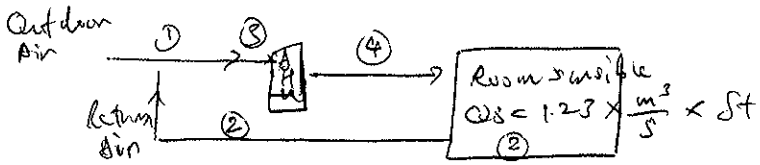
* Warm air is extracted from the room through return air grilles.
* Part of the warm air is exhausted to the outside atmosphere. Outside air is added and mixed with the remaining air.
* The mixed air is filtered and cooled in an air handling unit (AHU).
* The cooled air is delivered back to the room by a fan through diffusers.

damper curve

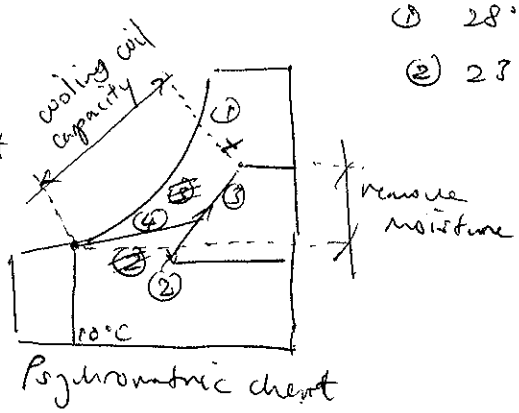


size of this region, proportional curve easy to control

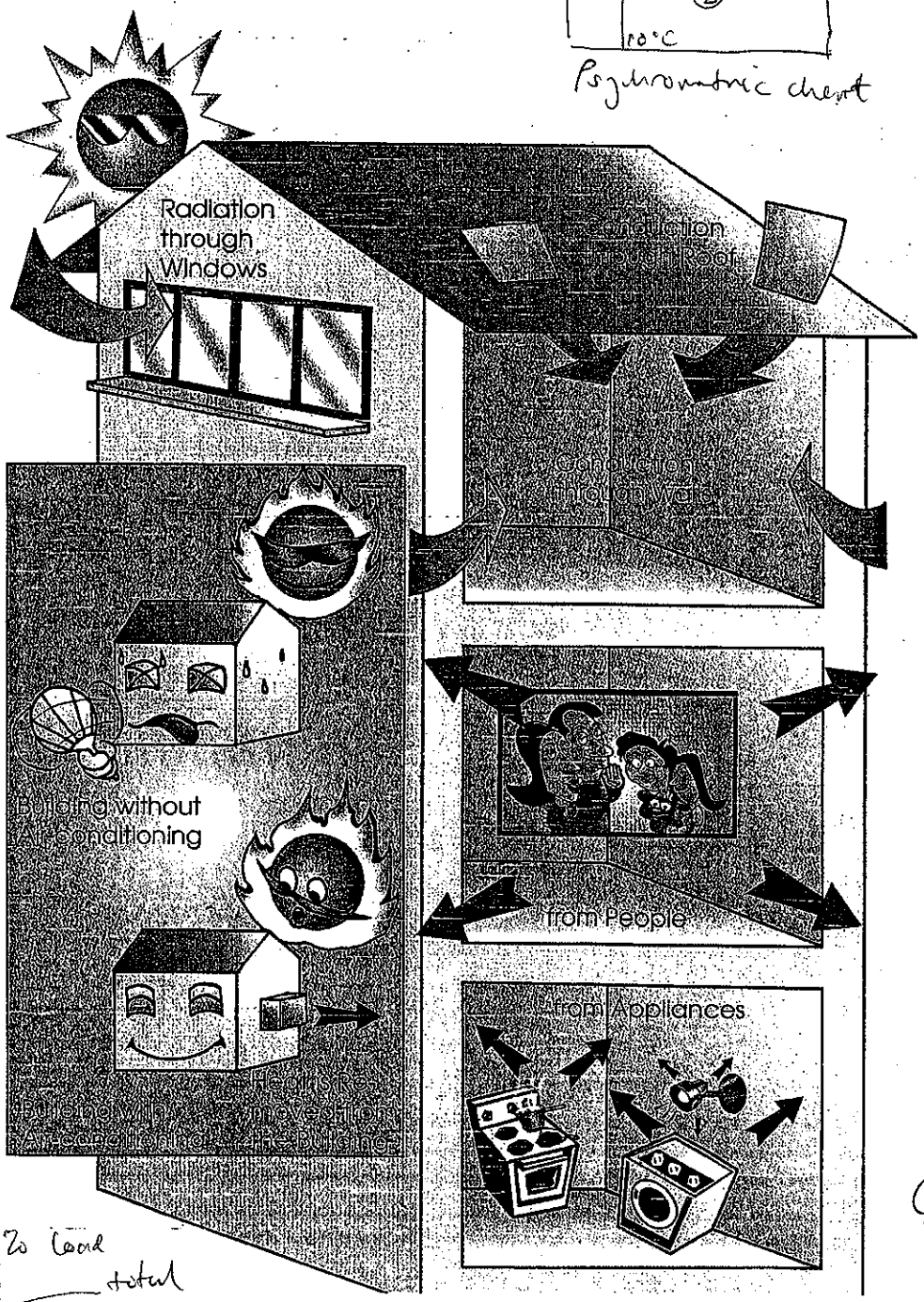
Cooling coil load = Room load + outdoor load

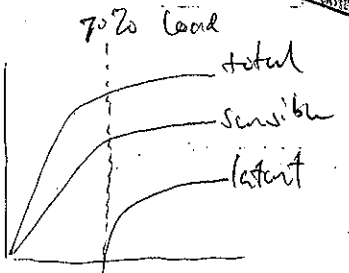


① 28°C, 35°C
② 23°C, 50%



Psychrometric chart





← can't remove latent load if partial load

Cooling coil performance curve

Control valve for chilled water

- Max load = 85% to 90% open
- Min load = 10% to 15% open

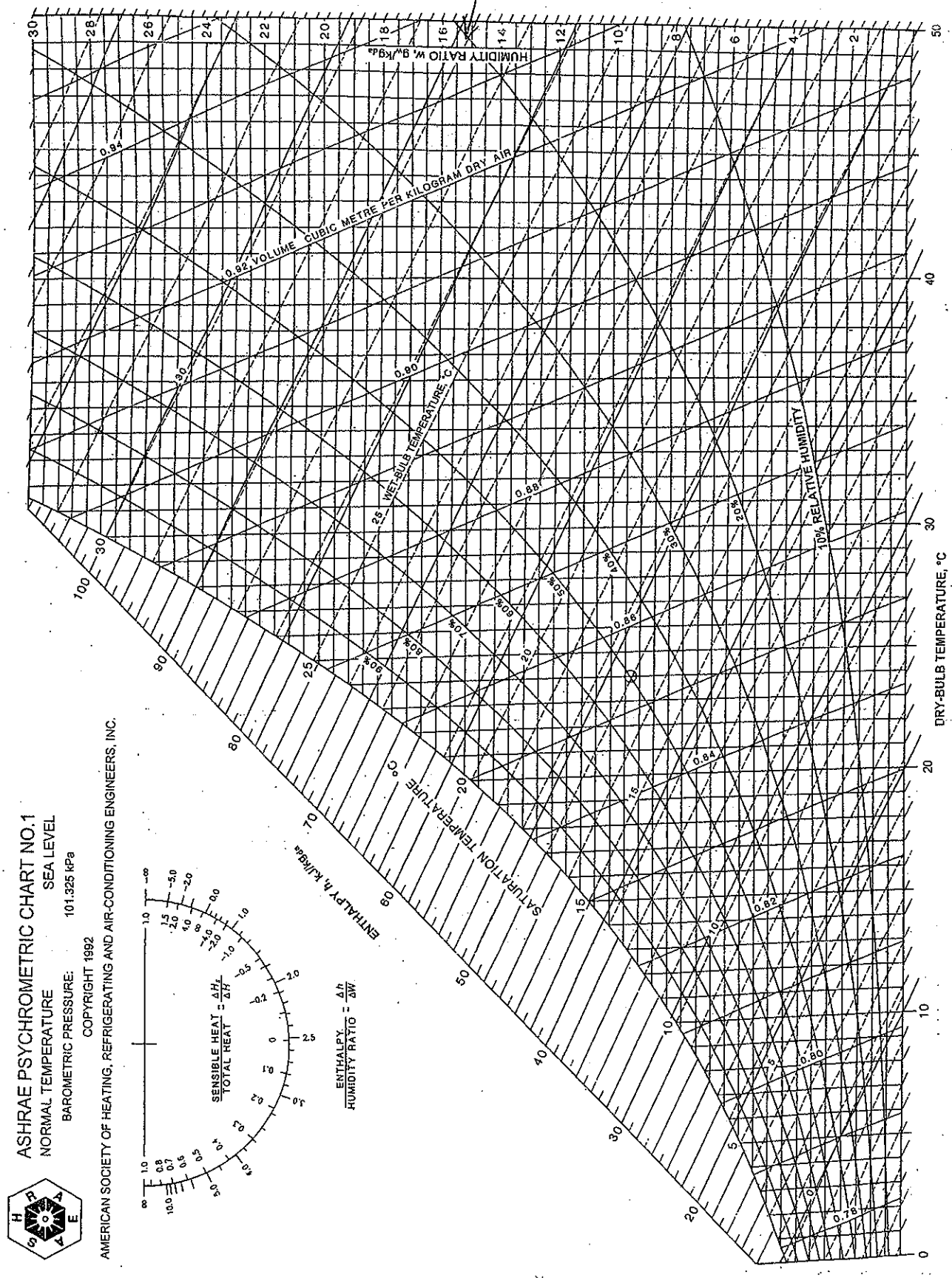


Fig. 1 ASHRAE Psychrometric Chart No. 1

UNITS AND CONVERSIONS

Table 1 Conversions to I-P and SI Units

(Multiply I-P values by conversion factors to obtain SI; divide SI values by conversion factors to obtain I-P)

Multiply I-P	By	To Obtain SI
acre (43,560 ft²)	0.4047	ha
	4046.873	m²
atmosphere (standard)	*101.325	kPa
bar	*100	kPa
barrel (42 U.S. gal, petroleum)	159.0	L
	0.1580987	m³
Btu (International Table)	1055.056	J
Btu (thermochemical)	1054.350	J
Btu/ft² (International Table)	11,356.53	J/m²
Btu/ft³ (International Table)	37,258.951	J/m³
Btu/gal	278,717.1765	J/m³
Btu·ft/h·ft²·°F	1.730735	W/(m·K)
Btu·in/h·ft²·°F (thermal conductivity *k*)	0.1442279	W/(m·K)
Btu/h	0.2930711	W
Btu/h·ft²	3.154591	W/m²
Btu/h·ft²·°F (overall heat transfer coefficient *U*)	5.678263	W/(m²·K)
Btu/lb	*2.326	kJ/kg
Btu/lb·°F (specific heat c_p)	*4.1868	kJ/(kg·K)
bushel (dry, U.S.)	0.0352394	m³
calorie (thermochemical)	*4.184	J
centipoise (dynamic viscosity μ)	*1.00	mPa·s
centistokes (kinematic viscosity *v*)	*1.00	mm²/s
clo	0.155	m²·K/W
dyne	1.0×10^{-5}	N
dyne/cm²	*0.100	Pa
EDR hot water (150 Btu/h)	43.9606	W
EDR steam (240 Btu/h)	70.33706	W
EER	0.293	COP
ft	*0.3048	m
	*304.8	mm
ft/min, fpm	*0.00508	m/s
ft/s, fps	*0.3048	m/s
ft of water	29,989.07	Pa
ft of water per 100 ft pipe	98.1	Pa/m
ft²	0.092903	m²
ft²·h·°F/Btu (thermal resistance *R*)	0.176110	m²·K/W
ft²/s (kinematic viscosity *v*)	92,900	mm²/s
ft³	28.316846	L
	0.02832	m³
ft³/min, cfm	0.471947	L/s
ft³/s, cfs	28.316845	L/s
ft·lb$_f$ (torque or moment)	1.355818	N·m
ft·lb$_f$ (work)	1.356	J
ft·lb$_f$/lb (specific energy)	2.99	J/kg
ft·lb$_f$/min (power)	0.0226	W
footcandle	10.76391	lx
gallon (U.S., *231 in³)	3.785412	L
gph	1.05	mL/s
gpm	0.0631	L/s
gpm/ft²	0.6791	L/(s·m²)
gpm/ton refrigeration	0.0179	mL/J
grain (1/7000 lb)	0.0648	g
gr/gal	17.1	g/m³
gr/lb	0.143	g/kg
horsepower (boiler) (33 470 Btu/h)	9.81	kW
horsepower (550 ft·lb$_f$/s)	0.7457	kW
inch	*25.4	mm
in. of mercury (60°F)	3.37	kPa
in. of water (60°F)	249	Pa
in/100 ft, thermal expansion	0.833	mm/m

To Obtain I-P	By	Divide SI

Multiply I-P	By	To Obtain SI
in·lb$_f$ (torque or moment)	113	mN·m
in²	645.16	mm²
in³ (volume)	16.3874	mL
in³/min (SCIM)	0.273117	mL/s
in³ (section modulus)	16387	mm³
in⁴ (section moment)	416 231	mm⁴
kWh	*3.60	MJ
kW/1000 cfm	2.118880	kJ/m³
kilopond (kg force)	9.81	N
kip (1000 lb$_f$)	4.45	kN
kip/in² (ksi)	6.895	MPa
litre	*0.001	m³
met	58.15	W/m²
micron (μm) of mercury (60°F)	133	mPa
mile	1.609	km
mile, nautical	*1.852	km
mile per hour (mph)	1.609344	km/h
	0.447	m/s
millibar	*0.100	kPa
mm of mercury (60°F)	0.133	kPa
mm of water (60°F)	9.80	Pa
ounce (mass, avoirdupois)	28.35	g
ounce (force or thrust)	0.278	N
ounce (liquid, U.S.)	29.6	mL
ounce inch (torque, moment)	7.06	mN·m
ounce (avoirdupois) per gallon	7.489152	kg/m³
perm (permeance at 32°F)	5.72135×10^{-11}	kg/(Pa·s·m²)
perm inch (permeability at 32°F)	1.45362×10^{-12}	kg/(Pa·s·m)
pint (liquid, U.S.)	4.73176×10^{-4}	m³
pound		
lb (avoirdupois, mass)	0.453592	kg
	453.592	g
lb$_f$ (force or thrust)	4.448222	N
lb$_f$/ft (uniform load)	14.59390	N/m
lb/ft·h (dynamic viscosity μ)	0.4134	mPa·s
lb/ft·s (dynamic viscosity μ)	1490	mPa·s
lb$_f$·s/ft² (dynamic viscosity μ)	47.88026	Pa·s
lb/h	0.000126	kg/s
lb/min	0.007559	kg/s
lb/h [steam at 212°F (100°C)]	0.2843	kW
lb$_f$/ft²	47.9	Pa
lb/ft²	4.88	kg/m²
lb/ft³ (density, ρ)	16.0	kg/m³
lb/gallon	120	kg/m³
ppm (by mass)	*1.00	mg/kg
psi	6.895	kPa
quad (10^{15} Btu)	1.055	EJ
quart (liquid, U.S.)	0.9463	L
square (100 ft²)	9.29	m²
tablespoon (approximately)	15	mL
teaspoon (approximately)	5	mL
therm (U.S.)	105.5	MJ
ton, long (2240 lb)	1.016	Mg
ton, short (2000 lb)	0.907	Mg; t (tonne)
ton, refrigeration (12 000 Btu/h)	3.517	kW
torr (1 mm Hg at 0°C)	133	Pa
watt per square foot	10.76	W/m²
yd	*0.9144	m
yd²	0.8361	m²
yd³	0.7646	m³

To Obtain I-P	By	Divide SI

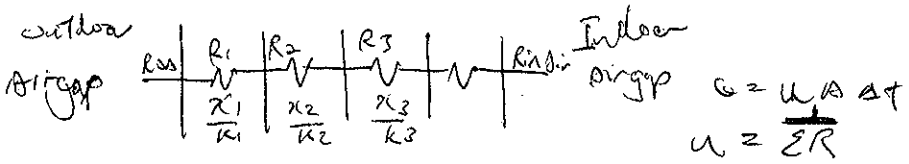
*Conversion factor is exact.

Notes: 1. Units are U.S. values unless noted otherwise.

2. Litre is a special name for the cubic decimetre. 1 L = 1 dm³ and 1 mL = 1 cm³.

The preparation of this chapter is assigned to TC 1.6, Terminology.

1 g/kg = 6.993 grain/lb



Chapter Building Heat Transmission

Apr 2008

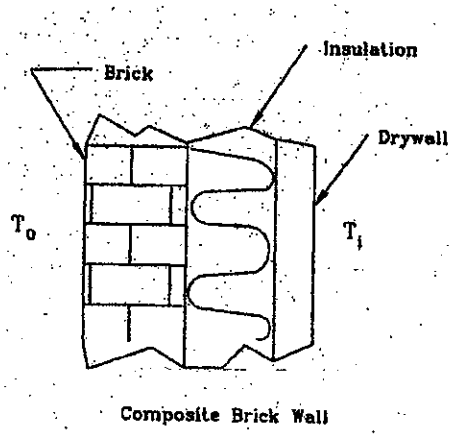
Introduction

- The rate of heat transfer per unit area through any given surface is a function of the temperature difference between the two sides and the overall thermal conductance through the composite surface. In order to estimate the heat loss or gain for a building, it is first necessary to determine an overall heat transfer coefficient for each type of surface and the area of each surface exposed to a temperature difference.

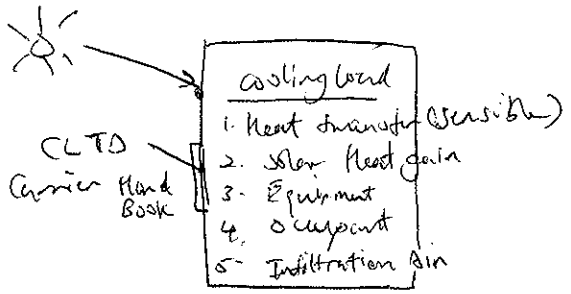
1. Overall Heat transfer coefficient

- In air-conditioning system design, it is necessary to calculate the heat transferred through a composite building section rather through a single element.
- In order to calculate the overall heat transfer through a composite section, a term called the overall heat transfer coefficient (U value) has been developed. The overall heat transfer coefficient (U value) is a single combined coefficient for conduction through all materials and the convection heat transfer through the surface film.

Figure 1 Composite Brick Wall



Composite Brick Wall



latent load
1. Occupant
2. Outdoor (Infiltration Air)

U -Value

- The U – value of the composite wall is a combination of the outside convection coefficient, the thermal conductance of the brick, the thermal conductance of the insulation, the conductance of the drywall, and the inside convection coefficient.

$q = UA(T_i - T_0)$

$q = the\ heat\ transfer\ rate\ in\ W$

$U = the\ overall\ heat\ transfer\ coefficient, W / m^2K$

$A = wall\ surface\ area\ in\ m^2$

$T_i = inside\ air\ temperature, ^\circ C$

$T_0 = outside\ air\ temperature, ^\circ C$

Solar Heat Gain = A x SC

CLTD = Q = UA Δt

2 U-value calculation procedures

- The conductance of a material is an indication of the case with which heat is conducted through the material.
- The reciprocal of the conductance (the resistance) is a measure of the extend to which a material resists the flow of heat.
- The resistance to heat flow through a material is frequently referred to as the "R value" of the material.

$$R = \frac{1}{C}$$

U-value cal

$$R_t = R_0 + \frac{b_1}{k_1} + \frac{b_2}{k_2} + \cdots + \frac{b_n}{k_n} + R_i$$

$$U = \frac{1}{R_t}$$

where

$R_0 = Air\ film\ resis\tan ce\ of\ external\ surface, m^2K/W$

$R_i = air\ film\ resis\tan ce\ of\ \text{int}ernal\ surface, m^2K/W$

$K = thermal\ conductivity\ of\ basic\ material, W/mK$

$b = thickness\ of\ basic\ material. m$

Table 1 Air film Resistance

Type of Surface	Thermal resistance (m^2 °K/W)
A. Surface Film resistances for Walls	
1. Inside surface (R_i)	
(a) High Emissively	0.120
(b) Low Emissively	0.299
2. Outside surface (R_o) (High Emissivity)	0.044
B Surface Film Resistances for Roofs	
1. Inside surface (R_i)	
(a) High Emissivity	
(i) Flat roof	0.162
(ii) Sloped roof 221/2°	0.148
(iii) Sloped roof 45°	0.133
(b) Low Emissivity	
(i) Flat roof	0.801
(ii) Sloped roof 221/2°	0.595
(iii) Sloped roof 45°	0.391

AIR CONDITIONING COOLING LOAD CALCULATIONS

a) PROJECT NAME..... ..... Restaurant, Singapore Date
b) EQUIPMENT NO..... .....
c) LOCATION SERVED... ..... mid-floor of office building

d) OUTDOOR CONDITIONS ..... 35 deg C DB 147.0 gr/lb
28 deg C WB

e) ROOM CONDITIONS... ..... 23 deg C DB 56.0 gr/lb
50 % RH

f) FLOOR AREA..... ..... 264 sq m 2842.2 sq ft

g) DESIGN TIME..... MONTH 9 13 .00 hr (between 8.00 hr to 20.00 hr)

I	EXTERNAL WINDOWS Description	Facing No	Area (sq m)	Shading Coeffn	U-value (W/m2K)	Solar (BTU/h ft2)	Q(sens) (Watts)	Q(latent) (Watts)
1	west window	7	20.0	0.70	3.000	47	2,804	
2	sliding door	5	6.0	0.80	3.000	14	429	
3	s glass panel	5	27.0	0.70	3.000	14	1,810	
						0	0	
						0	0	
II	SOLAR GAIN THRO' WALLS AND ROOF	Facing No	Area (sq m)	U-value (W/m2K)	EQTD (deg F)	Sub-Total	5,042	
1	south wall	5	12	0.33	25		55	
2	west wall	6	76	0.330	8		111	
					0		0	
					0		0	
					0		0	
III	GAIN THROUGH INTERNAL WALLS AND CEILINGS		AREA (sq m)	U-value (W/m2k)	T diff (deg C)	Sub-Total	166	
1	wall adj to store		96	0.33	3		95	
							0	
							0	
							0	
							0	
IV	INTERNAL HEAT GAIN					Sub-total	95	
a)	Lighting Load.		.....Area x 1.25 x		40	W/sq m	13,200	
b)	Equipment Load		.....		280	W (Sens)	280	
c)	People		140	Nos @...	72	W (Sens)	10,080	
					61	W (Lat)		8,540
	Supply air temp diff =		10	deg C		Sub-Total	23560	8540
	Supply Air Qty =		8556	CMH		TOTAL ROOM LOAD	28864	8540
V	Outdoor Air @		27.0	CMH/per =	3780.0	CMH	8,487	40,521
	SUMMARY		0.77	Room SHF	10.58	TR (Sens)	37,350	49,061
			103.4	Btu/hft2	24.48	TR (total)	86,411	W (Total)

AIR CONDITIONING COOLING LOAD CALCULATIONS

a) PROJECT NAME..... ..... office, singapore Date:
b) EQUIPMENT NO..... .....
c) LOCATION SERVED... ..... mid-floor of office building

d) OUTDOOR CONDITIONS ..... 35 deg C DB 147.0 gr/lb
28 deg C WB

e) ROOM CONDITIONS... ..... 23 deg C DB 56.0 gr/lb
50 % RH

f) FLOOR AREA..... ..... 264 sq m 2842.2 sq ft

g) DESIGN TIME..... MONTH 9 13 .00 hr (between 8.00 hr to 20.00 hr)

I	EXTERNAL WINDOWS Description	Facing No	Area (sq m)	Shading Coeffn	U-value (W/m2K)	Solar (BTU/h ft2)	Q(sens) (Watts)	Q(latent) (Watts)
1	west window	7	20.0	0.70	3.000	47	2,804	
2	sliding door	5	6.0	0.80	3.000	14	429	
3	s glass panel	5	27.0	0.70	3.000	14	1,810	
						0	0	
						0	0	
II	SOLAR GAIN THRO' WALLS AND ROOF	Facing No	Area (sq m)	U-value (W/m2K)	EQTD (deg F)	Sub-Total	5,042	
1	south wall	5	12	0.33	25		55	
2	west wall	6	76	0.330	8		111	
					0		0	
					0		0	
					0		0	
III	GAIN THROUGH INTERNAL WALLS AND CEILINGS		AREA (sq m)	U-value (W/m2k)	T diff (deg C)	Sub-Total	166	
1	wall adj to store		96	0.33	3		95	
							0	
							0	
							0	
							0	
IV	INTERNAL HEAT GAIN					Sub-total	95	
a)	Lighting Load.		.....Area x 1.25 x		20	W/sq m	6,600	
b)	Equipment Load		.....	.....	6600	W (Sens)	6,600	
c)	People		27	Nos @...	72	W (Sens)	1,944	
					61	W (Lat)		1,647
	Supply air temp diff =		10	deg C		Sub-Total	15144	1647
	Supply Air Qty =		6061	CMH		TOTAL ROOM LOAD	20448	1647
V	Outdoor Air	@	27.0	CMH/per =	729.0	CMH	1,637	7,815
	SUMMARY		0.93	Room SHF	6.26	TR (Sens)	22,084	9,462
			37.7	Btu/hft2	8.94	TR (total)	31,546	W (Total)

FORMULAE FOR EXAMPLE

1. CONVERT TO SQ FT - G11:(F1) +E11*10.766

2. LOOKUP SOLAR INSOLATION
   H18 : [W10] @IF(D18=0,0, @IF(F13<8,0,@IF(F13>17,0,
   @IF(E13 = 6,@HLOOKUP(D18,N2..V12,F13-7),
   @HLOOKUP(D18,N13..V23,F13-7)))))

3. HEAT FLOW THROUGH WALLS
   J18: (,0) [W10] +E18*(F18*H18*10.766/3.4+G18*(E5-E8))

4. LOOKUP EQV TEMP FOR WALLS
   G27 :@IF(D27=0,0,@IF(F13<8,0,@HLOOUP(D27,N26..V41,F13-7)))

5. HEAT FLOW THROUGH WALLS -J27: (,0) [W10] +E27*F27*G27/1.8

6. HEAT FROM LIGHTING -J44: (,0) [W10] +E11*G44*1.25

7. EQUIPMENT HEAT LOAD -J45: (,0) [W10] +G45

8. PEOPLE SENSIBLE HEAT -J46: (,0) [W10] +E46*G46

9. PEOPLE LATENT HEAT -K47: (,0) [W10] +E46*G47

10. SUPPLY AIR QUANTITY
    -E51: (F0)+RMSENSQ*3.4/(E49*1.8*1.08*0.59)

11. OUTDOOR AIR QUANTITY -G53 :(F1) +E53*E46

12. OUTDOOR SENSIBLE HEAT -
    J53 : (,0) [W10] +G53*0.589*1.08*(E5-E8)/3.4

13. OUTDOOR LATENT HEAT
    K53 : (,0) [W10] +G53*0.589*0.68*(G5-G8)/3.4

14. SENSIBLE HEAT FACTOR -E55:(F2) +RMSENSQ/(RMSENSQ + K51)

15. SENSIBLE LOAD IN TR -G55:(F2) +J55*3.4/12000

16. LATENT LOAD IN TR -E56: (F1) +J56*3.4/G11

17. TOTAL LOAD IN WATTS -G56: (F2) +J56*3.4/12000

Cooling load Example

A restaurant located in the mid-floor of a multi-storey office building in the centre business district, specializes in serving business lunches to office executives. It also has its own air-conditioning plant for independent operation.

The restaurant has just changed ownership. The new management strategy to increase business is to extend operation into Sundays and Public Holidays. To ensure crowd, popular seafood menu is offered at reduced rate.

Estimate the mid-day increased peak load of the restaurant during during Sundays and Public Holicdays.

The restaurant floor plan is as shown in Figure 1 and data available are given as below :

A) Indoor design condition --- 23 CDB, 55% RH

B) Indoor design condition (non-air-conditioned) --- 26 CDB

C) Outdoor condition ---------- 32 CDB, 29 CWD

D) Restaurant staff (in air-conditioned space) -------- 16 people

E) Maximum number of lunch –diners ----------120 people

F) Lighting --- 30% fluorescent, 70% incandesant -- 40W/m^2

G) Average food load per person

50% sensible, 50% latent ---------70W Sensible, 70 W latent

h) Electric tea and hot water urn are considered as equivalent to electric coffee urn – 10 litre

i) Floor to floor height ---------- 4 m

Assumption:

1. All wall, partition and floor slab are to be considered as of medium construction type
2. Return air to be consider through ceiling return
3. Sliding door to be considered as part of glass-panel
4. Kitchen dry Bulb temperature to e same as outside air conditioned

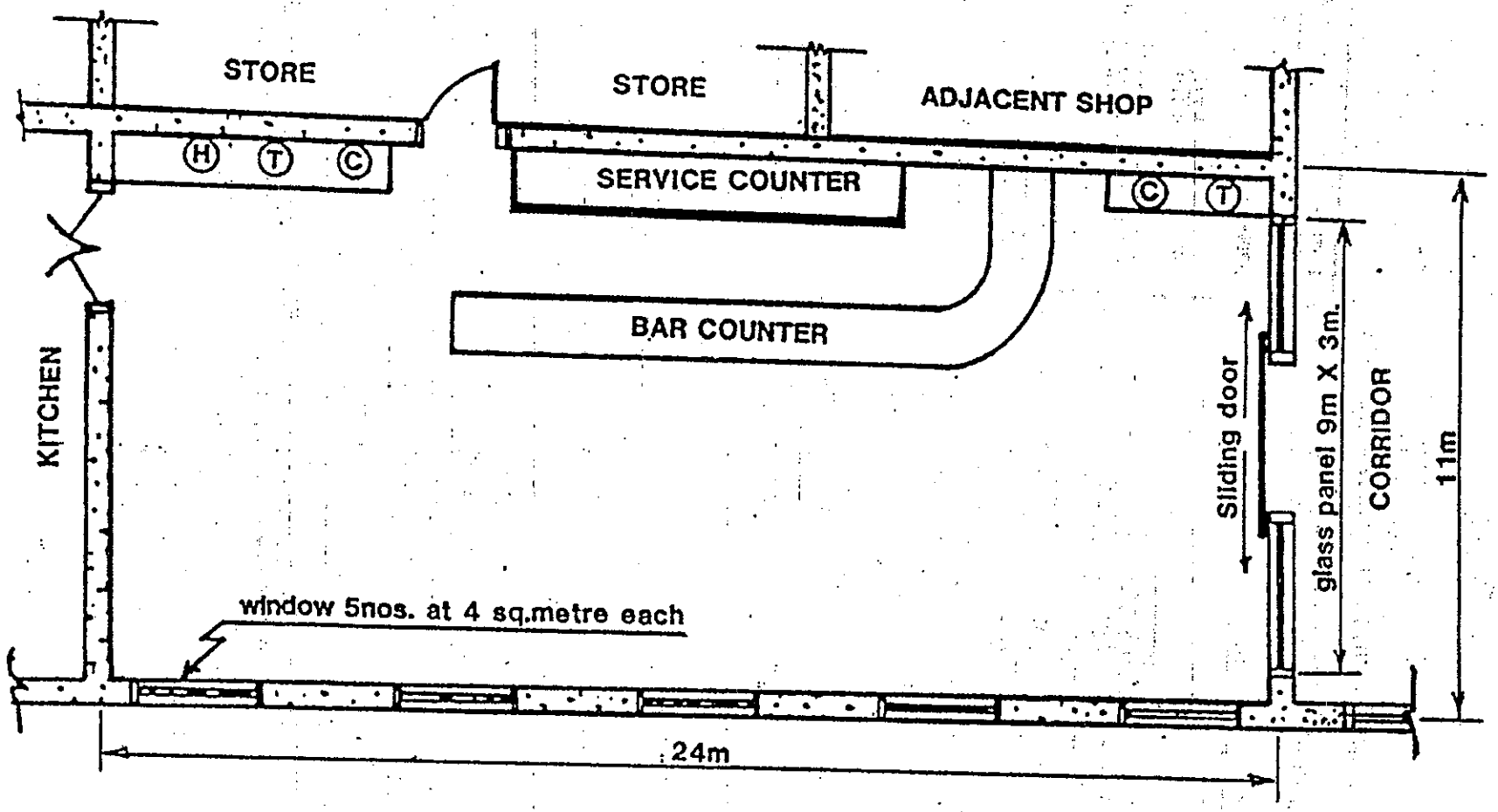
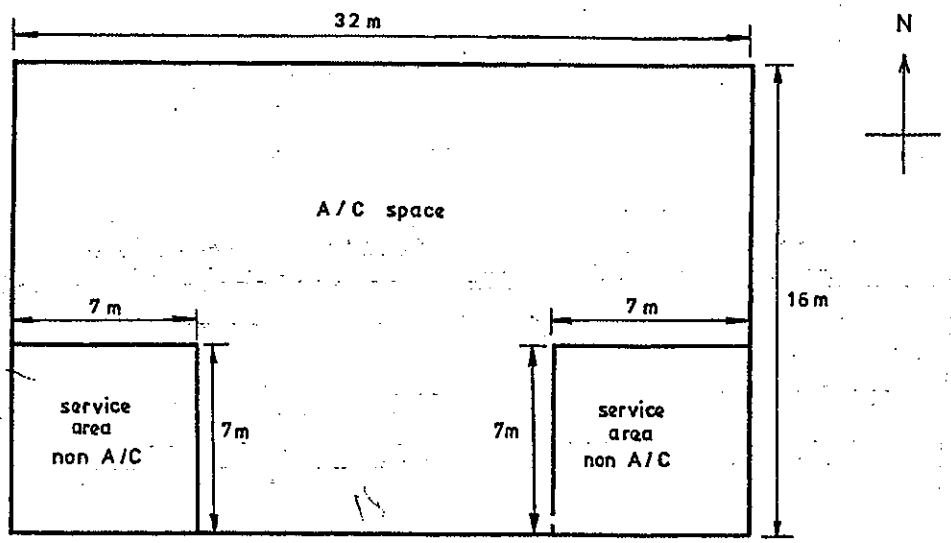

STORE
STORE
ADJACENT SHOP
SERVICE COUNTER
BAR COUNTER
KITCHEN
Sliding door
glass panel 9m X 3m.
CORRIDOR
11m
window 5nos. at 4 sq.metre each
24m
Legend
Ⓒ : coffee urn
Ⓣ : tea urn
Ⓗ : hot water urn
RESTAURANT(SEAFOOD)- FLOOR PLAN
E
N
S
W

FIGURE 1.

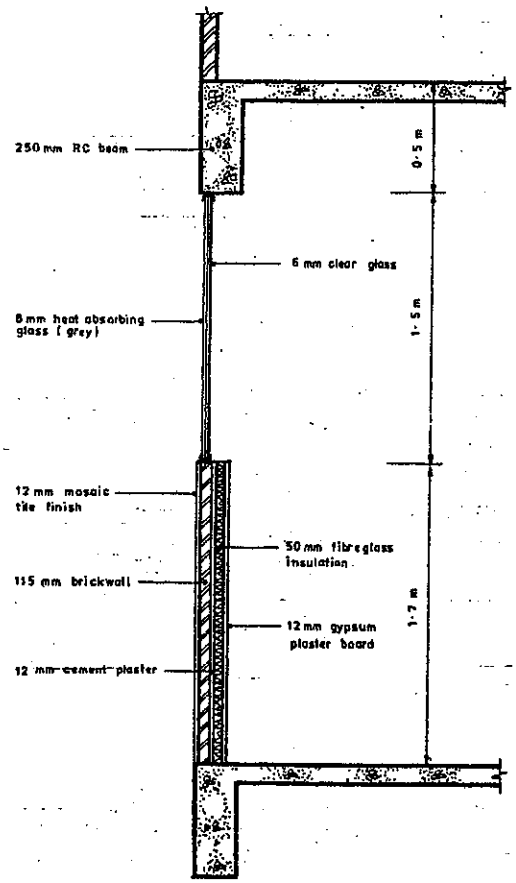
\0 /wgzy

SR NO	CODE NO	DESCRIPTION	DENSITY	K-VALUE
1	10	Asbestos cement sheet	1488	0.317
	20	Asbestos insulating board	720	0.108
	30	Asphalt, roofing	2240	1.226
	40	Bitumen		1.298
	51	Brick, dry	1760	0.807
	52	Brick, fairface	1760	1.154
	61	Concrete	2400	1.442
	62	Concrete	64	0.144
	71	Concrete, lightweight	960	0.303
	72	Concrete, lightweight	1120	0.346
	73	Concrete, lightweight	1280	0.476
	110	Glass, sheet	2512	1.053
	120	Glass wool, mat or guilt (dry)	32	0.035
	151	Metal, aluminium	2672	211
	152	Metal, copper	8784	385
	153	Metal, steel	7840	47.6
	171	Plaster, gypsum	1216	0.37
	172	Plaster, perlite	616	0.115
	173	Plaster, sand/cement	1568	0.533
	174	Plaster, vermiculite	960	0.303
	190	Polyurethane, foam	24	0.024
	201	Timber, Softwood	608	0.125
	202	Timber, Hardwood	720	0.138
	203	Timber, Plywood	528	0.138
	210	Vermiculite, loose granules	80	0.065
	220	Wood chipboard	800	0.144
	231	Woodwool slab, light	400	0.096
	232	Woodwool slab, heavy	480	0.101
	311	Air space, high emmissivity, 5mm		0.011
	312	Air space, low emmissive, 5mm		0.25
	313	Air space, high emmissive, 20mm		0.148
	314	Air space, low emmissive, 20mm		0.578
	315	Air space, high emmissive, 100mm		0.16
	316	Air space, low emmissive, 100mm		0.606
	410	Inside airfilm, high emissivity		0.12
	411	Inside airfilm, low emissivity		0.299
	420	Outside airfilm		0.044

Examples of U value Calculation for a typical singapore office building



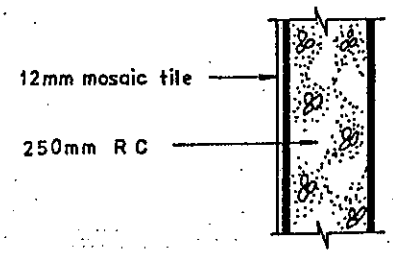
FLOOR PLAN



SECTIONAL DETAILS OF ENVELOPE

U-value Calculation

a) For rc beam



Component	$\frac{b}{K}$	R
outside air film		0.044
mosaic tile	$\frac{0.012}{1.298}$	0.009
rc	$\frac{0.250}{1.442}$	0.173
inside air film		0.120
Total R		0.346

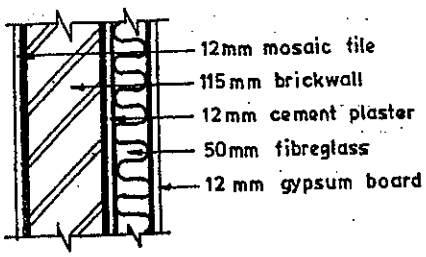
$$U = \frac{1}{R} = \frac{1}{0.346}$$

$$= 2.89 \text{ W/m}^2 \text{ °K}$$

Weight = $2640 \times 0.012 + 2400 \times 0.25 = 632 \text{ kg/m}^2$

TDeq = 10 °K

b) For brickwall



Component	$\frac{b}{K}$	R
outside air film		0.044
mosaic tile	$\frac{0.012}{1.298}$	0.009
brickwall	$\frac{0.115}{0.807}$	0.143
cement plaster	$\frac{0.012}{0.533}$	0.023
fibreglass	$\frac{0.050}{0.035}$	1.429
gypsum board	$\frac{0.012}{0.170}$	0.071
inside air film		0.120
Total R		1.839

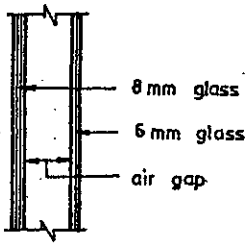
$$U = \frac{1}{R} = \frac{1}{1.839}$$

$$= 0.54 \text{ W/m}^2 \text{ °K}$$

Weight = $2640 \times 0.012 + 1760 \times 0.115$
$+ 1568 \times 0.012 + 32 \times 0.05 + 880 \times 0.012 = 265 \text{ kg/m}^2$

TDeq = 10 °K

c) For glass window



Component	$\frac{b}{K}$	R
outside air film		0.044
outer glass	$\frac{0.008}{1.053}$	0.008
air space		0.160
inner glass	$\frac{0.006}{1.053}$	0.006
inside air film		0.120
Total R		0.338

$$U = \frac{1}{R} = \frac{1}{0.338}$$

$$= 2.96 \text{ W/m}^2 \text{ °K}$$

SC = 0.5 (given)

TABLE I(1) K-VALUES OF BASIC MATERIALS

Sr No	Material	Density kg/m^3	K-value W/m°K
1	Asbestos cement sheet	1488	0.317
2	Asbestos insulating board	720	0.108
3	Asphalt, roofing	2240	1.226
4	Bitumen		1.298
5	Brick:		
	(a) dry (covered by plaster or tiles outside)	1760	0.807
	(b) common brickwall (brickwall directly exposed to weather outside)		1.154
6	Concrete	2400 64	1.442 0.144
7	Concrete, light weight	960 1120 1280	0.303 0.346 0.476
8	Cork board	144	0.042
9	Fibre board	264	0.052
10	Fibre glass (see Glass Wool and Mineral Wool)		
11	Glass, sheet	2512	1.053

Sr No	Material	Density Kg/m^3	K-value W/m ^{o}K
12	Glass wool, mat or guilt (dry)	32	0.035
13	Gypsum plaster board	880	0.170
14	Hard board:		
	(a) standard	1024	0.216
	(b) medium	640	0.123
15	Metals:		
	(a) Aluminium alloy, typical	2672	211
	(b) copper, commercial	8784	385
	(c) Steel	7840	47.6
16	Mineral wool, felt	32 – 104	0.035 – 0.032
17	Plaster		
	(a) gypsum	1216	0.370
	(b) perlite	616	0.115
	(c) sand/cement	1568	0.533
	(d) vermiculite	640 – 960	0.202–0.303
18	Polystyrene, expanded	16	0.035
19	Polyurethane, foam	24	0.024
20	PVC flooring	1360	0.713
21	Soil, loosely packed	1200	0.375
22	Stone, tile:		
	(a) sand stone	2000	1.298
	(b) granite	2640	2.927
	(c) marble/terrazzo/ceramic/ mosaic	2640	1.298
23	Tile, roof	1890	0.836
24	Timber:		
	(a) across grain softwood	608	0.125
	(b) hardwood	702	0.138
	(c) plywood	528	0.138
25	Vermiculite, loose granules	80 – 112	0.065
26	Wood chipboard	800	0.144
27	Woodwool slab	400	0.086
		480	0.101

TABLE I (2) Surface Film Resistances for Walls and Roofs

Type of Surface				Thermal Resistance m^2 °K/W
A	Surface Film Resistances for Walls			
	1	Inside surface (Ri)		
		(a)	High Emissivity	0.120
		(b)	Low Emissivity	0.299
	2	Outside surface (Ro) (High Emissivity)		0.044
B	Surface Film Resistances for Roofs			
	1	Inside surface (Ri)		
		(a)	High Emissivity	
			(i) Flat roof	0.162
			(ii) Sloped roof 22½°	0.148
			(iii) Sloped roof 45°	0.133
		(b)	Low Emissivity	
			(i) Flat roof	0.801
			(ii) Sloped roof 22½°	0.595
			(iii) Sloped roof 45°	0.391
	2	Outside surface (Ro) (High Emissivity)		
			Flat or sloped	0.055

Note: 1) Ordinarily, high emissivity is assumed for surfaces of building materials with reasonably smooth finishing. Low emissivity applies only to internal surface if the surface is very reflective, such as that of an aluminium foil.

2) Interpolation between the angle of slope from horizontal to 45° is permitted. For angle beyond 45°, the value for 45° can be used; no extrapolation is needed.

TIME	S O L A R G A I N T A B L E								
	1	2	3	4	5	6	7	8	9
8 am	65	156	147	42	11	11	11	11	87
9 am	74	154	135	42	27	13	13	13	147
10	78	133	93	27	15	14	14	14	191
11	80	95	43	15	14	14	14	20	217
12	82	53	14	14	14	14	14	53	226
1	80	20	14	14	14	14	43	95	217
2	78	14	14	14	14	15	93	133	191
3	74	13	13	13	13	27	135	154	147
4	65	11	11	11	11	42	147	156	87
5	45	6	6	6	6	37	116	119	28
	1	2	3	4	5	6	7	8	9
8	12	118	167	118	12	12	12	12	100
9	13	101	151	101	13	13	13	13	163
10	14	68	107	68	14	14	14	14	210
11	14	31	47	31	14	14	14	14	240
12	14	14	14	14	14	14	14	14	250
1	14	14	14	14	14	31	47	31	240
2	14	14	14	14	14	68	107	68	210
3	13	13	13	13	13	101	151	101	163
4	12	12	12	12	12	118	167	118	32
5	6	6	6	6	6	95	134	95	0
6	0	0	0	0		0	0		
	N	NE	E	SE	S	SW	W	NW	Horizontal

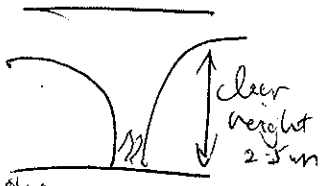
AIR CONDITIONING COOLING LOAD CALCULATIONS

6	0 EQUIVALENT TEMP DIFF								
	1	2	3	4	5	6	7	8	9
8am	3	3	3	3	3	3	3	3	6
9 am	3	5	21	13	4	3	3	3	7
10 am	4	24	30	20	5	4	4	4	8
11 am	4	22	31	24	7	5	4	4	11
12 noon	4	20	31	28	20	6	5	5	16
1 pm	5	15	19	25	25	8	7	6	22
2 pm	6	10	14	21	24	12	10	10	27
3 pm	8	11	13	18	25	24	19	12	31
4 pm	10	12	12	15	26	32	26	21	35
5 pm	11	13	13	14	23	35	34	30	38
6 pm	12	14	14	13	20	36	37	31	38
7 pm	12	13	13	12	15	35	41	32	38
8 pm	12	12	12	11	12	34	36	21	36
9 pm	10	11	11	10	10	20	28	12	34
10 pm	8	10	11	10	9	10	16	8	31
	NE	E	SE	S	SW	W	NW	N	Roof

Fire size: 5MW → M_{smoke} → smoke temp (above 600°C)

CP13, FSB – Chapter 7

- M_{smoke} = Max 175 kg/s
- Min smoke temp rise: 18°C above ambient
- max length of smoke reservoir – 60 m

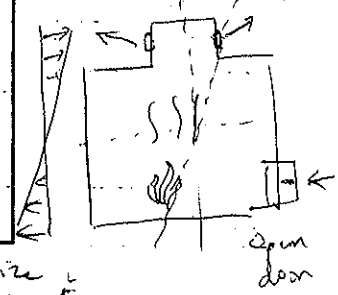


Chapter Eight

Zone smoke control

Apr 2008

- Size of smoke zone
  1. In the origin of fire
     2000 m² = NV
     2600 m² = MV
  2. Smoke allowed to enter circulation area:
     smoke zone: 1000 m² = NV
     1300 m² = MV



- Fire size:
  shops 5 MW 12 m perimeter
  offices 1 MW 14 m "
  Hotel guest room 0.5 MW 6 m "

- CIBSE, FRE

$m_s = 0.18 P Y^{3/2}$

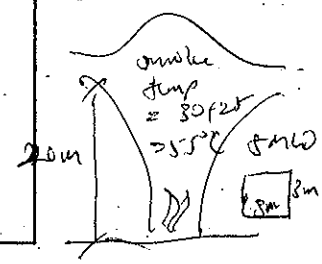
$= 0.18 \times 12 \times 20^{3/2}$

$= 193$ kg/s

$Q = m_s C_p \Delta t$

$\Delta t = \frac{5000 \text{ kW}}{193 \times 1.02}$

$= 25°C$



Purpose of Zoned Smoke Control

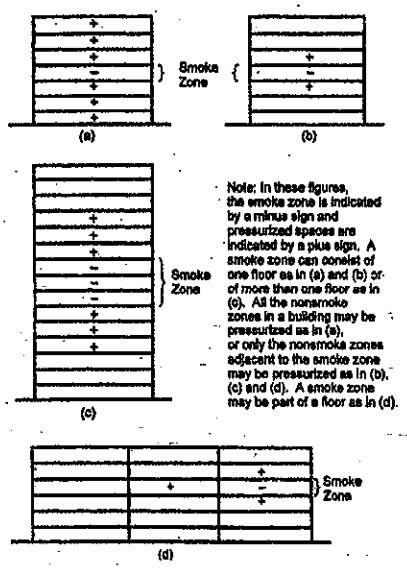
- The stairwell systems and elevator systems discussed in previous chapters were primarily intended to prevent smoke infiltration into these shafts.
- However, smoke can flow through cracks in floors and through unpressurized shafts to damage property and threaten life at locations remote from the fire.
- The concept of zone smoke control discussed in this chapter is to limit this type of smoke movement within a building.

Fire damper (total shut off) vs smoke damper (control by actuator)

Smoke Control Zones

- A building can be divided into a number of smoke zones, each separated from the others by partitions and floors.
- In the event of a fire, pressure differences produced by mechanical fans are used to limits the smoke spread to the zone in which the fire initiated.
- The concentration of smoke in this zone goes unchecked.

Figure 1 some arrangement of smoke control zones



- In zone smoke control systems, it is intended that occupants evacuate the smoke zone as soon as possible after the fire detection.
- Frequently, each floor of a building is chosen to be a separate smoke control zone. However, a smoke control zone can consist of more than one floor, or a floor can consist of more than one smoke control zone.
- When a fire occurs, all of the non-smoke zones in the building, or only zones adjacent to the smoke zone may be pressurized.
- When the fire floor is exhausted and only adjacent floors are pressurized, the system is sometimes called a pressure sandwich.

Smoke Zone Venting

- exterior wall vents
- smoke shafts , and
- mechanical venting (or exhaust)

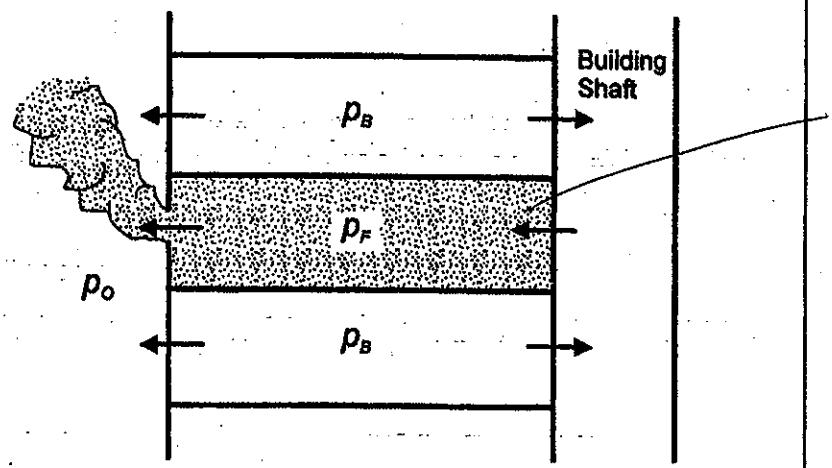
Exterior wall vents

- Exterior wall vents can consist of windows or panels that open automatically when the smoke control system is activated.
- The system considered here consists of a vented smoke zone without any mechanical exhaust and adjacent zones that are pressurized.
- In order to minimize adverse effects of wind, the area of wall vents should be evenly distributed among all of the exterior walls.
- For buildings that are much longer than wide, the vents can evenly be divided between two long sides.

Exterior wall vents ,2

- Exterior wall venting is most appropriate for buildings with open floor plans and least suitable when the floor plan is divided into many compartments. Because the flow of hot gases through a wall vent can be substantial, precautions should be taken in the design of exterior walls to minimize the possibility of exterior fire spread to floors above the vent.

Figure 2 Flow pattern due to smoke venting



$Q_o = A_v \sqrt{\Delta P_{fo}}$

Vent Areas

- For this system, the fire floor (smoke zone) is vented to the outside , supply and exhaust fans serving the fire floor are shut off, and the floors above and below the fire floor are pressurized.
- Air flows from floors adjacent to the fire floor and through the vent to the outside, as illustrated in Figure 12.2 . Because the supply and exhaust fans are shut off on the fire floor, the total airflow rate through the wall vents equals the total flow rate into the vented floor from the surrounding smoke control zones:

Vent areas, 2

$$A_v\sqrt{p_F - p_0} = A_e\sqrt{p_B - p_F}$$

A_V = *flow area of the exterior vent,* m^2

A_e = *effective flow area of the enclosure of the smoke zone to the other zones,* m^2

p_F = *smoke zone pressure, Pa*

p_0 = *outside pressure, Pa*

p_B = *building pressure on nonsmoke zones, Pa*

Vent areas

- The effective flow area, A_e, includes the flow areas of the walls of vertical shafts, floor constructions, and duct openings (return and exhaust) of the smoke zone.

$$\frac{p_B - p_F}{p_F - p_0} = \left(\frac{A_V}{A_e}\right)^2 \quad ---(1)$$

But $p_B - p_0 = (p_B - p_F) + (p_F - p_0)$

Let

$\Delta p_{B0} = p_B - p_0$

$\Delta p_{BF} = p_B - p_F$

$\Delta p_{F0} = p_F - p_0$

where

Δp_{BO} = *pressure difference from the nonsmoke zones to the outside, Pa*

Δp_{BF} = *pressure difference from the nonsmoke zone to the smoke zone, Pa*

Δp_{FO} = *pressure difference from the smoke zone to the outside, Pa*

Then

$\Delta p_{B0} = \Delta p_{BF} + \Delta p_{F0}$

$\Delta p_{F0} = \Delta p_{B0} - \Delta p_{BF}$

$$\frac{\Delta P_{BF}}{\Delta P_{BO} - \Delta P_{BF}} = \left(\frac{A_V}{A_e}\right)^2$$

$$\Delta P_{BF} = \Delta P_{BO}\left(\frac{A_V}{A_e}\right)^2 - \Delta P_{BF}\left(\frac{A_V}{A_e}\right)^2$$

$$\Delta P_{BF} + \Delta P_{BF}\left(\frac{A_V}{A_e}\right)^2 = \Delta P_{BO}\left(\frac{A_V}{A_e}\right)^2$$

$$\Delta P_{BF}\left[1 + \left(\frac{A_V}{A_e}\right)^2\right] = \Delta P_{BO}\left(\frac{A_V}{A_e}\right)^2$$

$$\frac{\Delta P_{BF}}{\Delta P_{BO}} = \frac{\left(\frac{A_V}{A_e}\right)^2}{1 + \left(\frac{A_V}{A_e}\right)^2}$$

Vent Areas

Substituting the above into eqn (1) and yields

$$\frac{\Delta p_{BF}}{\Delta p_{B0}} = \frac{\left(A_V / A_e\right)^2}{1 + \left(A_v / A_e\right)^2} \quad ---(2)$$

Pressurization air flow Rates, 1

- The effective flow area , A_e of the enclosure of the smoke zone to the other zones usually consists of the sum of the flow areas between the smoke zone and many other non-smoke zones.

$$A_e = \sum_{i=1}^{n} A_{BFi}$$

A_e = *effective flow area of the enclosure of the smoke zone to the other zones,* m^2

n = *number of adjacent non – smoke zones*

A_{BFi} = *flow area between non – smoke zone i and the smoke zone,* m^2

Pressurization of flow ,2

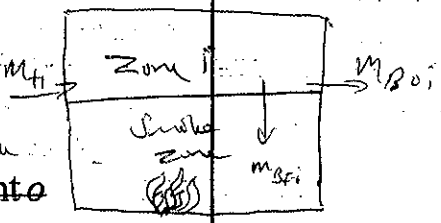
- Considering steady flow conditions,

$\dot{m}_{ti} = \dot{m}_{BFi} + \dot{m}_{BOi}$

$\dot{m}_{ti}$ = *mass flow rate of pressurization air into zone i, kg / s*

$\dot{m}_{BFi}$ = *mass flow rate of pressurization air into smoke zone, kg / s*

$\dot{m}_{BOi}$ = *mass flow rate from zone i to the outside, kg / s*



Pressurization,3

- The flow rate from zone i to the smoke zone can be expressed in the form of the orifice equation.

$$\dot{m}_{BFi} = K_m C A_{BFi} \sqrt{2\rho \Delta p_{BF}}$$

$\dot{m}_{BFi}$ = *mass flow rate from zone i to the smoke zone, kg / s*

C = *dimensionless flow coefficient*

A_{BFi} = *flow area between non – smoke zone i and the smoke zone, m^2*

Δp_{BF} = *pressure difference from the nonsmoke zones to the smoke zone, Pa*

$K_m = 1.00$

Pressurization air flow,4

- Mass flow rate to the outside is

$$\dot{m}_{BOi} = K_m C A_{BOi} \sqrt{2\rho \Delta p_{BO}}$$

$\dot{m}_{BOi}$ = *mass flow rate from zone i to the outside, kg / s*

C = *dimensionless flow coefficient*

A_{BOi} = *flow area between non – smoke zone i and the outside,* m^2

Δp_{BO} = *pressure difference from the nonsmoke zones to the outside, Pa*

$K_m = 1.00$

Pressurization flow rate,5

$$\dot{V}_{BFi} = K_f A_{BFi} \sqrt{\Delta p_{BF}}$$

$\dot{V}_{BFi}$ = *volume flow rate from zone i to the smoke zone,* m^3 / s

A_{BFi} = *flow area between nonsmoke zone i and the smoke zone,* m^2

Δp_{BF} = *pressure difference from the non – smoke zones to the smoke zone, Pa*

$K_f = 0.839$

Pressurization flow rate,6

$$\dot{V}_{BOi} = K_f A_{BOi} \sqrt{\Delta p_{BO}}$$

$\dot{V}_{BOi} = volume\ flow\ rate\ from\ zone\ i\ to\ the\ outside, m^3/s$

$A_{BOi} = Flow\ area\ between\ non-smoke\ zone\ i$ *and the outside,* m^2

$\Delta p_{BO} = pressure\ difference\ from\ the\ non-smoke\ zones$ *to the outside,* Pa

$K_f = 0.839$

Pressurization air flow ,7

- The pressure difference from the non-smoke is

$$\Delta p_{BO} = \Delta p_{BF} \left(\frac{1 + \left(A_v / A_e \right)^2}{\left(A_v / A_e \right)^2} \right)$$

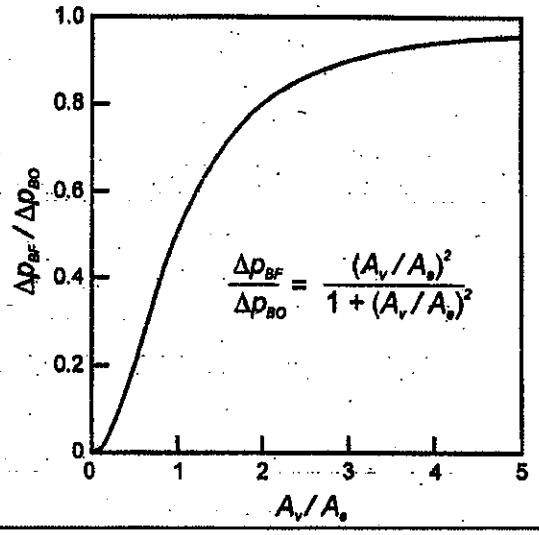
Δp_{BO} = *pressure difference from non-smoke zones to the outside,* Pa

Δp_{BF} = *pressure difference from the non-smoke zones to the smoke zone,* Pa

A_v = *flow area of the exterior vent of the fire floor,* m^2

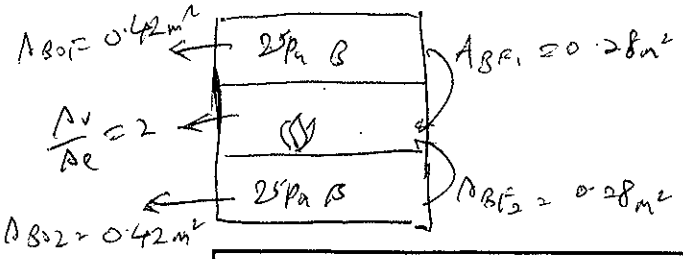
A_e = *effective flow area of the enclosure of the smoke zone to the other zones,* m^2

Figure 3 variation of pressure difference with size of exterior wall vents



Example 1 Supply air and exterior wall vents

- The smoke zone of a zoned smoke control system is to have exterior wall vents and two adjacent non-smoke zones. Supply and return are shut off to the smoke zone, and the adjacent zones are pressurized. The non-smoke zones have the same flow areas:



Example 1, cont

$$A_{BO1} = A_{BO2} = 0.42m^2$$

$$A_{BF1} = A_{BF2} = 0.28m^2$$

$$Use \frac{A_v}{A_e} = 2 \; and \; \Delta p_{BF} = 25Pa$$

How much pressurization air is needed ?

Example 1, Soln

$$A_e = 0.28 + 0.28 = 0.56\,m^2$$

$$\Delta p_{BO} = 25\left[\frac{1+2^2}{2^2}\right] = 31\,Pa$$

$$\dot{V}_{BF1} = 0.839(0.28)\sqrt{25} = 1.2\,m^3/s$$

$$\dot{V}_{BO1} = 0.839(0.42)\sqrt{31} = 2.0\,m^3/s$$

The sup *ply air is*

$$= 1.2 + 2.0 = 3.2\,m^3/s$$

$\frac{\Delta P_{BF}}{\Delta P_{BO}} = \frac{(A_v/A_e)^2}{1+(\frac{A_v}{A_e})^2}$

$0.827 A\sqrt{\Delta P}$ or $0.839 A\sqrt{\Delta P}$ depending on constant C used by some books (Eq 120)

Smoke Shaft, 1

- A smoke shaft is a vertical shaft intended to be a path for smoke movement from the fire floor to above the level of the roof.
- The driving force of smoke movement is buoyancy, although the flow through some smoke shafts is aided by mechanical fans.
- A smoke shaft can serve one floor, a group of floors, or all the floors in a building. Smoke shafts have openings above the roof level and on the floors they serve. These openings are fitted with dampers that are normally closed.
- In a fire situation, only the damper on the fire floor and the top outside damper open to vent smoke outside. Smoke shafts should be constructed in accordance with local codes.

Smoke shaft ,2

- Smoke shafts lend themselves to use in buildings with open floor plans. The air movement caused by smoke shafts operating during normal stack effect tends to pull smoke toward the smoke shaft inlet on the fire floor. It is recommended that smoke shafts be located as far as possible from exit stairwells, so that smoke in the vicinity of the shaft inlet does not pose an increased hazard during evacuation or fire fighting.
- Because hot smoke frequently stratifies near the ceiling, it is recommended that smoke shaft inlets be located in or near the ceiling.

Mechanical Exhaust

- Mechanical exhaust of the smoke zone can be accomplished by either a dedicated exhaust system or by the exhaust fans of the MVAC system. Generally, such exhaust is done in conjunction with pressurization of nonsmoke zones. These systems can also include stairwell pressurization.
- Mechanical exhaust by itself can result in sufficient pressure difference to control smoke. However, in the event of window breakage or another large opening to the outside from the smoke zone, the pressure differences can decreases significantly. For this reason, mechanical exhaust alone does not constitute an adequate smoke control system when there is significant probability of window breakage or an opening from the smoke zone to the outside.

Mechanical Exhaust

- In the smoke zone, the location of the exhaust inlets is important. These inlets should be located away from exit stairs so that smoke in the vicinity of the shaft inlet does not pose an increased hazard during evacuation or fire fighting. Because hot smoke frequently stratifies near the ceiling, it is recommended that smoke exhaust inlets be located in or near the ceiling.
- Exhausting air from the smoke zone results in air from the outside and from other zones being pulled into the smoke zone. This air flowing into the smoke zone can provide oxygen to the fire.
- Current designs are based on the assumption that the adverse effect of supplying oxygen at six air changes per hour is insignificant in comparison with the benefit of maintaining tenable conditions in zones away from the fire. Thus , six air changes is recommended as the upper limits for exhaust airflow.

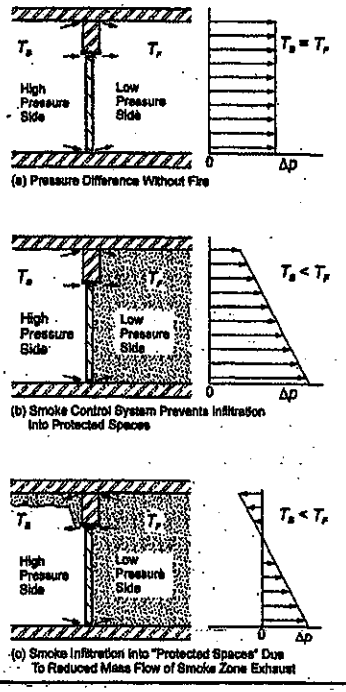
Mechanical Exhaust

Most commercial air-conditioning systems are capable of moving about four to six air change per hour, which probably accounts for the popularity of six air changes in smoke control applications. Current designs are based on the assumption that the adverse effect of supplying oxygen at six air changes per hours is insignificant in comparison with the benefit of maintaining tenable conditions in zones away from the fire. Thus, **six air changes is recommended as the upper limit for exhaust airflow.**

Minimum pressure difference required

- When the gases in the smoke zone are 'hot' , the buoyancy of the hot gases results in a non-uniform pressure difference.

Figure 4
Pressure difference at boundary of smoke zone



Use of MVAC System

- The smoke damper in the supply duct to the smoke zone is closed
- The smoke dampers in the return duct to non-smoke zone are closed
- If the system has a return air damper, it is closed.

Figure Schematic of zoned smoke control system using mvac system that serves many smoke control zones

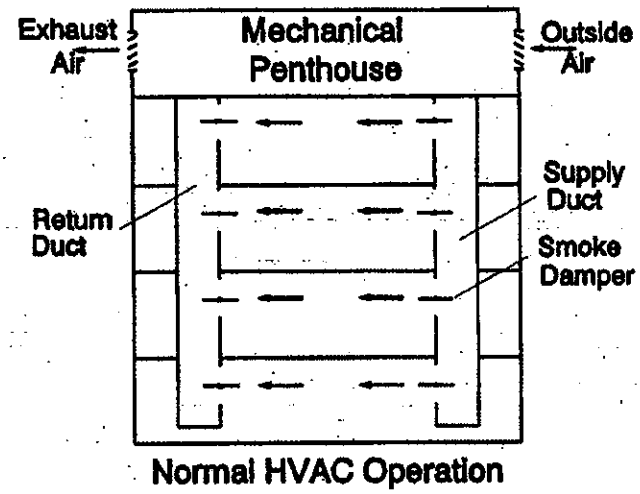
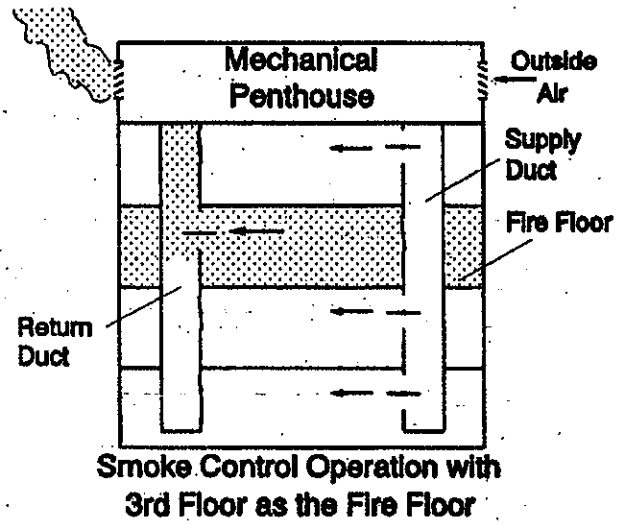


Figure 2 Schematic of zone smoke control system using an HVAC system that serves many smoke control zones



Smoke damper

- Smoke dampers are supplied in several leakage classifications. The particular class of damper specified should be selected based on the requirements of the application.
- For example, the dampers in the supply and return ducts shown in Figure 2 can have some leakage without adversely affecting smoke control system performance. Thus a designer might select class II, III or IV smoke dampers for such an application.

Class I dampers

- Further, a designer might choose class I dampers for applications that require a very tight damper (For example, the return damper illustrated in figure 3 and 4).

Figure 3 HVAC system with recirculation capability in the normal HVAC mode

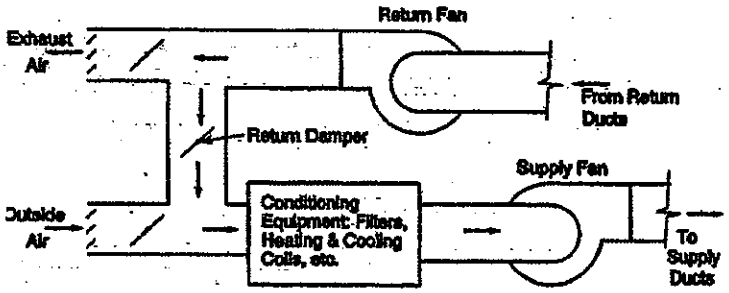


Figure 4 HVAC system with recirculation capability in the smoke control mode

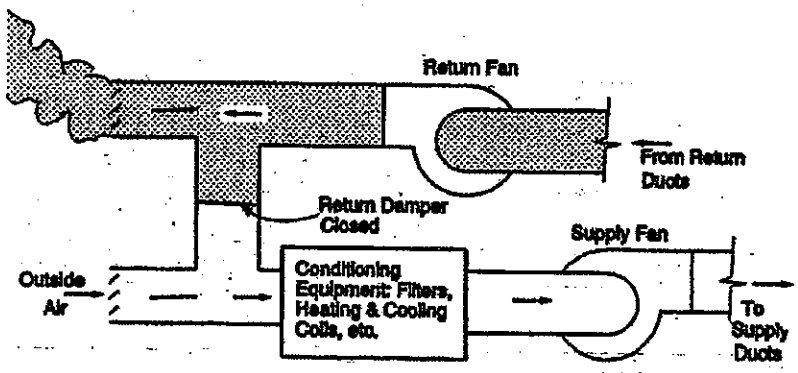
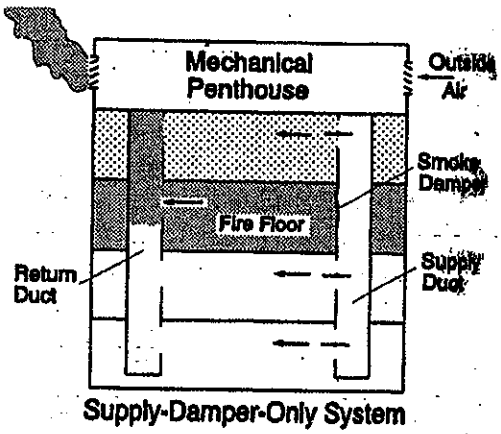


Figure 5 Schematic of failure to achieve smoke control by only shutting a smoke damper in the supply duct to the smoke zone

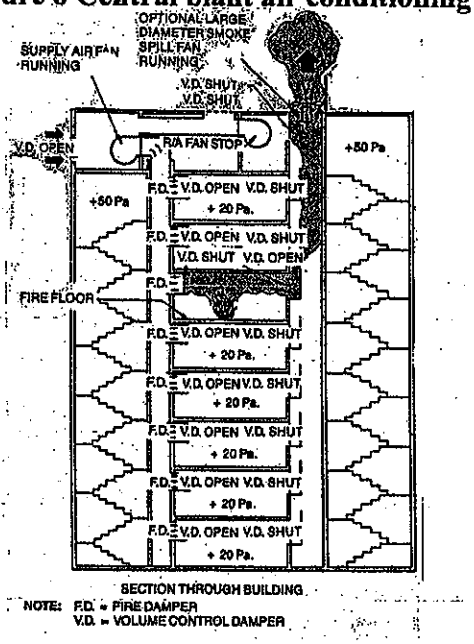


This system is not recommended because it generally Does not achieve satisfactory pressure differences to control Smoke movement

Example in Australia

- Sandwich pressurization system for smoke control, these systems have proven to be efficient and cost-effective in numerous multistory buildings in Australia.
- In Adelaide, there are now more than 25 buildings (ranging from 10 to 30 floors in height that have sandwich (zone) pressurization smoke control systems. Now optional within the new AS 1668, this method employs the building air-conditioning system to pressurize the non-fire floors, and to provide either relief or exhaust of the fie affected floor (See Figure 6 and 7).
-

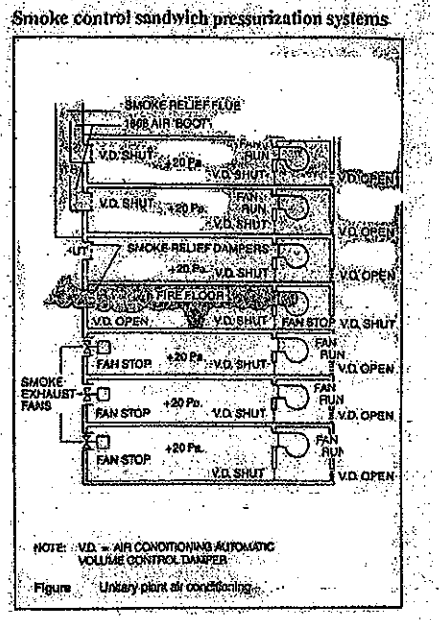
Figure 6 Central plant air conditioning



NFPA 92 A

- 3 phases of Fire zones functions
- 1. Smoke control – normal 'active' ambient –350 F, negative pressure , fan on
- 2. Containment – 450 F to 2000 F, 'Passive", Fan off
- 3. Smoke control --- under 150 F Purge (Air change)

Figure 7 Unitary plant air conditioning



The New Australian Standard 1668.1

- AS-1668.1 (Fire and Smoke Control) now specifies zone smoke control systems. These employ the same components as a traditional purging system except that additional plant systems. It is the control of these dampers, or the unitary plant, that achieves the necessary pressure differential between floors.
- Section 5, clause 5.8 of the new standard details requirements for central plant zone pressurization systems. These require volume control dampers within the supply air ductwork to each floor **that supply air to every floor except the fire floor**. The fire floor is either relieved or exhausted by a shaft to outside the building.

AS 1668.1

- Exhaust fans must provide six air changes per hour in the largest compartment or handle the volume entering the largest floor when all pressurized exits serving that floor are open, which ever is greater.
- In a traditional office building with two pressurized exits, a velocity of not less that 1 m/s (197 fpm) is required through each open door. At around 1.75 m^3/s (3,700 cfm) per door, this equates to a required exhaust of at least 3.5 m^3/s (7,400 cfm) per floor.

AS 1668.1

- To this air quantity, we must add infiltration from leakage paths such as lift doors, expansion due to temperature, curtain façade, and other factors. This will , in practice, probably make the at-floor exhaust not less than 4 m^3/s (8,475 cfm).
- Section 6 of the new standard details zone smoke control using on-floor unitary air handling systems. These must supply 100% fresh air to non-fire floors and stop on the fire floor, which is relieved to outside the building or exhausted by shaft or local fan. Under the Building Code of Australia, zone pressurization systems are mandatory in all buildings employing unitary plants.

AS 1668.1

- Central plant and on-floor system may be used to pressurize the stairs via leakage from the pressurized floor. In practice, this works well and has been utilized in South Australia for many years. However, it is important that stairwells have a relief path to atmosphere to prevent over-pressurization.
- It is intended that central plant zone pressurization systems used to pressurize stairs incorporate :

AS 1668.1

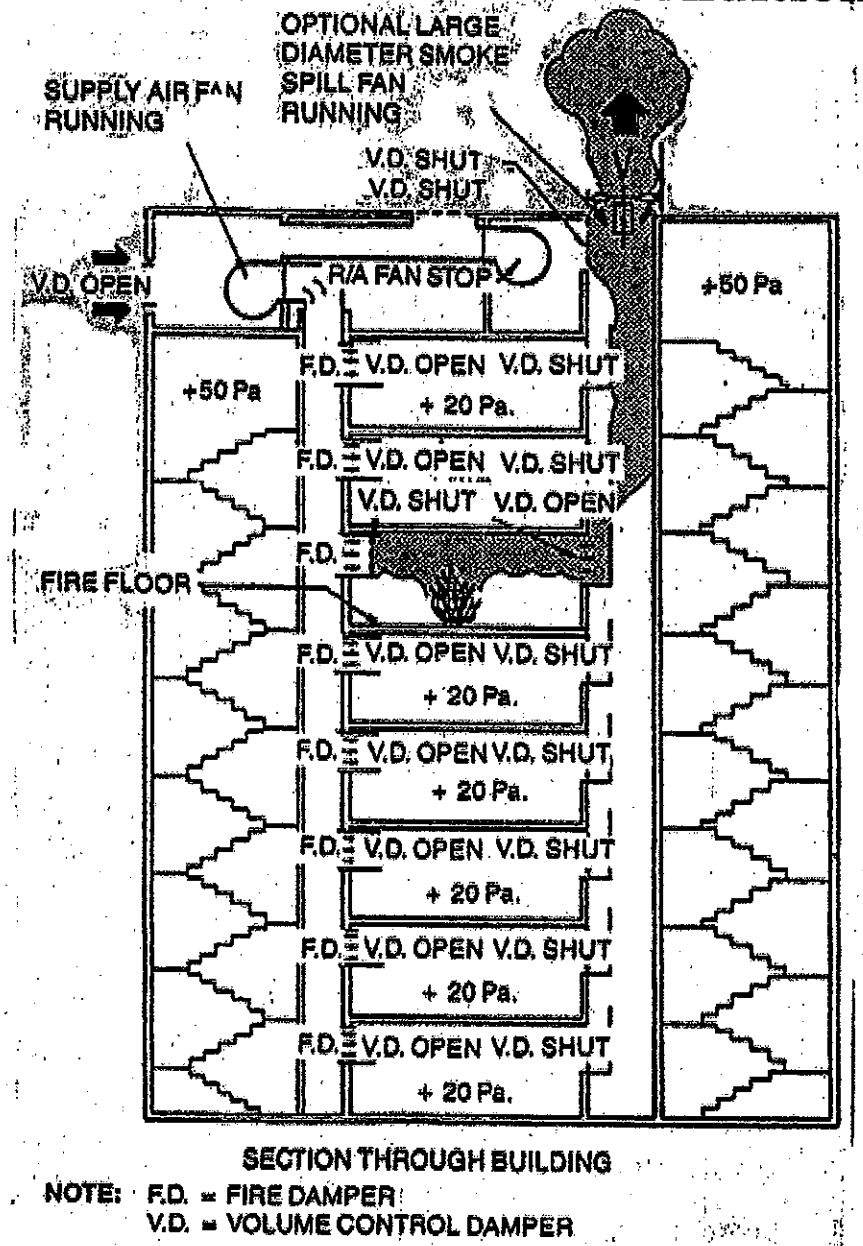
- **A standby supply air fan, or**
- **A standby drive arrangement, and**
- **Dual fresh air intakes on opposite sides of the building such that, if one is contaminated, it will shut down the other intake be utilized.**
- **The control of the dual air intakes must be arranged so that, if one intake becomes contaminated, enough time is available to permit it to close down and the fresh air plenum to clear, before the fan itself is stopped by the supply air smoke sensor.**

Fire safety in Twin Tower

- In the event of a fire, the smoke control system exhausts the floors in alarm, pressurizes all floors above and bellow the floor in question and pressurizes the exit stairs serving that particular section.
- This creates a negative pressure on the fire floor relative to other floors and any shaft serving that floor, thereby containing smoke to the level of origin and exhaust it form the building.
- It also provides conditioned air on all levels except the floor in alarm, the floor above and the floor below, thus allowing continued occupancy on on-affected levels.

- The fire safety plan anticipates that if a fire occurs on a floor served by the high-rise elevator banks, the shuttle elevators (providing an express service from the ground floor lobby to the sky lobby) will be available for use in evacuating occupants.
- As total evacuation is not practical and would be extreme disruptive to building function, the plan anticipates phased evacuation, including relocation of occupants on the fire floor and the floors above and below. Other floors would only be evacuated or occupants relocated if required subsequently.

Figure 6 Central plant air conditioning

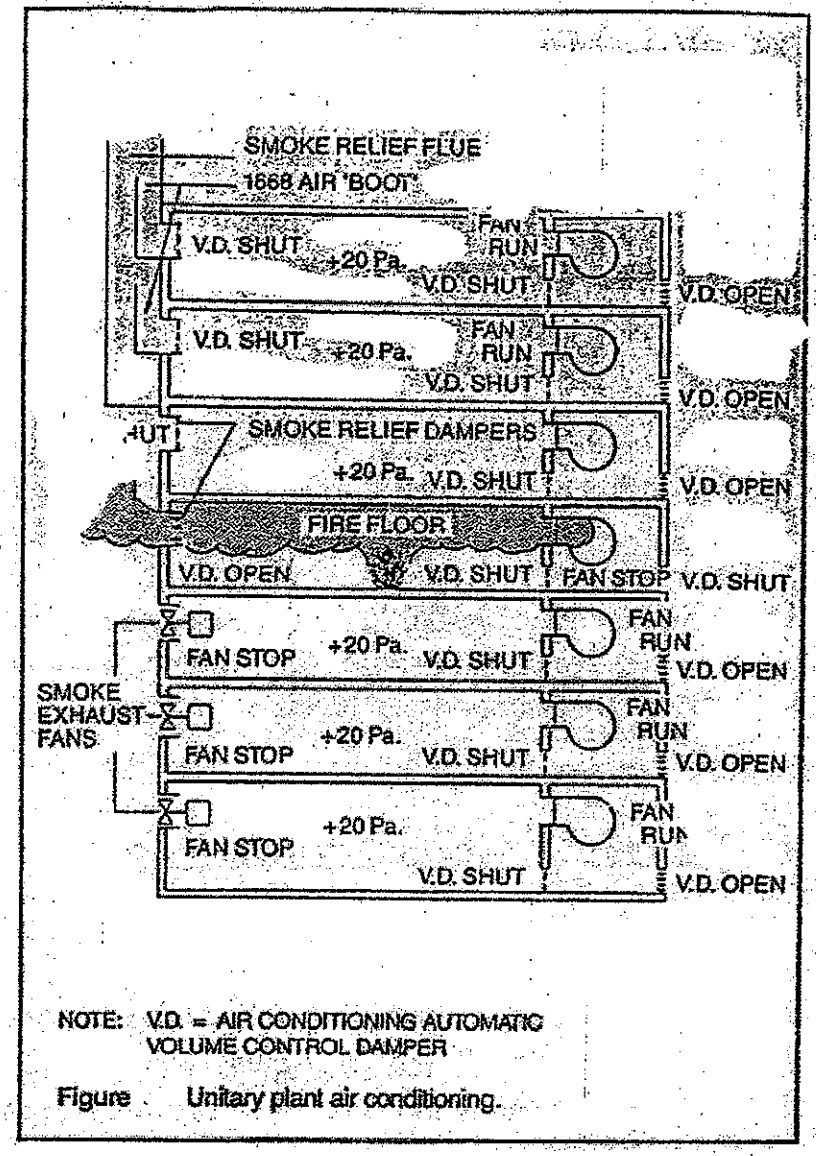


SECTION THROUGH BUILDING

NOTE: F.D. = FIRE DAMPER
V.D. = VOLUME CONTROL DAMPER

Figure 7 Unitary plant air conditioning

Smoke control sandwich pressurization systems

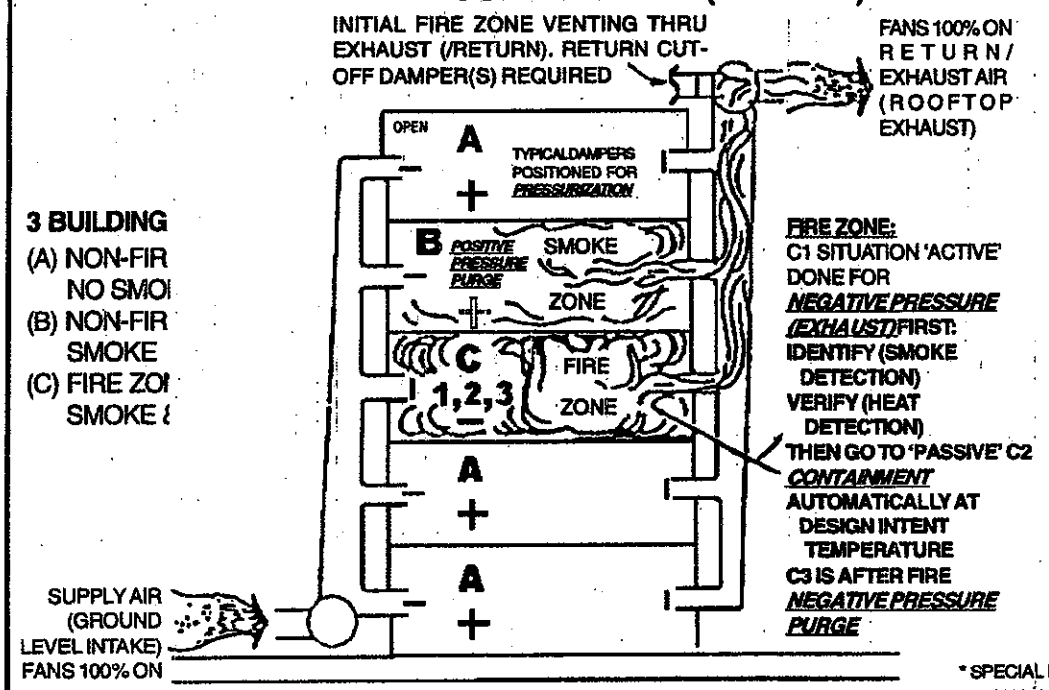


NOTE: V.D. = AIR CONDITIONING AUTOMATIC VOLUME CONTROL DAMPER

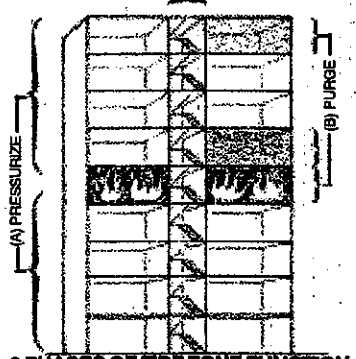
Figure . Unitary plant air conditioning.

FULL BUILDING SYSTEMS
SMOKE SYSTEMS FUNDAMENTALS SITUATIONS/OBJECTIVES

COMPARTMENT (NFPA92A) TYPE BUILDINGS



FIRE ZONE:
C1 SITUATION 'ACTIVE' DONE FOR ***NEGATIVE PRESSURE (EXHAUST)*** FIRST:
IDENTIFY (SMOKE DETECTION)
VERIFY (HEAT DETECTION)
THEN GO TO 'PASSIVE' C2 ***CONTAINMENT***
AUTOMATICALLY AT DESIGN INTENT TEMPERATURE
C3 IS AFTER FIRE ***NEGATIVE PRESSURE PURGE***

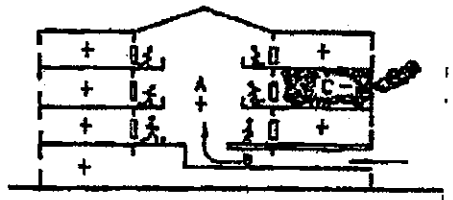


3 PHASES OF FIRE ZONE FUNCTION CONTROL OBJECTIVES
(C1) SMOKE CONTROL - NORMAL "ACTIVE" AMBIENT - 350°F NEGATIVE PRESSURE (FANS ON)
(C2)* CONTAINMENT - 450°F - 2000°F* "PASSIVE" (FANS OFF)
(C3) SMOKE CONTROL - UNDER 150°F PURGE (AIR CHANGES)

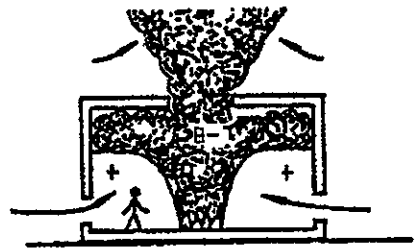
* SPECIAL NOTE: Smoke controls' success is dependent on sprinkler effectiveness in reducing fire zone pressure to .03 W.G. and limiting fire zone temperatures to 450°F – which would eliminate the C2 situation: ASHRAE RP677.

ATRIUM (NFPA92B) TYPE BUILDINGS

ATRIUM SITUATION - A+C



ATRIUM SITUATION - B



NOTES: The Basic Functions of a Smoke System (Pressurization, Exhaust, Containment & Purge):

1. If the intakes are rooftop mounted, there should be at least two intakes for the supply air side, and the two intakes are to be controlled and placed so as to differentiate the windward and leeward (smoke free vs. smoke filled) sides of the building;
2. Venting is not applicable in all designs (See NFPA 92B and 204M);
3. Fire-zone detector establishes pressurization;
4. If temperatures rise past fire link and/or actuator ratings, pressurization gives way to containment for the remainder of the equipment's fire-rated periods, except for designated long-term escape, safe refuge, or fire-fighting areas;
5. After the fire is under control, the building may be purged of smoke by opening all the return/exhaust dampers, along with all the supply dampers.

SPECIAL NOTES:

Fire fighters have great need to identify and heat-verify the fire zone before negative pressures are introduced in any smoke zone that would cause smoke migration out of the fire zone.

During the "active" fans-on period no fire zone return dampers should be allowed to close before the supply damper -- they should have synchronized operation to prevent spreading smoke by accidentally pressurizing the fire zone.

Chapter

Fundamental Concepts for Atria

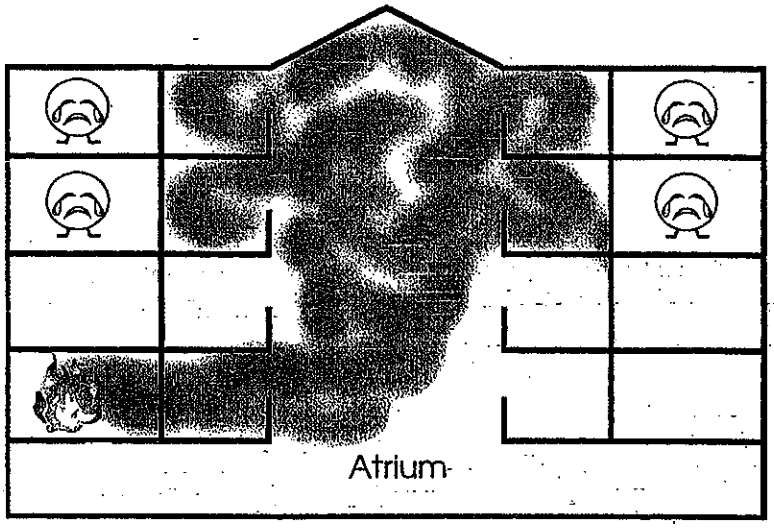
3 Apr 2008

Introduction

- This chapter present the fundamental concepts for atria smoke management, including smoke plume dynamics, plugholing, horizontal smoke flow , airflow to prevent smoke backflow, and empirical smoke filling equation

Smoke Control Systems

Smoke generated by a fire in an atrium of a building can prevent people from escaping to safety. An atrium is a central hall in a building.



If a smoke control system is installed, air from supply fans can keep the smoke away from the escape routes. Extract fans at the top of the atrium will remove the smoke from the building.

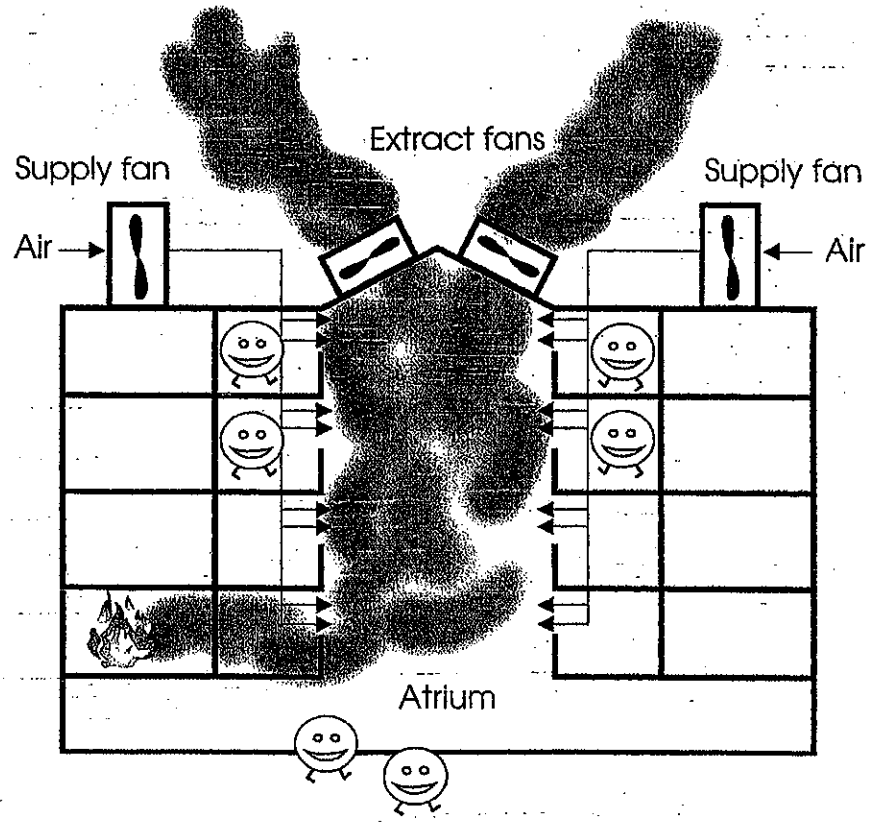


Figure 1a Axisymmetric smoke plume of an atrium fire

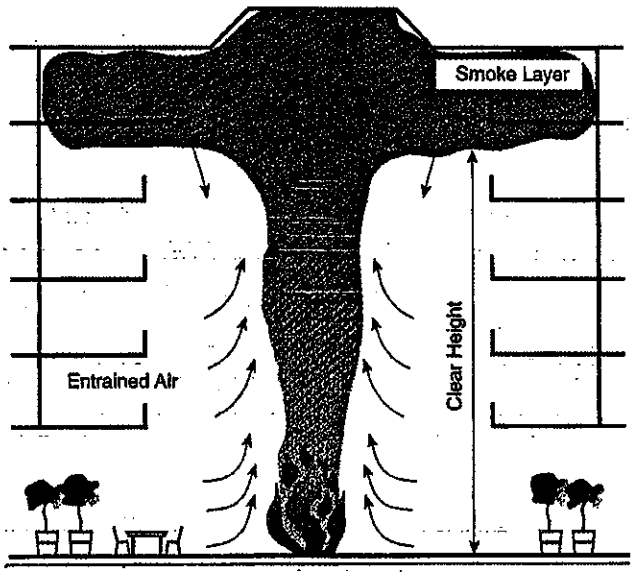
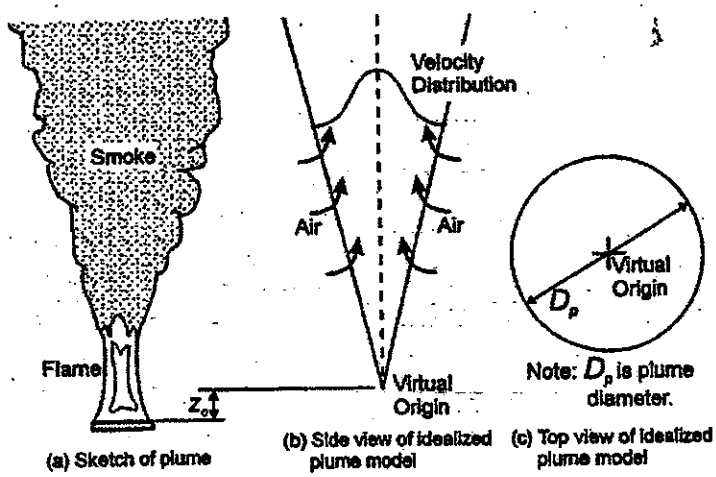


Figure 1 b Axisymmetric point source plume:(a) sketch (b) side view of idealized model ,and (c) top view of idealized model



Axisymmetric Plume

- The plume of figure 1 is called an axisymmetric plume.
- Because Heskestad's plume equations and his related work were the basis of many codes and standards in the US, these equations are used in the chapter.

Mass Flow with Virtual Origin Correction

Heskestad's equation for the mass flow of an axisymmetric plume is

$$\dot{m} = C_{a1}\dot{Q}_c^{1/3}(z - z_0)^{5/3}\left[1 + C_{a2}\dot{Q}_c^{2/3}(z - z_0)^{-5/3}\right]$$

for $z \geq z_1$, *and*

$$\dot{m} = \frac{C_{a3}\dot{Q}_c z}{C_{a4}\dot{Q}_C^{2/5} + z_0} \quad \text{for } z < z_1$$

$\dot{m}$ = *mass flow in axisymmetric plume at height, kg / s*

$\dot{Q}_c$ = *convective heat release rate of fire, kW*

z = *height above fuel, m*

z_0 = *virtual origin correction of the axisymmetric plume, m*

z_1 = *mean flame height, m*

$C_{a1} = 0.071$, $C_{a2} = 0.026$

$C_{a3} = 0.0054$, $C_{a4} = 0.166$

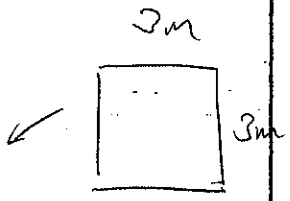
or $\dot{m}_s = 0.188 A Y^{3/2}$

Condition for axisymmetric plume

- The fire is circular, and many experimental studies have used liquid pool fires in round pans.
- However, a fire need not be round for the axisymmetric plume equation to be useful.
- The effective fire diameter can be expressed as :

$$D_f = 2\sqrt{A/\pi}$$

$$A = Area\ of\ the\ fire$$



Virtual Origin

- Heskestad's (1983) relationship for the virtual origin is

$$z_0 = 0.083\dot{Q}^{2/5} - 1.02D_f$$

$$\dot{Q} = Heat\ release\ of\ the\ fire, kW$$

$$D_f = Diameter\ of\ fire, m$$

Convective Heat

- In figure 1, the virtual origin is shown above the top of the fuel, but it can also be below the fuel.
- The sign convention is : for the virtual origin above the top of the fuel, z_0 is positive,and for virtual origin below the top of the fuel, z_0 is negative.
- The convective portion of he heat release rate, Q_C , can be expressed as :

$$\dot{Q} = \chi_C \dot{Q}$$

where

χ_c *is the convective fraction of heat release*

χ_c *of* 0.7 *is common for most design applications*

Flame Height

- The flame height depends on the fire geometry, the ambient conditions, the heat of combustion ,and the stoichiometric ratio.
- A relationship (Heskestad 2002) for flame height that can be used for many fuels is

$$z_1 = 0.235\dot{Q}^{2/5} - 1.02 D_f$$

$$z_1 = \textit{mean flame height}, m$$

Flame Height

- This flame height is the elevation where the maximum plume temperature is 500 K. The ceiling heights of atria are relatively high and it is the nature of atria smoke management that the elevation, z, of interest are much greater than either virtual origin, Z_0, or the flame height, z_1.

Centreline Plume Temperature

- The temperature varies over the plume corss section, and the plume temperature is greatest at the centreline of the plume, as shown in figure 2.
- The centerline temperature is of interest in the unusual cases when atria are tested by real fire.
- The centerline temperature equation (Heskestad 1986) is :

$$T_{cp} = T_0 + 9.1\left(\frac{T_a}{gC_p^2\rho_a^2}\right)^{1/3}\frac{\dot{Q}^{2/3}}{(z-z_0)^{5/3}}$$

T_{CP} = *absolute centerline axisymmetric plume temperature at elevation z, K*

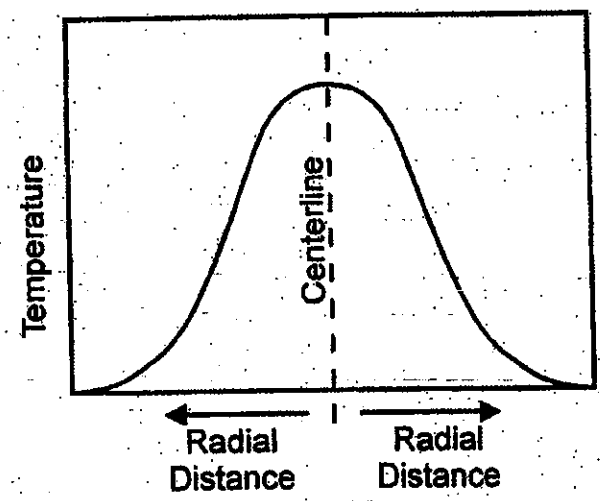
Centreline Plume Tempeature

For the conditions of $294K$, ρ *of* $1.2\ kg/m^3$ *and* C_p *of* $1.00kJ/kg^oC$

$$T_{cp} = T_0 + 25\frac{\dot{Q}^{2/3}}{(z-z_0)^{5/3}}$$

$T_0 =$ *ambient temperature*

Figure 2 Radial temperature variation of axisymmetric plume



Plume Diameter

- The following equation is a way of expressing the expected range of diameter of an axisymmetric plume :

$$D_P = \frac{z}{C_{PD}} \text{-----(9)}$$

D_P = *diameter of visible axisymmetric plume, m*

z = *height above fuel, m*

C_{PD} = *plume diameter coefficient ranging from 2 to 4*

FRS calculation for axisymmetric Plume

$$m_{smoke} = 0.18 P Y^{3/2} kg/s$$

$P = 12\ m$ perimeter

$Y = 20\ m$

$$m_{smoke} = 0.18 x 12 x 20^{3/2} = 193\ kg/s$$

Axisymmetric Plume Without z_0

Heskestad's equation for the mass flow of an axisymmetric plume is

$$\dot{m} = 0.071 \dot{Q}_c^{1/3} z^{5/3} + 0.0018 \dot{Q}_C \quad ---(10)$$

for $z \geq z_1$, *and*

$$\dot{m} = 0.032 \dot{Q}_C^{3/5} z \text{ for } z < z_1 \quad ----(11)$$

$\dot{m}$ = *mass flow in axisymmetric plume at height, kg / s*

$\dot{Q}_c$ = *convective heat release rate of fire, kW*

z = *height above fuel, m*

z_0 = *virtual origin correction of the axisymmetric plume, m*

z_1 = *mean flame height, m*

Axisymmetric Plume

- The plume of figure 1 is called an axisymmetric plume.
- Because Heskestad's plume equations and his related work were the basis of many codes and standards in the US, these equations are used in the chapter.

Wall and Corner Plumes

- A fire that is located next to a wall will entrain air over a smaller perimeter than a fire located far from a wall.
- The fire and wall plume may be considered half that of the axisymmetric plume.
- Thus the mass flow rate of a wall plume is half that of an axisymmetric plume, but the analogous fire for the axisymmetric plume is twice that of the "real" fire creating the wall plume.

Wall and Corner Plumes

- The mass flow rate of a wall plume can be estimated as follows:
- 1. Let Q be twice that of the wall plume
- 2. Apply a plume equation to calculate the mass flow m, of an axisymmetric plume
- 3. The smoke production of the wall plume is m/2.

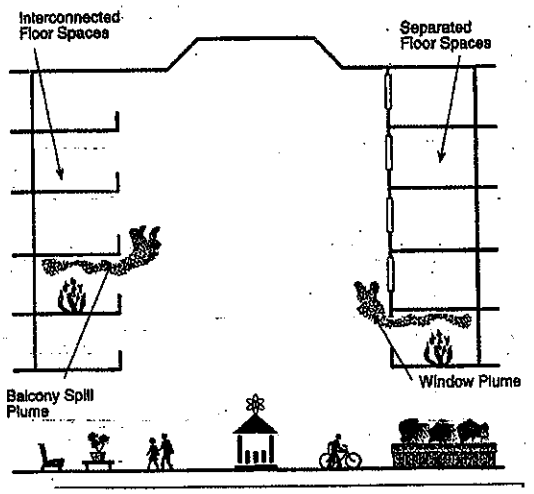
For Corner Plume

$$Q_C = 4\,x\,Q = 20{,}000\,kW$$

$$m = 0.071(0.7\,x\,20{,}000)^{1/3}\,20^{5/3} + 0.0018(0.7x20{,}000)$$
$$= 288.4\,kg/s$$

$$therefore,\ mass\ flow\ rate = \frac{288.4}{4} = 72.1\,kg/s$$

Figure 3 Window and balcony spill plumes



Example 1 Simple Plume Calculate

- For a 4220 kW fire , what is the mass flow and centreline temperature of the plume at z of 10.7 m above the fuel? Use a convective fraction of 0.7 , and the ambient temperature is 22.2°C.

Example 1, cont.

$\dot{Q}_C = \chi_C \dot{Q} = 0.7(4220) = 2950\ kW$

The mean flame height is

$z_1 = 0.166 \dot{Q}_C^{2/5} = 0.166(4220)^{2/5} = 3.9\ m$

Because $z_1 < z$, *the plume mass flow is*

$m = 0.071 \dot{Q}_c^{1/3} z^{5/3} + 0.0018 \dot{Q}_C$

$= 0.071(2950)^{1/3} 10.7^{5/3} + 0.0018(2950)$

$= 58.2\ kg/s$

The centreline temperature is

$T_{cp} = T_0 + 25 \frac{\dot{Q}_C^{2/3}}{z^{5/3}} = 22.2 + 25 \frac{2950^{2/3}}{10.7^{5/3}}$

$= 122°C$

Flame temperature not smoke temperature

153

From US
Design of Smoke Management Systems
By ASHRAE and Society of Fire Protection Engineers

- **Axisymmetric Plume;Fire size : 5 MW**

$$\text{Limiting Height } Z_f = 0.166 E_c^{2/5}$$

$$= 0.166(0.7 \times 5000)^{2/5} = 4.34m < 20\,m$$

$$m = 0.071 E_c^{1/3} Z^{5/3} + 0.0018 E_c$$

$$= 0.071(0.7 \times 5000)^{1/3} 20^{5/3} + 0.0018 \times 0.7 \times 5000$$

$$= 165.2\ kg/s$$

$$Q = \frac{m}{\rho} = \frac{165.2}{1.2} = 137.6\ m^3/s$$

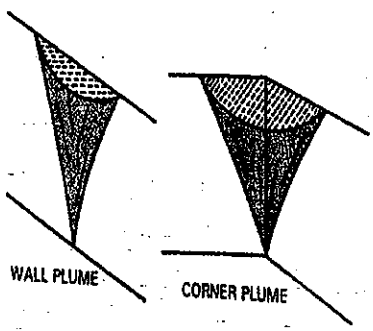
$$ACH = \frac{137.6 \times 3600}{35 \times 35 \times 20} = 20.2$$

Wall and Corner Production

- For wall plume
- E = 5MW
- Z = 20 M

$$\dot{m} = 0.071 Q_C^{1/3} Z^{5/3} + 0.0018 Q_C$$

$$Q_C = 2 \times Q = 10{,}000\ kW$$



$$m = 0.071(0.7 \times 10{,}000)^{1/3} 20^{5/3} + 0.0018(0.7 \times 10{,}000)$$

$$= 218\ kg/s$$

$$\text{therefore, mass flow rate} = \frac{218}{2} = 109\ kg/s$$

Figure 4 Configurations of fire

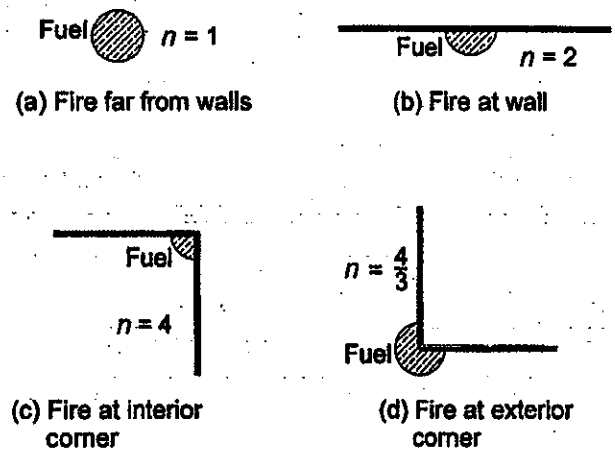
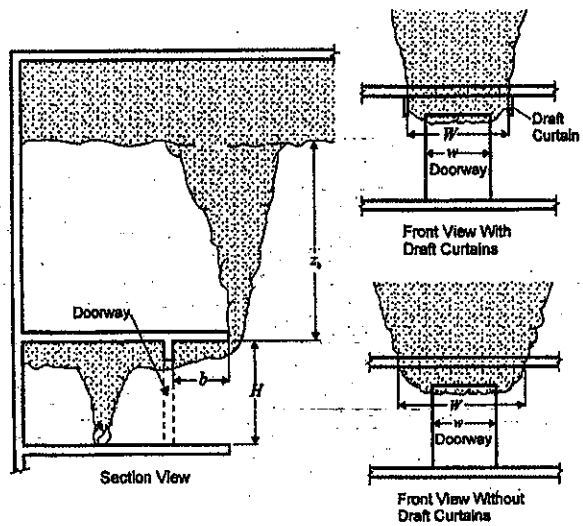


Figure 5 Balcony Spill Plume



Zb can be controlled by smoke curtain

W = 14 m (civil engineering handbook)

Balcony Spill Plumes

- A balcony spill plume originates from a fire when the smoke flows under a balcony and spills into the atrium (figure 5).
- When the fire is in a room that opens onto the balcony, the mass flow rate can be approximated as (Law 1986; CIBSE 1995):

Balcony Spill Plumes

$$m = 0.36\left(\dot{Q}W^2\right)^{1/3}(z_b + 0.25H)$$

m = *mass flow in plume height* z_b, *kg / s*

$\dot{Q}$ = *Heat release of the fire, kW*

W = *plume width as it spills under balcony, m*

z_b = *height above balcony, m*

H = *height of balcony above top of fuel, m*

When draft curtains are used, the width, W, of the spill plume is the distance between the curtains. In the absence of draft curtains, the following approximation can be used.

$W = w + b$

w = *width of the opening from the fire room, m*

b = *distance from the opening to the balcony edge, m*

Balcony Spill Plume

- The above two equations only apply when the height of the opening to the fire room is sufficiently below the ceiling such that the momentum of the ceiling jet in the fire room does not directly contribute to the flow out of the opening.
- The thickness of the ceiling jet is in the range of 10% to 20% of the height from the base of the fire to the ceiling.

Example 2 Balcony Spill Plume

- What is the mass flow of a balcony spill plume with the parameters listed below ?

Heat release rate, $\dot{Q} = 528\ kW$
Height above balcony $= 6.10\ m$
Height of balcony above top of fuel, $H = 3.05m$
Width of the opening from the fire room, $w = 1.83\ m$
Distance from opening to the balcony edge $= 3.66\ m$

Example Balcony Spill Plume

The plume width as it spills under the balcony is approximated as

$W = w + b = 1.83 + 3.66 = 5.49m$

The mass flow of the spill plume is

$$\dot{m} = 0.36\left(\dot{Q}W^2\right)^{1/3}\left(z_b + 0.25H\right)$$

$$= 0.36\left(528 x 5.49^2\right)^{1/3}\left(6.10 + 0.25 x 3.05\right)$$

$$= 66.7\ kg/s$$

Window Plume

$a = 2.4 A_W^{2/5} H_W^{1/5} - 2.1 H_W$

A_W = *Area of ventilation opening*

$$\dot{m} = 0.071 Q_C^{1/3}\left(Z_W + a\right)^{5/3} + 0.0018 Q_C$$

If window = 3 *ft by* 5 *ft* $(0.91m x 1.52m)$

$$Q_C = 1260\, A_W H_W^{1/2}$$

$$= 1260(0.91x1.52)(1.52) = 2649\ kW$$

$$a = 2.4[0.91 x 1.52]^{2/5}[1.52]^{1/5} - 2.1 x 1.52$$

$$= 2.97 - 3.192 = -0.222\ m$$

$$\dot{m} = 0.071[2649]^{1/3}[30.5 - 0.222]^{5/3} + 0.0018\, x 2649$$

$$= 293.3\ kg/s$$

Window Plume

$$a = 2.4 A_W^{2/5} H_W^{1/5} - 2.1 H_W$$

A_W = *Area of ventilation opening*

$$\dot{m} = 0.071 Q_C^{1/3} (Z_W + a)^{5/3} + 0.0018 Q_C$$

If window = 3 *ft by* 5 *ft* $(0.91m \times 1.52m)$

$$Q_C = 1260 A_W H_W^{1/2}$$

$$= 1260(0.91x1.52)(1.52) = 2149\,kW$$

$$a = 2.4[0.91 \times 1.52]^{2/5}[1.52]^{1/5} - 2.1x1.52$$

$$= 2.97 - 3.192 = -0.222\,m$$

$$\dot{m} = 0.071[2149]^{1/3}[3.05 - 0.222]^{5/3} + 0.0018\,x2149$$

$$= [5.16 + 3.87]\,kg/s$$

$$= 9.04\,kg/s$$

Window Plumes

- Accordingly, the size of the fire depends on the size and shape of the opening to the room and the material burning.

$$\dot{m} = 0.68\left[\left(A_w H_w^{1/2}\right)^{1/3}\left(z_w + a\right)^{5/3} + 1.59 A_w H_w^{1/2}\right]$$

A_w = *Area of ventilation opening*, m^2

H_w = *Height of ventilation opening*, m

$$= 0.68\left[\begin{array}{l}\left((0.91x1.52)x1.52^{1/2}\right)^{1/3}\left(3.05 - 0.222\right)^{5/3} \\ +1.59x(0.91x1.52)x1.52^{1/2}\end{array}\right]$$

$$= 6.54 + 2.71 = 9.25\,kg/s$$

Window Plume

- A window plume is one that flows through an opening such as a window or door to a room with a fully involved fire.
- The high intensity of such a fire explains why window plumes are not normally considered design fires in sprinklered buildings.
- In such a fire, fuel volatilized in the room would burn outside the opening.
- The heat release rate of a fully developed fire is constrained by the combustion air that can reach the fire, and such a fire is referred to as being ventilation controlled.

Window Plumes

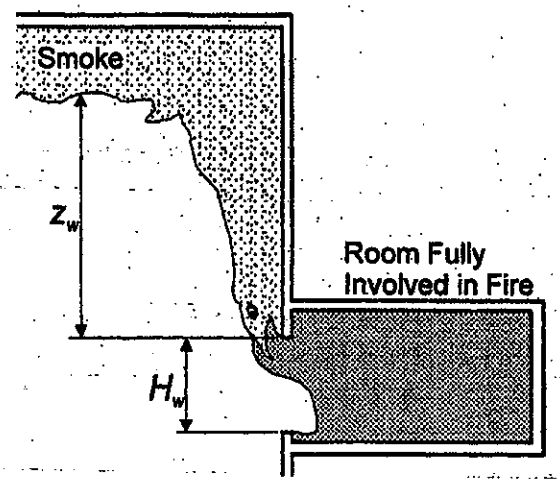
- Accordingly, the size of the fire depends on the size and shape of the opening to the room and the material burning.

$$m = 0.68\left[\left(A_w H_w^{1/2}\right)^{1/3}\left(z_w + a\right)^{5/3} + 1.59 A_w H_w^{1/2}\right]$$

$A_w = Area\ of\ ventilation\ opening, m^2$

$H_w = Height\ of\ ventilation\ opening, m$

Figure 6 Window plume



Entrainment into Spill Plumes rising through the atrium

- One method of calculating the entrainment into the line plume is due to Thomas:
- The relevant formula is :

$$m_{smoke} = 0.58\rho\left[\frac{gQ_W L^2}{\rho C T_0}\right]^{1/3}(h_b+\Delta)\left[1+\frac{0.22(h_b+2\Delta)}{L}\right]^{2/3}$$

m_{smoke} = *mass flow rate of smoke entering the smoke layer at height* h_b, *kg / s*

Q_W = *Convective heat flux, kW*

L = *Length of void edge past which gases spill, m* (Spill = max 14m)

C = *specific heat of air*

Δ = *Empirial height of virtual source below void edge, m*

h_b = *Height of rise of thermal plume above void edge, m*

For practical purpose it can be taken that

$\Delta = 0.3$ *times the height of the compartment opening,* H

i.e. $\Delta = 0.3H$

Average Plume Temperature

$$T_p = T_0 + \frac{\dot{Q}_c}{\dot{m} C_P}$$

T_p = *average plume temperature at elevation z,* ^oC

T_0 = *ambient temperature,* ^oC

C_P = *specific heat of plume gases,* kJ / kg^oC

Example 3 Average Plume Temperature

- What is the average temperature of the Plume in the example 1?

$$T_p = T_0 + \frac{\dot{Q}_c}{\dot{m} C_p}$$

$$= 22.2 + \frac{2950}{58.2x1} = 73^oC$$

Maximum Plume Height

- The plume mass flow equations were developed for strongly buoyant plumes.
- When smoke is not hot enough to rise, it will stagnate or be carried away by existing air currents.

$$z_{max} = \left[\dot{Q}_C^{2/3} \left(\frac{14}{\Delta T_{max}} - 0.0254 \right) \right]^{3/5}$$

z_{max} = max*imum height at which plume is considered buoyant, m*

$\dot{Q}_C$ = *Convective heat release rate, kW*

ΔT_{max} = min *imum temperature rise of plume above ambient, °C*

Stone: Height
Max 20 m.

Maximum Plume Height

- No research has been conducted to determine the appropriate value of the minimum temperature rise for various applications and velocities of air currents. A 2 degree C is suggested.
- For a total heat release rate of 530 KW , the maximum plume height is about 40 m (from figure 9).

Volumetric Flow

The volumetric flow of a plume is

$$\dot{V} = \frac{\dot{m}}{\rho_p} \quad (m^3/s)$$

ρ_p = *density of plume gases at elevation*, $z, kg/m^3$

Air and Plume Density

The density of air and plume gases is calculated from the perfect gas law:

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

P = *Absolute pressure*, Pa

R = *Gas constant*, J/kgK

T = *Absolute temperature*, K

$P = \rho RT$

Natural Venting

- Natural vents consist of openings in the ceiling through which smoke flows due to buoyancy.
- The hot smoke layer under the ceiling acts to force smoke out of the vent and to pull makeup air through other openings.

Natural Venting

$$\dot{m}_v = \frac{CA_v\rho_o\left[2gd_p(T_S - T_0)\left(T_0/T_S\right)\right]^{1/2}}{\left[T_S + \left(A_v/A_i\right)^2 T_0\right]^{1/2}}$$

$\frac{A_v}{A_i} \geq 2$ (Very efficient, ideal!)

$\dot{m}_v$ = *mass flow rate through the vent, kg / s*

C = *discharge coefficient*

A_v = *vent area, m^2*

A_i = *inlet opening, m^2*

ρ_0 = *outside air density, kg / m^3*

d_p = *depth of smoke layer below the smoke vent, m*

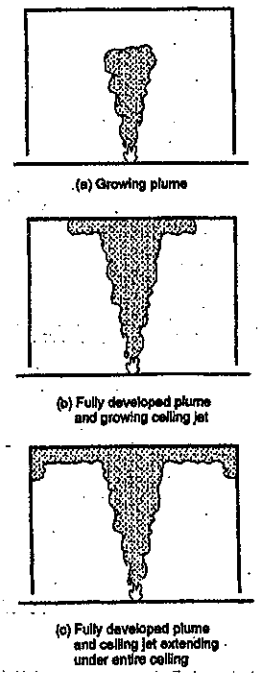
T_0 = *absolute temperature of outside air, K*

T_s = *absolute temperature of smoke, K*

Figure 7

Transport Time Lag

- A plume takes time to rise to the ceiling, and a ceiling jet takes time to form a smoke layer under the ceiling.



Ceiling Jet Lag

- Newman and Mowrer also developed the following relationships for the time lag of ceiling jets resulting from steady and unsteady fires.

For a steady fire

$$t_{cj} = \frac{0.833 r^{11/6}}{\dot{Q}^{1/3} H^{1/2}}$$

For a t-square fire

$$t_{cj} = \frac{0.72 r t_g^{2/5}}{H^{1/5}}$$

where

t_{cj} = *transport time lag of ceiling jet*

r = *radius or horizontal distance from centreline of plume, m*

Plume Lag

- Newman (1988) and Mowrer (1990) developed relationships for the time lag of plumes from steady and unsteady fires.
- Because it is the nature of fires to grow, the time lag time for steady fires is probably less significant.

For a steady fire

$$t_{pl} = 0.67\frac{H^{4/3}}{Q^{1/3}}$$

For a t – square fire

$$t_{pl} = 0.1H^{4/5}t_g^{2/5}$$

t_{pl} = *transport time lag of plume, s*

t_g = *growth time, s*

H = *Height of ceiling above top of fuel, m*

$\dot{Q}$ = *Heat release rate, kW*

Total Transport Lag Time

- The total transport lag time is the sum of the plume lag time and the ceiling jet lag time,

$$t_t = t_{pl} + t_{cj}$$

t_t *is the total transport lag time, S*

Figure 8 Control volume for idealized plume

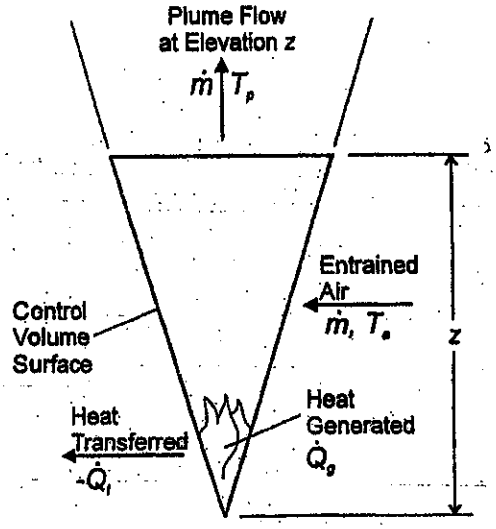


Figure 9 Maximum plume height

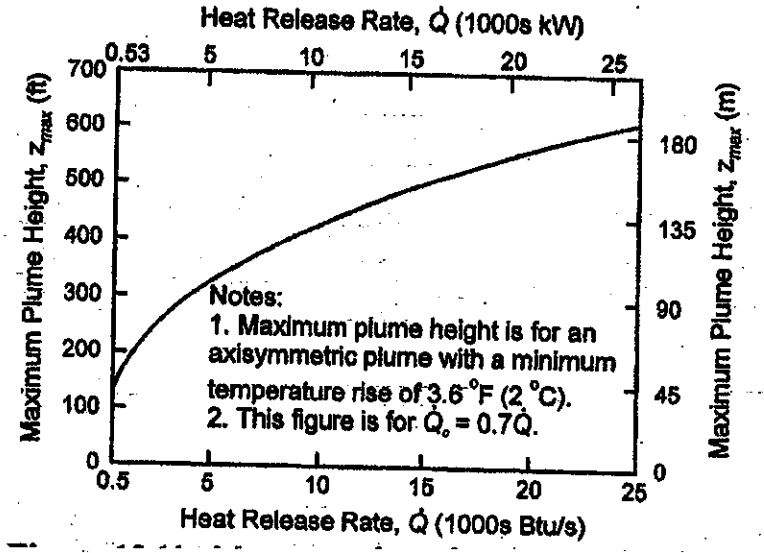


Figure 10 Gravity smoke venting

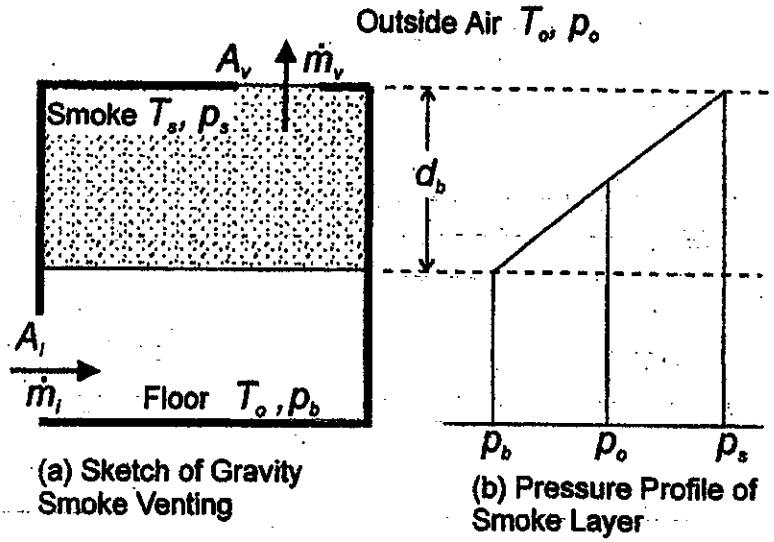


Figure 11 Plume transport lag

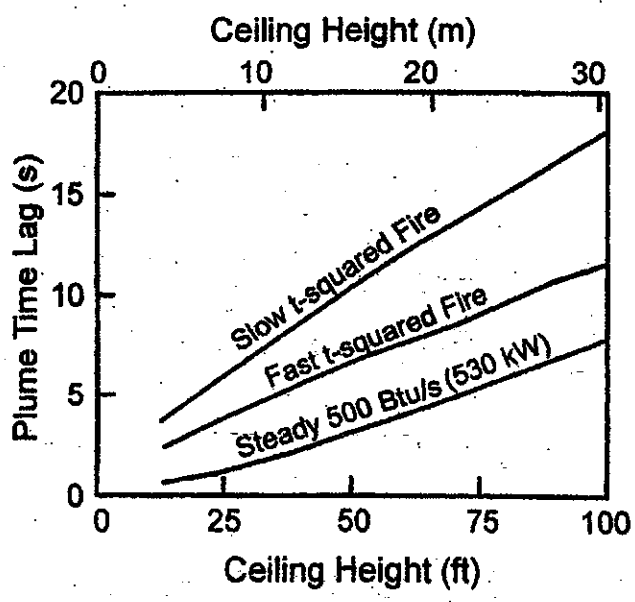


Figure 12
Development of minimum smoke layer in an enclosed room or atria

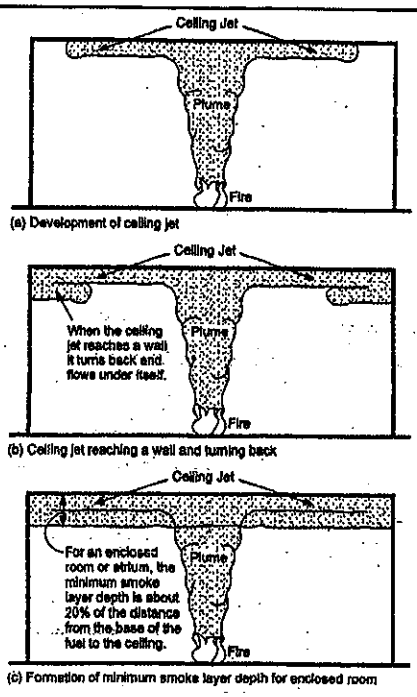


Figure 13 Smoke layer in a room or atria with an opening at the ceiling

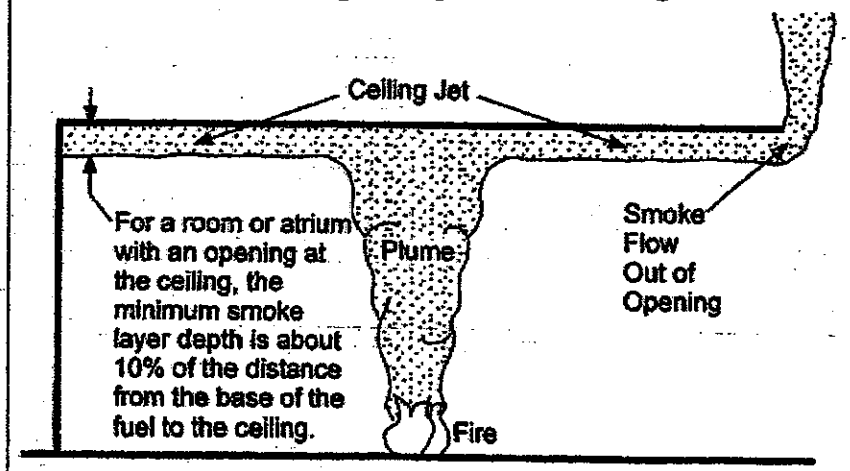


Figure 14 horizontal flow thru an opening or vent

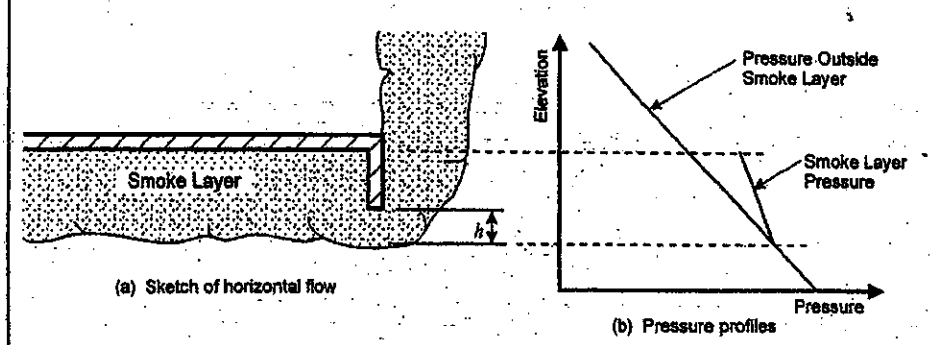
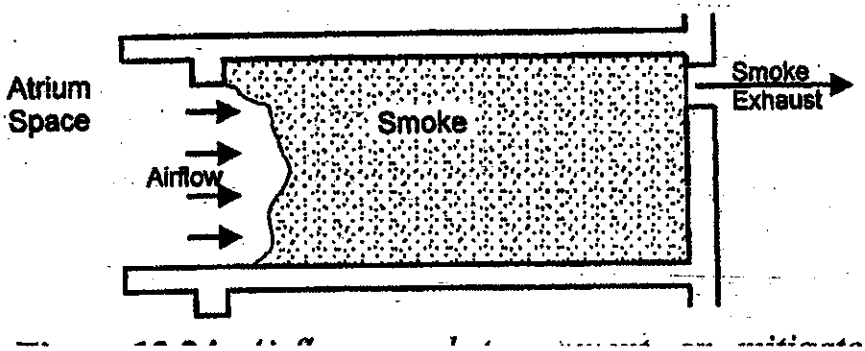
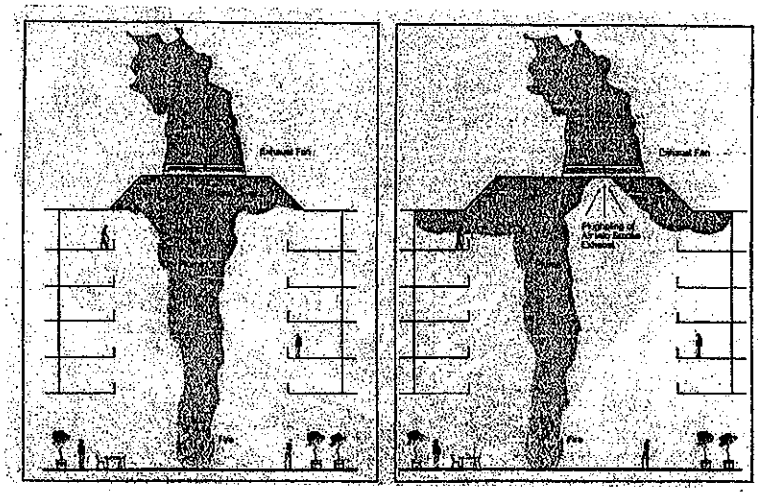


Figure 15 Airflow used to prevent or mitigate smoke originating in a communicating space from migrating to the atrium



Plugholing

- **When the smoke layer depth below an exhaust inlet is relatively shallow, a high exhaust rate can lead to entrainment of cold air from the clear layer. This phenomenon is called <u>plugholing.</u>**



Number of Exhaust Inlets

- To prevent plugholing, more than one exhaust point may be needed.
- The maximum mass flow rate than can be efficiently extracted using a single exhaust inlet is given by [CIBSE 1995].

Number of Exhaust Inlets

$$m_{max} = C_{ph1}\beta d^{5/2}\left(\frac{T_S - T_O}{T_S}\right)^{1/2}\left(\frac{T_O}{T_S}\right)^{1/2}$$

where

m_{max} = max*imum mass rate of exhaust without plugholing, kg / s*

T_S = *absolute temperature of the smoke layer, K*

T_O = *absolute ambient temperature, K*

d = *depth of smoke layer below bottom of exhaust inlet, m*

β = *exhaust location factor* (dim*ensionless*)

$C_{ph1} = 3.13$

Maximum Volume flow rate to prevent Plugholing

$$\dot{V}_{max} = 0.0087\,\beta\, d^{5/2}\sqrt{T_0(T_S - T_0)}$$

$\dot{V}_{max}$ = max*imum volumetric flow rate at* $T_s, m^3 / s$

Number of inlets

$$N_{inlets} = ROUND\left(\frac{\dot{V}}{\dot{V}_{\max}}\right)$$

Number of extract vents (N)

- The number of extract Vents is given by:
- $N > m_{SMOKE}/m_{max}$

$\frac{M_{smoke}}{M_{max}}$ = No. of openings

plugholing

- In the context of plugholing, the smoke layer depth is always a distance from the smoke interface to the bottom of the exhaust inlet.
- For an exhaust inlet located in a wall, the depth of the smoke layer below the bottom of exhaust inlet is illustrated in the figure.

Figure 16 Depth, d, of smoke layer below bottom of exhaust inlet

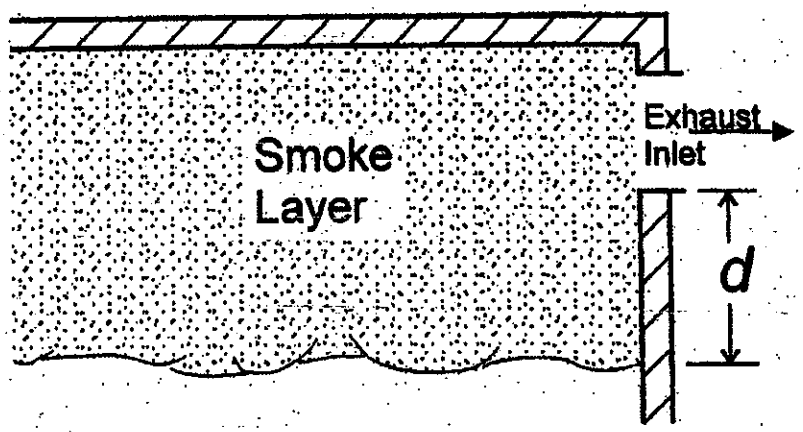


Figure 17 Plugholing of air into smoke exhaust inlets can result in failure of a smoke exhaust system

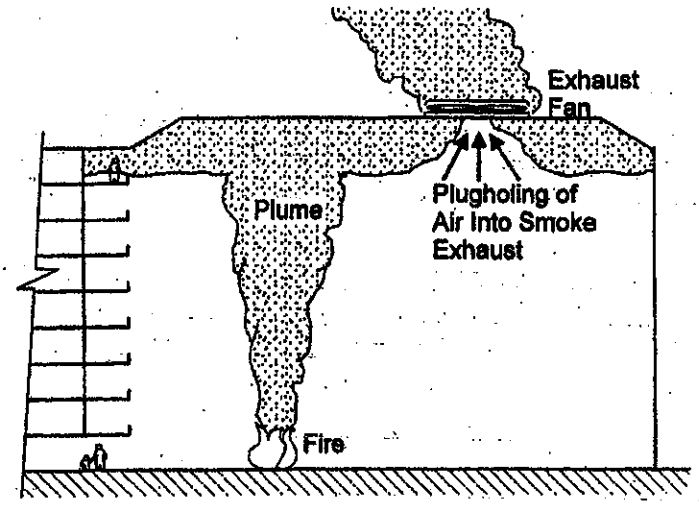
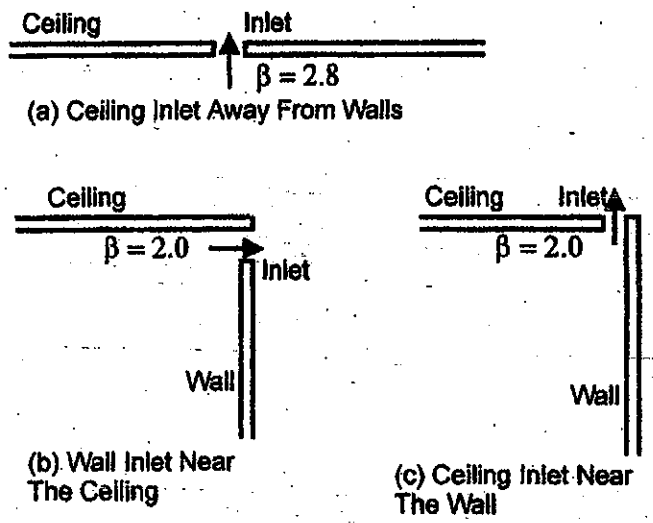


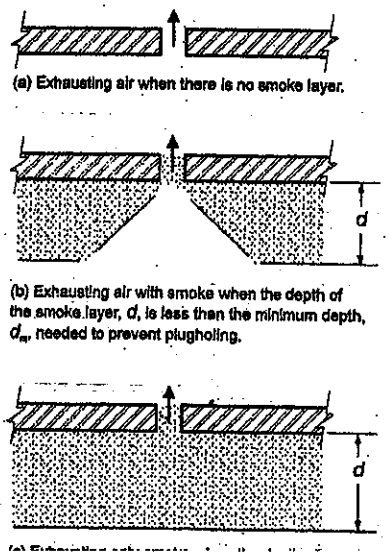
Figure 18 Values of exhaust location factor, β



β value

- 2. 0 for a ceiling exhaust inlet near a wall
- 2.0 for a wall exhaust inlet near the ceiling,
- 2.8 for a ceiling exhaust inlet far from any walls.
- It is suggested that d/D be greater than 2. Where D is the diameter of the inlet. For exhaust inlets, use D = 2ab/(a+b), where a and b are the length and width of the inlet.

Figure 19 Smoke and air exhaust conditions



(a) Exhausting air when there is no smoke layer.

(b) Exhausting air with smoke when the depth of the smoke layer, d, is less than the minimum depth, d_m, needed to prevent plugholing.

Separation between inlets

- When the exhaust at an inlet is near this maximum flow rate, adequate separation between exhaust inlets needs to be maintained to minimize interaction between the flows near the inlets.
- One criterion for the separation between inlets is that it be at least the distance from a single inlet that would result in arbitrarily small velocity, the minimum separation distance for inlets located in a wall near the ceiling (or in the ceiling near the wall) is :

$$S_{\min} = C_{ph3}\beta\sqrt{\dot{V}_e}$$

$S_{\min} = \min imum\ separation\ between\ inlets, m$

$\dot{V}_e = volumetric\ flow\ rate, m^3/s$

$\beta = exhaust\ location\ factor$

$C_{ph3} = 0.32$

Minimum Depth to Prevent Plugholing

- The minimum smoke layer depth to prevent plugholing can be expressed as :

$$d_m = \frac{\dot{V}_e^{2/5}}{C_{ph4}\beta^{2/5}\left[T_O\left(T_S - T_O\right)\right]^{1/5}}$$

$d_m = \min imum\ smoke\ layer\ depth\ to\ prevent\ plugholing, m$

$C_{ph4} = 0.15$

Satisfying the above equation is equivalent to meeting The criterion for the number of exhaust inlets of the previous session.

Horizontal Smoke Flow

- Horizontal flow out of a smoke layer and the corresponding pressures are shown in figure 14.
- These pressures are hydrostatic, that is ,the dynamic pressure components are negligibly small.
- The smoke flow thru the opening depends on the buoyancy of the smoke and the thickness of the smoke layer under the first level:

Horizontal flow

$$\dot{m} = \frac{2}{3} C W h^{3/2} \sqrt{2 \rho_s g (\rho_a - \rho_s)}$$

$\dot{m} = mass\ flow\ from\ smoke\ layer\ through\ opening, kg/s$

$W = width\ of\ opening, m$

$h = depth\ of\ smoke\ layer, m$

$\rho_a = ambient\ air\ density, kg/m^3$

$\rho_s = smoke\ density$

$C = flow\ coefficient, 0.7$

Horizontal Flow

Substituting the ideal gas equation int*o the eqn*

$$\dot{m} = 730 W h^{3/2} \frac{1}{T_S} \sqrt{\frac{T_S}{T_0} - 1}$$

$T_S = smoke\ temperature, K$

$T_0 = ambient\ air\ temperature, K$

Communicating Spaces

- Communicating spaces are spaces within a building that have an open pathway to an atrium such that smoke from a fire in the communicating space can move unimpeded into the atrium.
- Smoke from a fire in an atrium can also move unimpeded into the communicating space.
- Communicating spaces can open directly to the atrium or can be connected through other open spaces.

Airflow for fire in communicating space

- Airflow can be used to prevent or mitigate smoke originating in a communicating space from migrating to the atrium.
- This can be accomplished by exhausting the communicating space and supplying air to the atrium such that the velocity at the opening of the communicating space is sufficient to prevent smoke from flowing to the atrium.
- To prevent such smoke migration, the average velocity in the opening must equal or exceed the limiting velocity (Heskestad 1989)

Airflow for fire in communicating space

$$v = 0.64\sqrt{\frac{gH(T_f - T_0)}{T_f}}$$

$v = \lim iting\ average\ air\ velocity, m/s$

$H = height\ of\ opening, m$

$T_f = absolute\ temperature\ of\ the\ fire\ space, K$

$T_0 = absolute\ ambient\ temperature, K$

Airflow for Fire in Atrium

- Airflow also can be used to prevent smoke originating in the atrium from flowing into a communicating space.
- The limiting air velocity can be calculated from :

$$v_e = 0.057\left(\frac{\dot{Q}}{z}\right)^{1/3}$$

This eqn is not applicable when z is less than 3m .

v_e = limiting average air velocity, m/s

$\dot{Q}$ = heat release rate, kW

z = distance above the base of fire to the bottom of the opening, m

V should not exceed 1 m/s

Fan Pressure Required

$$P = \frac{g\rho V^2 d}{T_a} = \frac{9.81 \times 1.2 \times 4000 \times 1.5}{273 + 27 + 400}$$

$= 10 Pa$

Wind Pressure

Assume wind velocity = 10m/s

$C_p = 0.5$

$$P_W = C_p \rho_V \frac{V_W^2}{2} = 0.5 \times 1.2 \times \frac{10^2}{2}$$

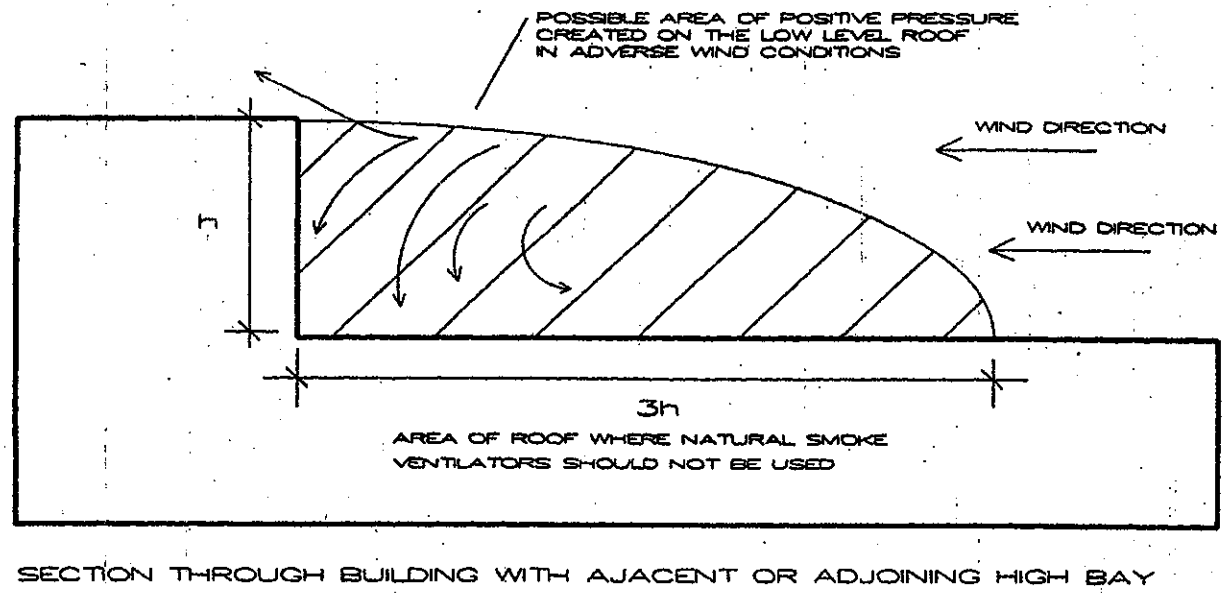
$= 30 Pa$

Guide to Good Practice on the Application of Smoke Control Equipment

28 sep 2004

1.Natural ventilator –Wind Effect,1

1. All Natural Ventilator



SECTION THROUGH BUILDING WITH AJACENT OR ADJOINING HIGH BAY

Figure 1

The performance of all natural ventilators' can be affected by adverse wind conditions , especially if back-pressure is generated by surrounding buildings. If wind hits a taller structure adjacent to the roof of the building into which natural ventilators are installed, a back-pressure may be created which can adversely affect the performance of the ventilators.

A generally accepted norm in the industry is that natural ventilators should not be installed in a lower roof area unless it is at a distance exceeding 3 times the difference in height.

1.2. Louvred Ventilators ,1

Generally, this type of unit operates satisfactory. However, the following arrangements must be avoided.

1.2.1 ventilators installed in the vertical

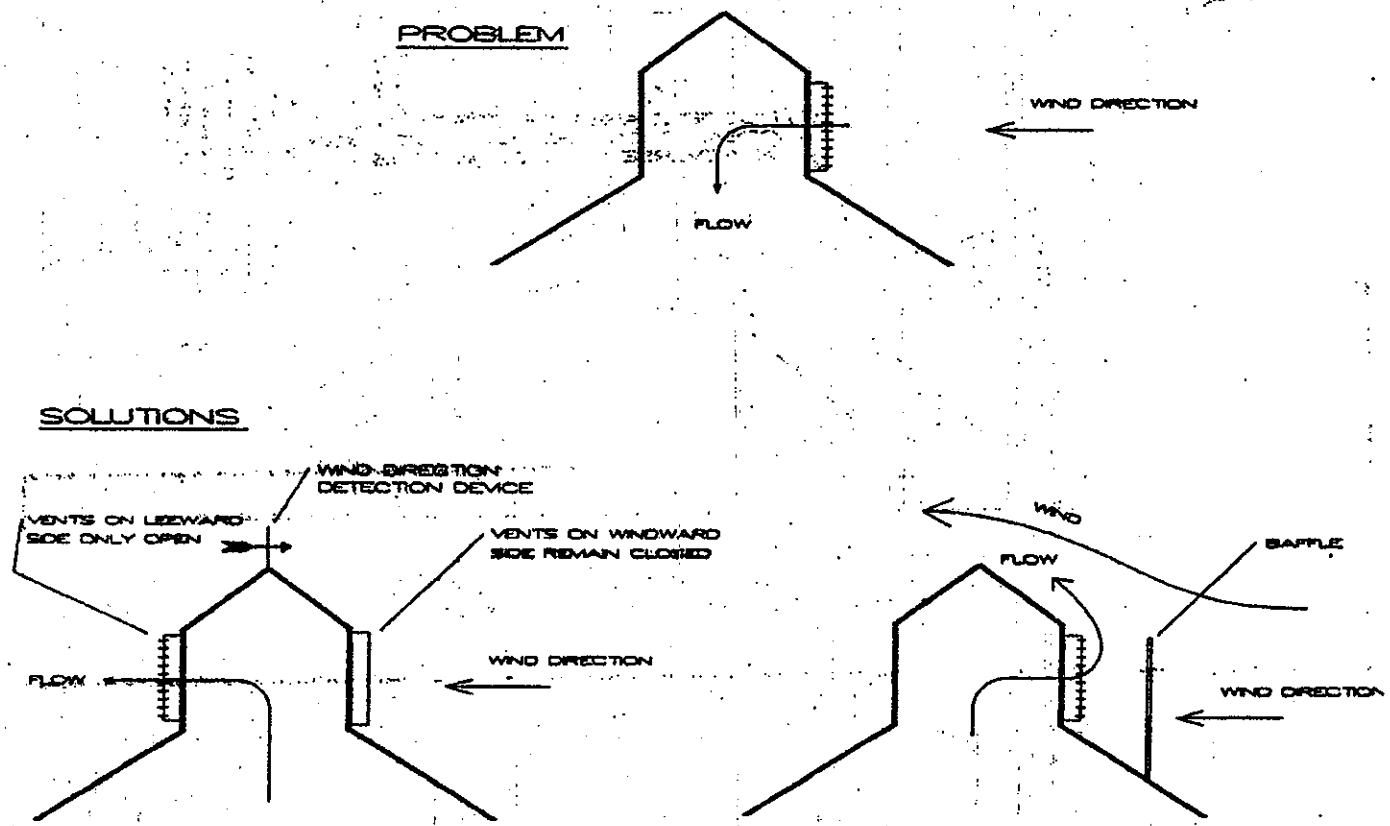


Figure 2

1.2. Louvred Ventilators ,2

1.2.2 Ventilators installed at an angle of 30 – 90 degree above the Horizontal

Research has shown that this type of product installed in a roof at an angle from the vertical down to 30 degree above the horizontal can have its performance affected in adverse wind conditions.

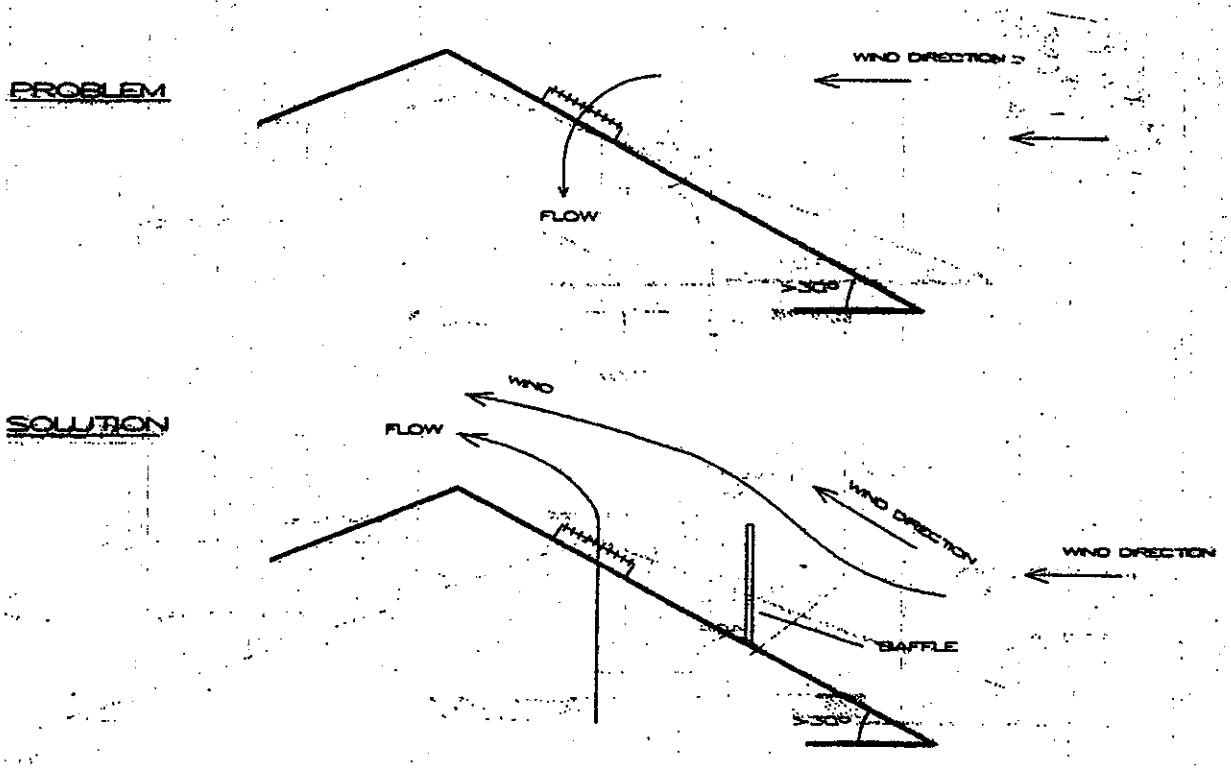


Figure 3

1.3. Single flap or hatch type ventilators, including Domelights-Top Hinged

These units are manufactured in a wide variety of materials, configurations and operating methods. As a general rule, they open to a maximum of 90 degree to the angle at which they are installed.

These units are adversely affected by wind conditions in the following situations.

1.3.1. Ventilators Installed at an angle of 0 to 10 degree above the horizontal.

Even in the horizontal condition, this type of unit will act as a wind scoop, harnessing the wind to the detriment of the unit's performance.

The only way this can be overcome is for the unit to open to 140 degree.

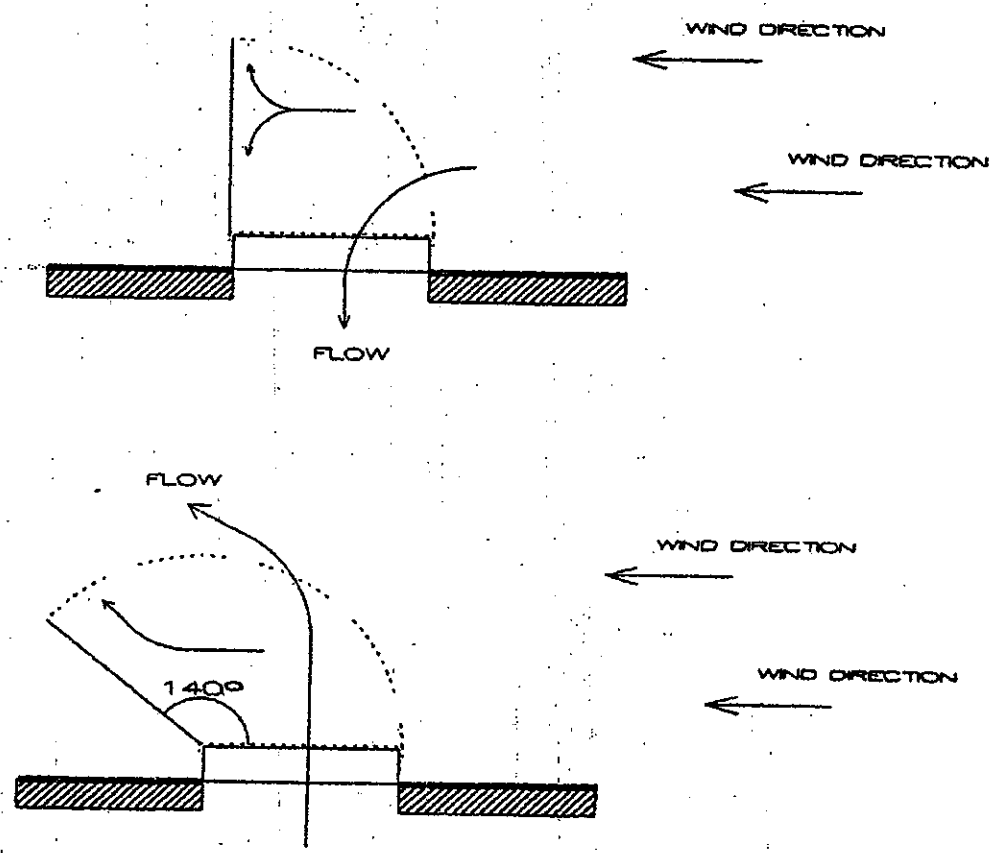


Figure 4 Single flap ventilators installed at 0 to 10 degree to the horizontal

1.3.2 Ventilator installed at an angle of 10 to 30 degree above the horizontal

This type of unit is often a proprietary window to which some control mechanism has been fitted. As a result, it is extremely unlikely to comply with the stringent requirement of BS 7346.

This product opens at the bottom of the unit, presenting an open area for wind to blow straight into the building, irrelevant of the angle to which the unit opens.

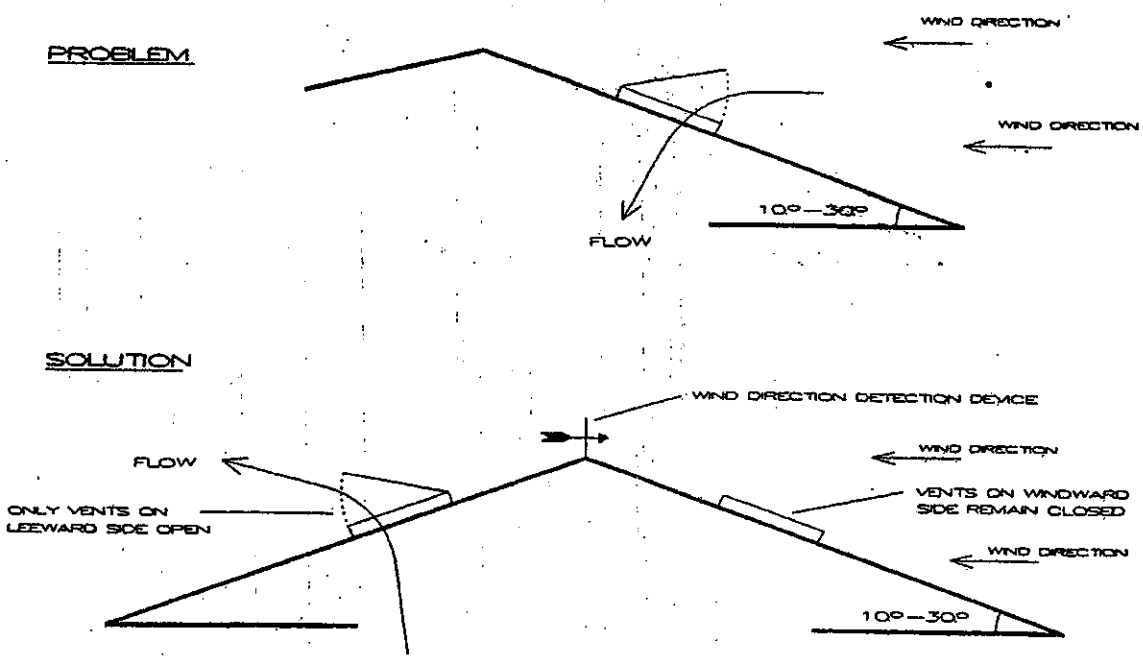


Figure 5

There is little that can be suggested to resolve this problem with a practical solution. Figure 5 illustrate this.

1.4. Single Flap or Hatch Type Ventilators, including Domelights- Bottom Hinged

This type of product is generally a specialist manufactured item by members of the smoke ventilation Association.

These products are normally installed into glazing and curtain walling structures at an angles between 15-90 degree above the horizontal.

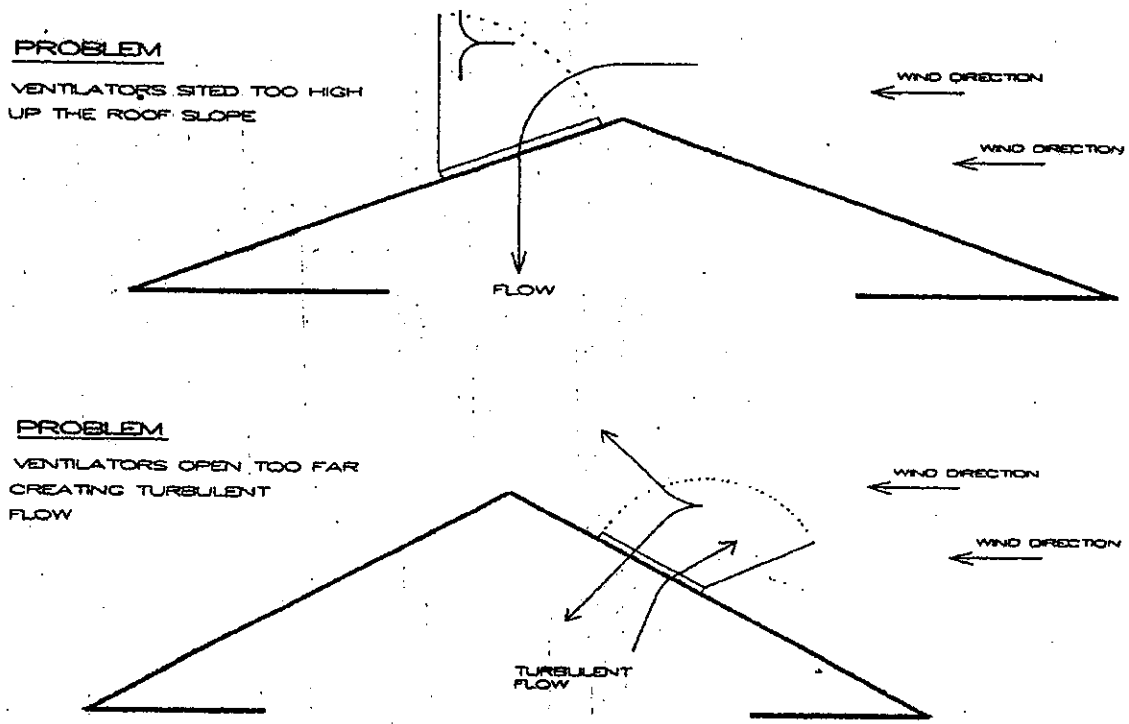


Figure 6

1.4.1 inclined installation

Unless the flap of the unit is hinged along the lowest edge parallel to the ridge of the roof, so that as the ventilator opens the flap forms a baffle to the wind, then the units will be affected in the same manner as described in 1.2.2.
However, even this measure does not resolve the problem in all circumstances. For example, if the flap opens too far, or if it is too high relative to the ridge of the building. Figure 6 illustrate the concept.

1.4.2 Vertical Installation

In this type of installation, the problem is the angle to which the ventilator opens. If the opening angle exceeds 30°, then the flap of the unit does not protect the aperture into the building, and as a result the system could fail to operate as required in adverse wind conditions.

Wind striking the vertical face of the building above the ventilators may also create turbulent air flow, affecting the ventilator's performance.

Figure 7 shows this.

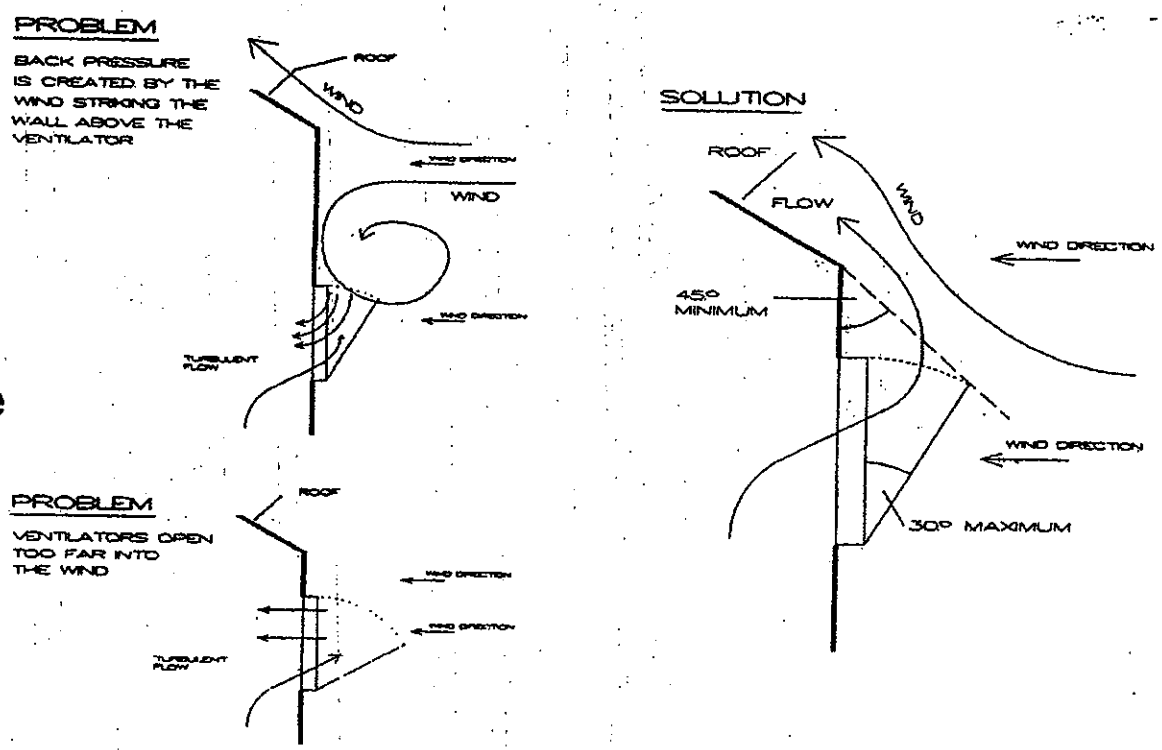


Figure 7. Vertical Installation

2. Fan Powered Ventilators

- High temperature fan powered units are frequently used to overcome the problems outlined in section 1.
- However, when using fan powered ventilators to surmount the pressures generated due to adverse wind conditions, it is frequently forgotten to add the additional resistance caused by the wind to the total system resistance. As a result, the whole system could fail to perform to duty and not meet the design criteria in adverse winds.

Fan powered ventilator,2

- 2.1 An additional value for the resistance created by the stagnation pressure generated by wind on the outlet or inlet of the fan should be allowed.
- 2.2 The use of bifurcated fans is often proposed as a solution for smoke extraction systems. It must be remembered that although the motor is not within the smoke being extracted, it might be located within the smoke layer, and therefore needs the same level of protection from the effects of high temperature as motors within the airstream, unless mounted externally, or in a totally fire-separated compartment from the space being vented.
- 2.3 The performance of the fans (pressure and volume) must always be specified at normal ambient temperature and pressure.

Fan powered ventilator,3

- 2.4 The motor must always be rated at a temperature compatible with the system design, and size to operate at ambient conditions to permit testing, and to allow for any size of fire up to that which is considered the worst likely design condition.

Design of smoke ventilation for

Single Storey Industry Buildings including those with mezzanine floors and high racked storage warehouses

Fire size

Hazard	Fire size m	Perimeter (m)	Area,m^2	Building type
Group 1	3 x 3	12	9	Breweries
Group 2	4.5 x 4.5	18	20-25	bakeries
Group 3	6 x 6	24	36	Cotton Mills
Group 4	9 x 9	36	81	Paint manufacturing
Group 5/1	3 x 3	12	9	Electrical warehouse
Group 5/2	4.5 x 4.5	18	20/25	Pharmaceutical warehouse
Group 5/3 Storage	6 x 6	24	36	Paper warehouse
Group 5/4	9 x 9	36	81	Plastic (foam)warehouse

High Racked Storage Warehouse

- The potential for fire growth needs to be fully considered since it is much greater within racked goods than with material stored only at ground level
- Although the minimum fire size should not be less than 3 m x 3 m, this minimum size should only be considered in exceptional circumstances

Heat output of Fires

Stacked chipboard	85 Kw/m^2
Books	93
Furniture	93
Cellulosics generally	160
Cardboard Reels	210
Stacked cardboard	320
Vehicles/petrol/paint	260
Cartons/electrical goods	310
Light fuel oil	1475
Stacked cartons	630

Convective Heat Output

- When heat is given off some of it is lost to the structure and contents by radiation from the hot gases.Also if the roof compartment exceed 2000 m^2 then consideration should be given to the radiative and conductive losses to the roof structure.
- A 20 % loss should be applied in the design for roof areas of 2000 m^2 or less.

Clear layer

- Minimum clear layer should be 3 m
- Where the chances of flashover present a considered risk, the clear layer should be 500 mm above the stacked goods
- To avoid disruptive of the smoke layer in any single compartment, the base of the smoke layer should be at least 500 mm above any door/opening which might either by design or chance be a source for replacement air

T ≤ 600 °C
to prevent
flashover

High racked storage warehouse

- The smoke layer base should be 500 mm above the top of the height level of stored goods.
- To avoid disruption of the smoke layer in any single compartment, the base of the smoke layer should be at least 500 mm above any door/opening which might either by design or chance be a source for replacement air
- Where the floor level changes within a compartment, the height of rise should be measured from the shallowest end for safety purposes and the overall height from the deepest end for the calculation

Smoke compartment

For single storey industry building (excluding those with mezzanine floors) and high racked storage warehouse

- Smoke compartments should have a maximum area of 2000 m^2 to 3000 m^2.
- The maximum length of any side of a smoke compartment should be 60 m
- Smoke curtains should be terminate 500 mm below the design smoke layer base

single storey industry building with mezzanine floors

- Whilst the perimeter given in the previous slide should be applied.
- The design philosophy is to allow the smoke to spill into an adjoining area, care must be taken not to exceed the total reservoir size of 2000 to 3000 m^2.
- The size of the reservoir is the combined total of the under mezzanine floor smoke control zone plus the area of the smoke control zone into which the smoke is design to flow.

Depth of smoke layer
Single storey industry buildings (including those with mezzanine floors)

- In order to avoid the risk of flashover, sufficient ventilation should be provided to ensure the smoke layer temperature does not exceed 600 C.

Depth of smoke layer
High racked storage warehouse

- Since it is extremely unlikely for a high racked storage area to be built without sprinklers. Flashover is not likely to occur.
- Since the smoke layer base is, by design, at a high level within the building, it is likely that cool smoke temperatures will results.
- Smoke temperature less than 20 C above ambient have resulted in the smoke layer stratifying within a building and not reaching roof level despite the provision of a smoke extraction system. This scenario must be considered by the designer.
- The cooling effect of the operation of the sprinklers must be considered.

Air inlet

- For any scheme to be effective, sufficient replacement air must be provided in order that extract ventilators operate properly.
- Doors and windows, as well as ventilators can be used for replacement air.
- However, if intended for use in this way they should be fitted with automatic controls compatible with the system. The designer should state the ratio of extract to inlet area used for calculations.

Air inlet

- The design inlet velocity shall not exceed 5.0 m/s
- Replacement air should be introduced at low level from all direction. However, natural ventilators in the non-fire compartments can be used if smoke curtains/screens are fitted to prevent short circuiting.
- Powered ventilators can be used for replacement air. Care should be taken as they may cause turbulence which can increase smoke/fire spread.

Wind Effect

- Installation of natural ventilators on roof over 30° from the horizontal should not be considered without some form of baffling . This baffling can be external to the unit or part of the ventilator design.
- Where changing wind directions may cause positive or negative pressure fluctuation in the building structure, natural extract ventilators should be installed in sufficient numbers and positions and controlled via wind sensor/pressure monitors to ensure that an appropriate number open at any one time.
- In in doubt about pressure distribution on the building structure, a powered system should be used.

Wind effects

- Any effects caused by the wind which may affect the smoke ventilation system proposed, must be clearly stated by the designer.

Sizing/Siting of equipment

- The size of natural and powered extract units should be limited to prevent puncture of the smoke layer
- Natural extract ventilators should not be spaced more than 20 m apart
- For natural systems, the preferred aerodynamic area of inlet should be at least equal to the aerodynamic area of extract. The designer should state the actual ratio taken into account.
- For mechanical system, inlet resistance should be considered when sizing equipment extraction and the value should be stated by the designer.

- Fan motors should be sized to operate at ambient conditions (288 K)
- Compressed air receivers should be capable of three operation of the system after power failure

Basement Smoke Control System

21 Mar 2008

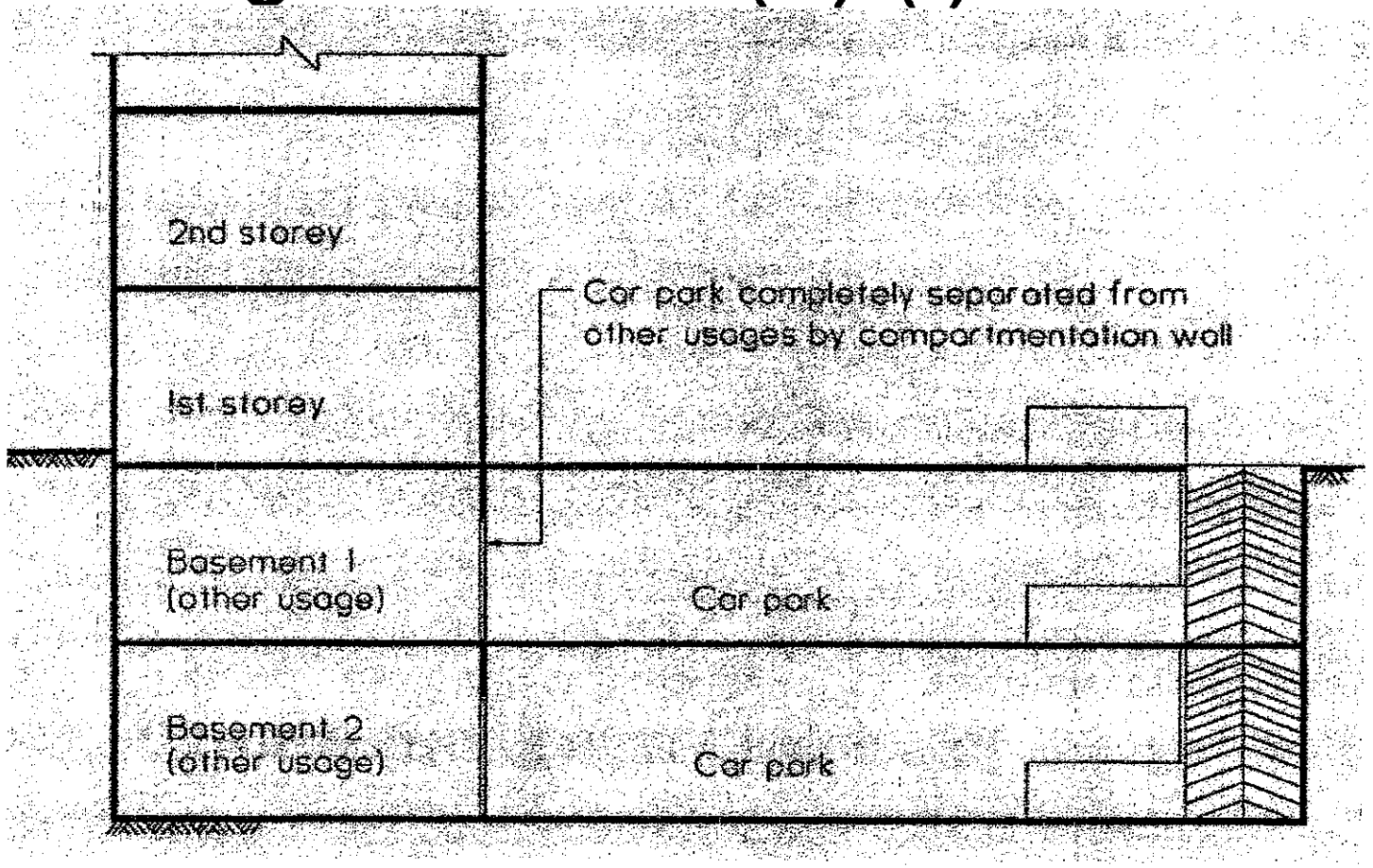
Floor area <1900 sq m

- 7.4.1. When the total aggregate floor area of all basement storeys does not exceed 1900 sq m , smoke vents in accrordance with CL.7.4.2. shall be provided.

Floor area > 1900 sq m

- 7.4.1. (b) where the total aggregate floor area of all basement storeys exceeds 1900 sq m, engineered smoke system that complies with the requirements stipulated in CL.7.4.3. shall be provided for all parts of basement with the following exceptions:
- Exception
- (i) Where the basement or a portion of the basement is used as carpark, CL. 7.1.9 can be adopted to the carpark provided it is compartmented from rest of the basement:

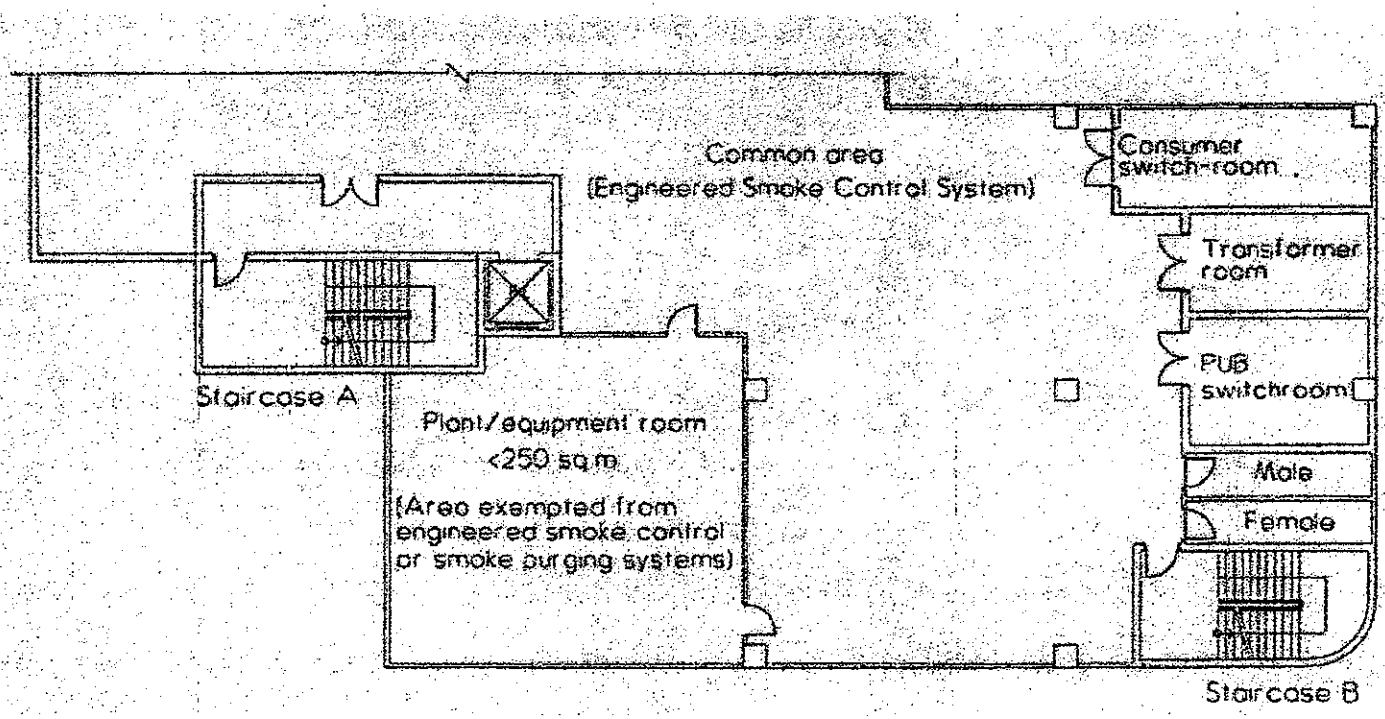
Diagram 7.4.1 (b) (i)



If the total floor area of basements 1 & 2 (other usage + car park) > 1900 sq m , engineered smoke control is required to be provided in the basement storeys; except the car parking areas in basements 1 & 2 which need to be provided with smoke purging system under clause 7.1.9.

7.4.1 (b) 9(ii)

- (ii) Plant/equipment room with floor area not exceeding 250 sq m and compartmented from rest of the basement, and provided with two doors for better reach in fire fighting operations.

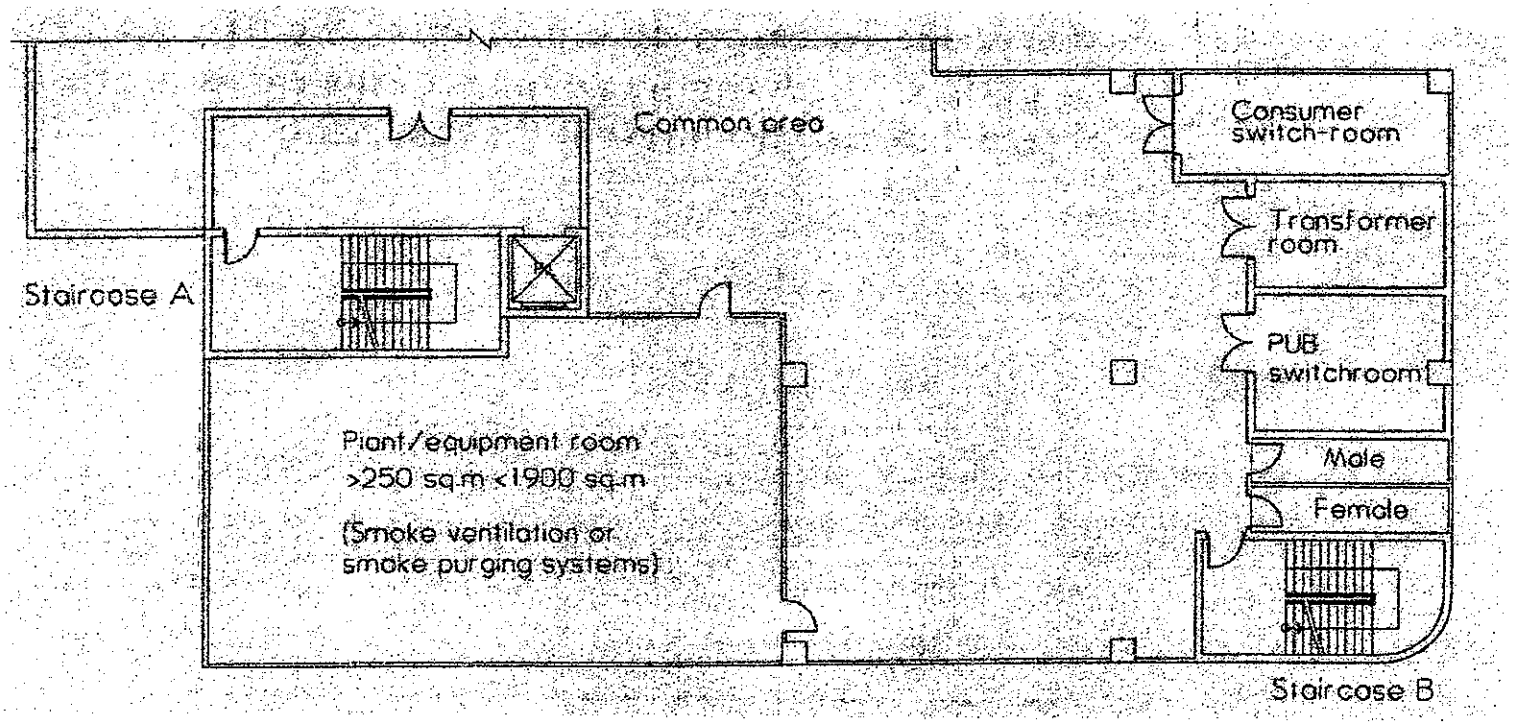


Part plan of basement storey

Where the plant/equipment room is not greater than 250 sq m , fire fighters can fight a fire in that room from its doorway, if he plant/equipment room is not larger than 100 sq m , one access door would be considered acceptable.

7.4.1 (b) (iii)

- (iii) Plant/equipment room with floor area exceeding 250 sq m but not exceeding 1900 sq m , smoke vents in accordance with cl 7.4.2 or smoke purging system of at least 9 air-change per hour shall be provided.



7.4.1 (b) (iii),2

- Where floor area of the plant/equipment room is excess of 250 sq m , but not exceeding 1900 sq m , provision of smoke vents in accordance to cl 7.4.2 or smoke purging system in accordance to cl 7.1.9 would be acceptable. Also ,for a plant room in excess of 1900 sq m , the provision of either engineered smoke control system or smoke purging system is acceptable.
- Subclause (iii) is meant to grant relaxation over the general requirement as service rooms are usually of low occupancy load. Common corridors serving multiple plant rooms/service rooms may be pressurized. Common areas outside the plant room/service rooms should be provided with engineered smoke control system.

7.4.2. Smoke vents

Smoke vents shall be adequately distributed along perimeter of basement and their outlets shall be easily accessible during fire fighting and rescue operations. Installation shall comply with the following requirements:

(a) The number and their sizes shall be such that the aggregate effective vent openings shall not be less that 21/2 per cent of the basement floor area served.

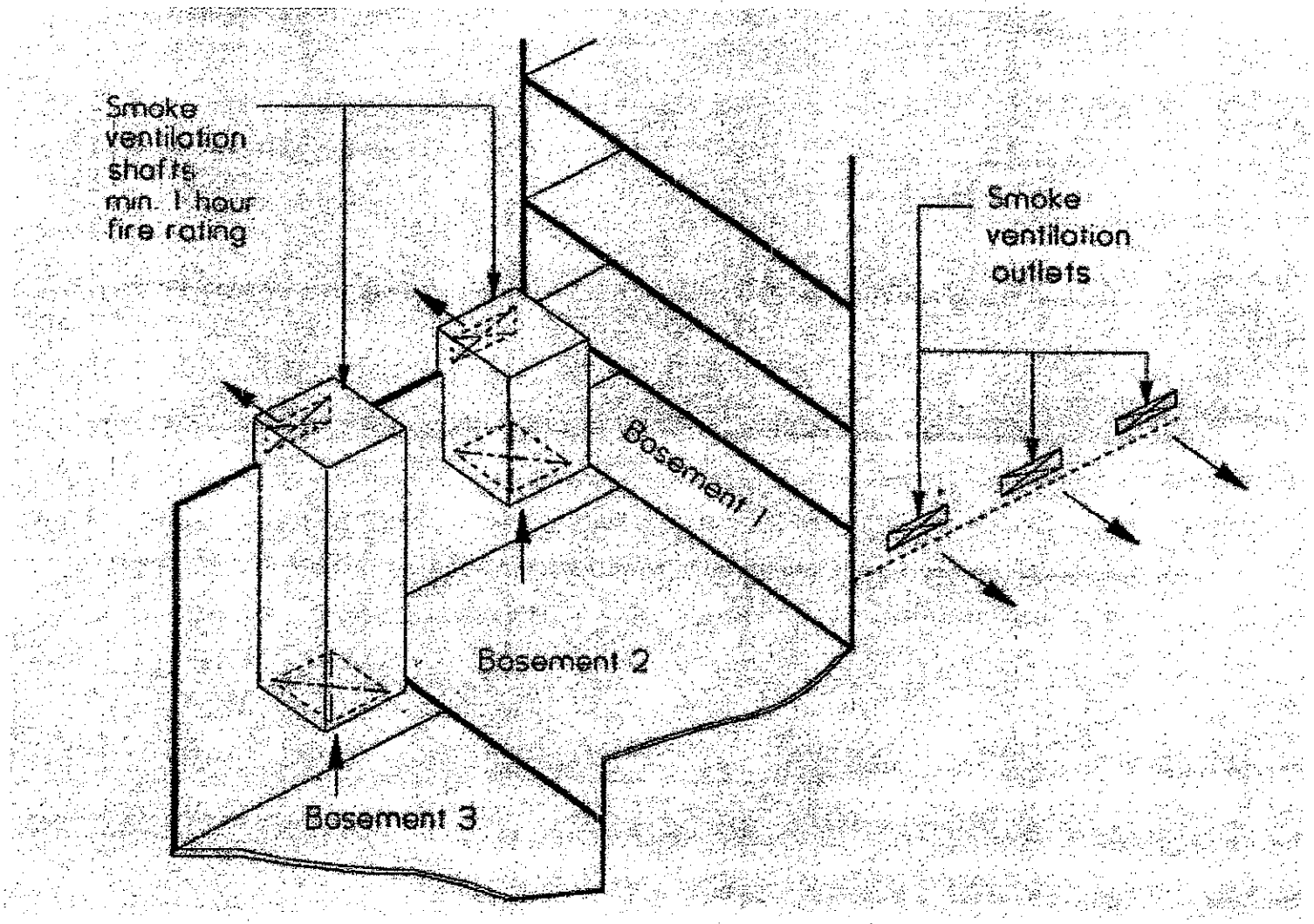
(b) The vent outlets if covered under normal conditions shall be openable in case of fire.

(c) The position of all outlets and the areas they serve shall be suitably indicated adjacent to such outlets.

(d) Where ducts are required to connect the vent to outlets, the ducts shall either be enclosed in structure or be constructed to give at least 1 hour fire resistance.

(e) Separate ducts and vent outlets shall be provided for each basement storey.

7.4.2. Smoke vents ,2



7.4.2. Smoke vent,3

- Smoke ventilation shafts which are extended through storeys above , shall be enclosed with minimum 1 hour resistance
- Separate smoke ventilation shafts and outlets shall be provided for each basement storey
- Smoke venting outlets shall be so arranged that a through draught can be created. The outlets shall be evenly and strategically located so that they can serve the general spaces in the basement and not be hidden in room
- Outlets covered by stalled boards, or approved type pavement lights shall be readily openable/breakable.

The positions of all smoke vent outlets and the basement level or areas they serve shall be suitably indicated on the external face of the building adjacent to such outlets. Smoke vents and pavement glass blocks serving only one basement storey need not be indicated by signage.

7.4.3 Engineered Smoke Control

- Where engineered smoke control system is required, it shall be provided as specified in cl 7.6.

Mechanical Ventilation Based on Heat Load (Basement Workshop : Machine and People's Heat)

- **Hotel Service Areas**
(i) At the first basement level of the hotel tower, there is a workshop cum in-house laundry which are restricted for staff only. This service area is 30 m by 10 m = 300 m^2.
This area must be fire compartmented from the car park areas.
(ii) **Heat Load Estimation**
It is estimated that the number of staff in this area is 30 (from ASHRAE Fundamentals, heat output of each person for light bench work is 0.22 kW), 25 W per m^2 is used for the the lighting load, the equipment heat load is estimated to be 25 kW, and the external heat transmission load is 15 kW.
Therefore, the total heat to this compartment = (30 * 0.22) + (0.025 * 300) + 25 + 15 = 54.1 kW.

- (iii) **Air Flow Requirement**
Using a designed outdoor temperature of 33 deg C and a maximum indoor temperature of 36 degree C, then the air flow volume required,
Q = 54.1 / [(36 – 33) * 1.2]
= 15.03 m^3/s = 54,100 m^3/hr.
(iv) **Air Change Per Hour (ACH)**
The floor to floor height of the basement level is 3.0 m. Hence, the volume of these service areas = 300 * 3 = 900 m^3.
Therefore, the ACH = $\frac{15400}{900 * 1.2}$
= 14 approx.

Mechanical Ventilation Based on Heat Load (Basement Workshop : Fire Mode Requirement)

- **Code Requirement**
  (i) In accordance to code, a min. of 6 ACH is required during normal mode and a min. of 9 ACH is required during fire mode. The calculated air flow requirement far exceeds both code requirements. Therefore, acceptable.
  (ii) The service areas in basement car park is to be maintained at positive pressure. Hence, supply air shall be, say 12% more than the exhaust air at 15.03 m^3/s.The fans are to be connected to emergency power supply in addition to the normal supply. Power (& control) cables shall be fire-rated.

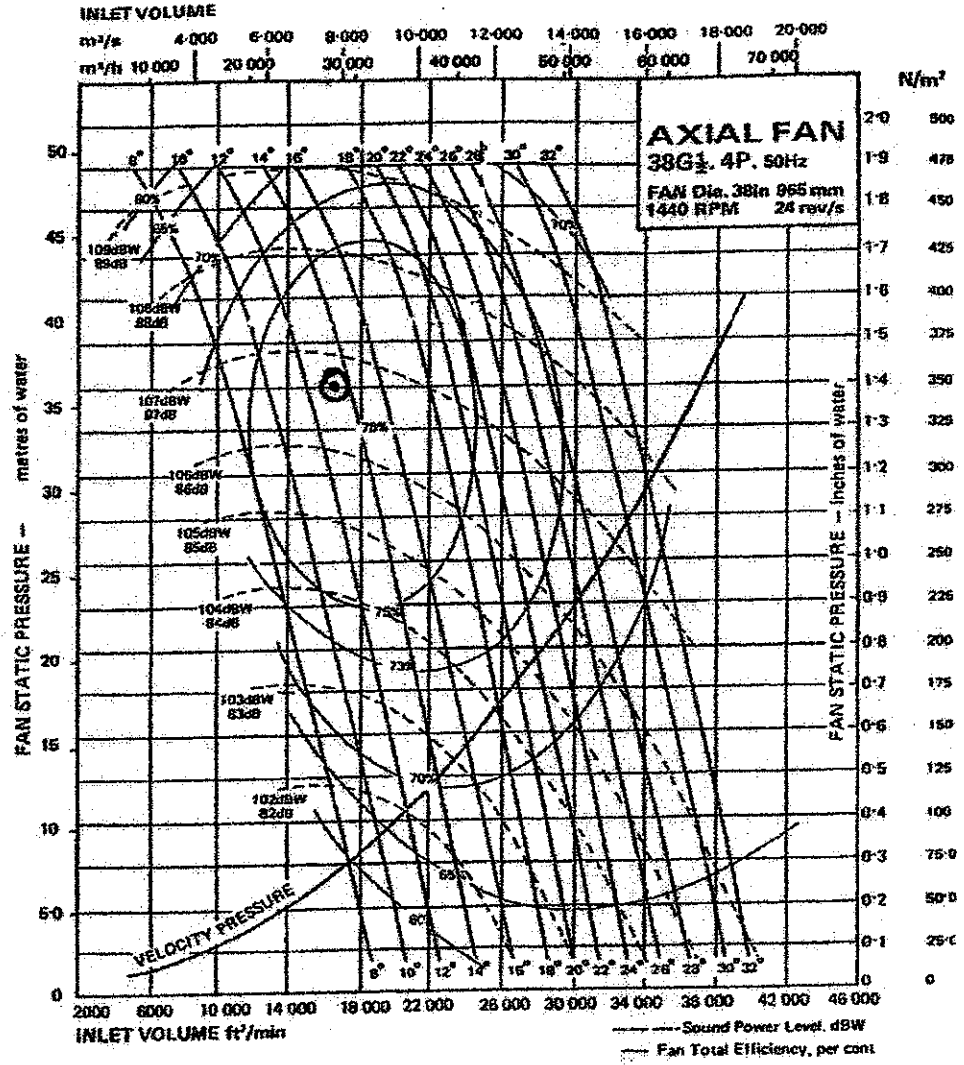
- (iii) 2 ducted supply air axial fans of capacity $\frac{15.03 * 1.12}{2} = 8.42$ m^3/s is used. (Estimated total fan pressure, say is 400 Pa).
  Correspondingly, 2 ducted axial fans of capacity 15.03 / 2 = 7.52 m^3/s is used for the exhaust system. (Estimated total fan pressure is, say 350 Pa.

  The selected fan characteristic curves are as shown in the next 2 pages.

Mechanical Ventilation Based on Heat Load (Basement Workshop Exhaust Air Fan Characteristic)

(7.52 m^3/s @ 350 Pa)

(1 Pa = 1 N/m^2)



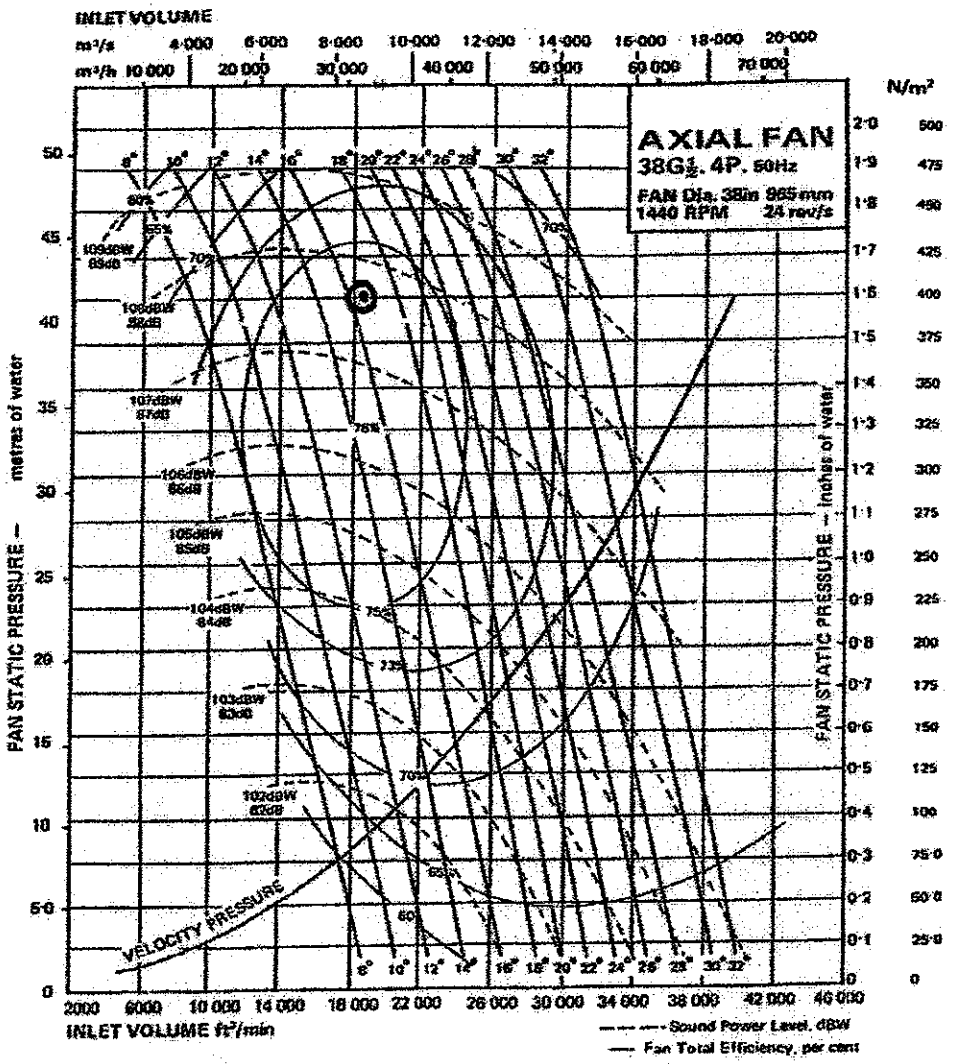
THREE PHASE 415v

ANGLE	8°	10°	12°	14°	16°	18°	20°	22°	24°	26°	28°	30°	32°
PHASE	3	3	3	3	3	3	3	3	3	3	3	3	3
kW Power Absorbed Per Stage	2.68	3.43	4.03	4.70	5.37	6.19	7.16	7.98	8.90	9.84	10.89	12.36	13.67
hp Power Absorbed Per Stage	3.60	4.60	5.40	6.30	7.20	8.30	9.59	10.7	11.9	13.2	14.6	16.6	18.8

ACOUSTIC ANALYSIS

IN DUCT SPECTRUM	OCTAVE BAND MID-FREQUENCY Hz					
	125	250	500	1000	2000	4000
CONSTANTS	6	6	10	11	14	20

To obtain Spectrum subtract above constants from Sound Power level of the fan selected



THREE PHASE 415v

ANGLE	8°	10°	12°	14°	16°	18°	20°	22°	24°	26°	28°	30°	32°
PHASE	3	3	3	3	3	3	3	3	3	3	3	3	3
kW Peak absorbed Per Stage	2.58	3.43	4.03	4.70	5.37	6.19	7.16	7.36	8.80	9.84	10.89	12.38	13.87
hp Peak absorbed Per Stage	3.60	4.80	5.40	6.30	7.20	8.30	9.60	10.7	11.8	13.2	14.6	16.6	18.6

ACOUSTIC ANALYSIS

IN DUCT SPECTRUM	OCTAVE BAND MID-FREQUENCY Hz					
	125	250	500	1000	2000	4000
CONSTANTS	6	6	10	11	14	20
To obtain Spectrum subtract above constants from Sound Power level of the fan selected						

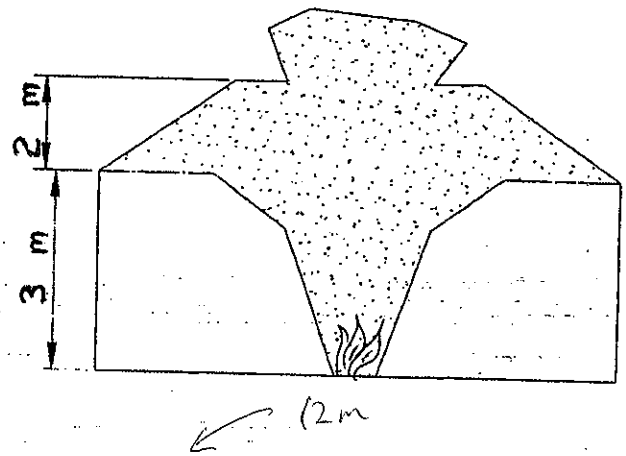
Mechanical Ventilation Based on Heat Load (Basement Workshop Supply Air Fan Characteristic)

(8.42 m^3/s @ 400 Pa)

(1 Pa = 1 N/m^2

216

Example 1
Single storey shopping mall



$$m = 0.2PY^{3/2}$$

$$= 0.2x12x3^{3/2} = 12.5\ kg/s$$

$$\theta_{cd} = \frac{Q}{m} = \frac{5000}{12.5} = 400\ K$$

Using natural ventilation

$$A_V C_V = \frac{M}{\rho_o}\left[\frac{T_C^{\,2} + \left(\frac{A_V C_V}{A_i C_i}\right)^2 T_O T_C}{2 g d_p \theta_C T_O}\right]^{1/2}$$

Let $A_V C_V = A_i C_i$

$$T_o = 32 + 273 = 305\ K$$

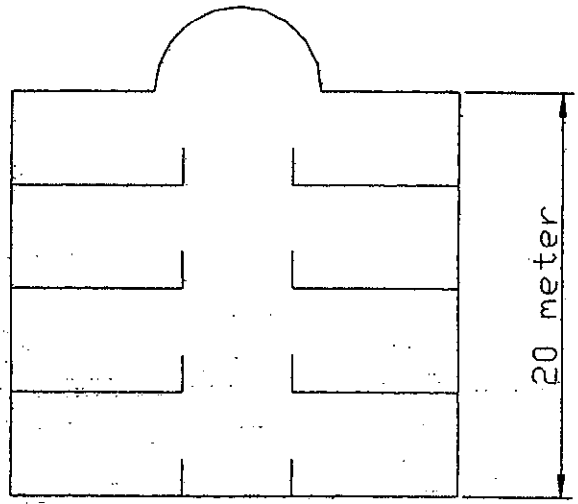
$$T_C = 273 + 400 = 673$$

$$A_C V_C = \frac{12.5}{1.2}\left[\frac{673^2 + 1x305\,x673}{2x9.81x2x400x305}\right]^{1/2} = 3.86$$

if $C_V = 0.6$, *then*

$$A_V = \frac{3.86}{0.6} = 6.43m^2$$

Example 2 Shopping center



Fire size = 12 m ,5 MW
L = 10 m

$$m = 0.58\rho\left[\frac{gQL^2}{\rho C_p T_1}\right]^{1/3}(Z+\Delta)\left[1+\frac{0.22(Z+2\Delta)}{L}\right]^{2/3}$$

$$= 0.58x1.2\left[\frac{9.81x5000x10^2}{1.2x1.02x305}\right]^{1/3}(12+0.9)\left[1+\frac{0.22(12+2x0.9)}{10}\right]^{2/3}$$

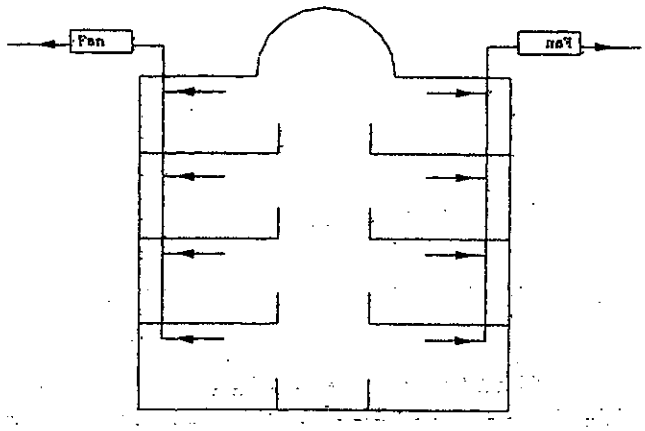
$$= 376\ kg/s$$

$$\theta = \frac{Q}{mC_p} = \frac{5000}{376x1.02} = 13.29^{\circ}C$$

Temperature of smoke is too low to rise, cannot use ventilation.

The solution is to use smoke extract from shop units.

Example 2



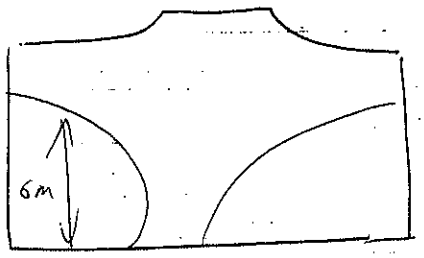
Example 3 :Warehouse
Fire size = 12 m perimeter

9 square meter Electrical Goods, Heat output = 310 kW/m^2

Y = 6 meter

$$m = 0.2PY^{3/2} = 0.2x12x6^{3/2} = 35.3\,kg/s$$

$$Q_f = 0.8A_f q_f = 0.8x9x310 = 2232\,kW$$



$$\theta = \frac{Q}{m\,C_p} = \frac{2232}{35.3x1.02} = 62.6\,K$$

$$T_C = 273 + 32 + 62.6 = 367.6\,K$$

$$T_O = 273 + 32 = 305\,K$$

$$If\ \frac{A_v C_V}{A_i C_i} = \frac{1}{2}$$

$$A_V C_V = \frac{M}{\rho_o}\left[\frac{T_C^{\,2} + (\frac{A_V C_V}{A_i C_i})^2 T_O T_C}{2g\,d_p \theta_C T_O}\right]^{1/2}$$

$$= \frac{35.3}{1.2}\left[\frac{367.6^2 + 0.5^2 x305x367.6}{2x9.81x1.5x\,62.6\,x\,305}\right]^{1/2} = 15.9$$

$$\theta_c = \frac{5000}{12.47 x 1.02} = 394\ K$$

$$T_C = 273 + 32 + 394 = 699^\circ K$$

$$V = \frac{M}{1.2} x \frac{T_C}{T_O} = \frac{12.47}{1.2} x \frac{699}{305} = 23.8 m^3/s$$

$$A = \frac{23.8}{10} = 2.38\ m^2$$

A large duct, approximately 1.5 m x 1.5 m can be used to extract smoke from the rear of each shop. Dampers open from the shop on fire into the duct. Dampers in all other shops connected to the ductwork must remain closed. A shaft then rises from the common duct to a fan at roof level.

Smoke detection is required within each shop. It is linked to a control system which opens the dampers to the shop on fire, ensures all others are closed, and starts the fan at roof level. The ductwork will need to be fire-rated as it will pass through compartment walls and floors.

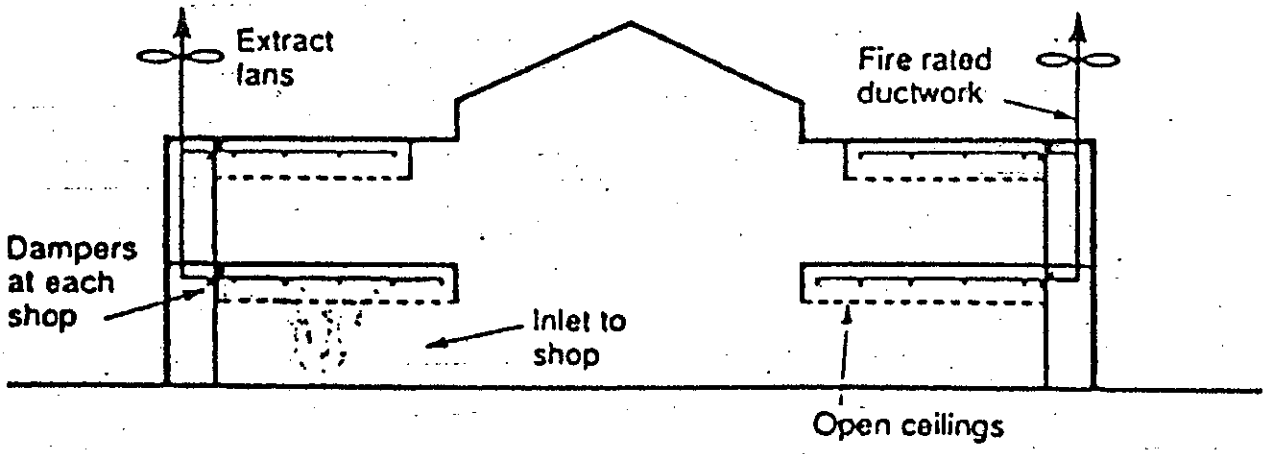


Figure 1: Shop smoke extract system.

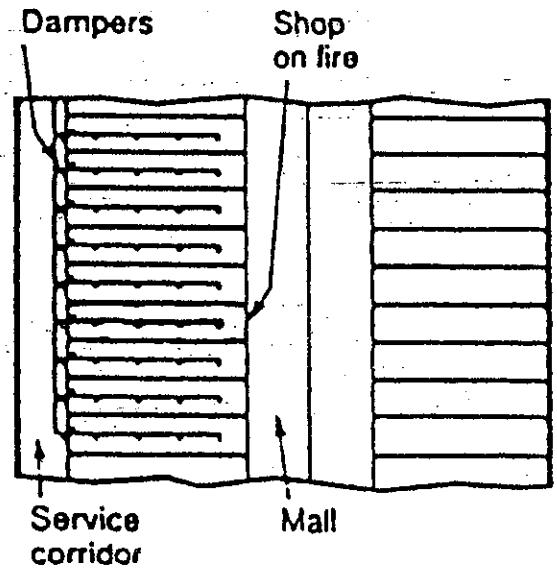


Figure 2: Shop extract – plan view.

Example 4 Carpark

Fire size = 2.5 MW

P = 12 meter

Y = 3 meter

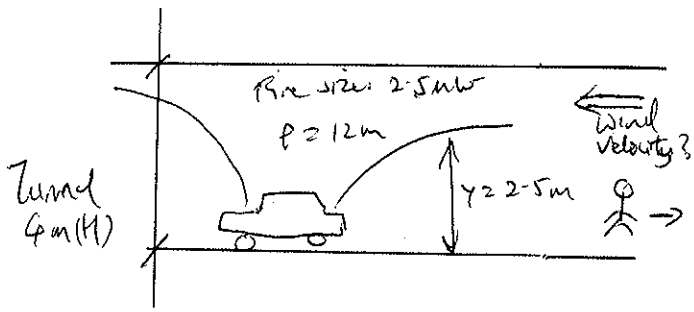
$$m = 0.2PY^{3/2} = 0.2x12x3^{3/2} = 12.47\ kg/s$$

$$\theta_c = \frac{Q}{mC_p} = \frac{2500}{12.47x1.02} = 196.5\ K$$

$$T_C = T_O + \theta_C = 273 + 32 + 196.5 = 501.5\ K$$

$$V = \frac{m}{1.2} x \frac{T_C}{T_O} = \frac{12.47}{1.2} x \frac{501.5}{305} = 17.1\ m^3/s$$

Eg 5



$M_S = 0.188 Y^{3/2}$

$= 0.18 \times 12 \times 2.5^{3/2}$

$= 8.53\ kg/s$

$\theta = \frac{Q}{m} = \frac{2500}{8.53} = 293°C$

$T_s = 273 + 30 + 293 = 596K$

$T_o = 273 + 30 = 303K$

$V = 0.64 \sqrt{\frac{gH(T_s - T_o)}{T_s}}$

$= 0.64 \sqrt{\frac{9.81 \times 4 \times (596 - 303)}{596}}$

$= 7.8\ m/s$

Example
Smoke Extract Example from UK

If smoke from a fire in a shop unit is to be prevented from flowing into the mall, every shop unit and even small sales kiosks must have a smoke extract system. Ordinary, non-fire-rated glazing is likely to shatter at an early stage.

Within the shop unit, a smoke extract rate of between 20 and 30 m^3/s at very high temperature is required, to prevent smoke entering the mall. It is, therefore, very unlikely that environmental ventilation system could also serve as a smoke extract.

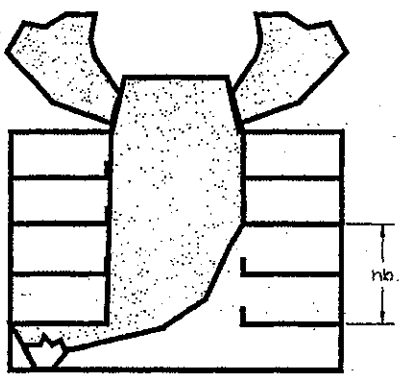
Mass flow rate of smoke from a 3 x 3 m fire

$M = 0.2\ PY^{3/2} = 0.2 \times 12 \times 3^{3/2} = 12.47$ kg/s

Example for Atrium

Lim Eng Seng

Example Atrium



H_b=7 m

Formula

$$m_{smoke} = 0.58\rho\left[\frac{gQ_W L^2}{\rho C T_0}\right]^{1/3}(h_b+\Delta)\left[1+\frac{0.22(h_b+2\Delta)}{L}\right]^{2/3}$$

m_{smoke} = *mass flow rate of smoke entering the smoke layer at height* h_b, *kg / s*

Q_W = *Convective heat flux , kW*

L = *Length of void edge past which gases spill, m*

C = *specific heat of air*

ρ = *density of warm gases at height* h_b

Δ = *Empirial height of virtual source below void edge, m*

h_b = *Height of rise of thermal plume above void edge, m*

T_0 = *Absolute ambient temperature*

For practical purpose it can be taken that

$\Delta = 0.3$ *times the height of the compartment opening, H*

i.e. $\Delta = 0.3H$

Example Shopping Centre

Fire size $= 12m, 5\ MW$

$$m = 0.58\rho\left[\frac{gQL^2}{\rho C_p T_1}\right]^{1/3}(h_b+\Delta)\left[1+\frac{0.22(h_b+2\Delta)}{L}\right]^{2/3}$$

$$= 0.58x1.2\left[\frac{9.81x5000x10^2}{1.2x1.02x305}\right]^{1/3}(7+0.9)\left[1+\frac{0.22(7+2x0.9)}{10}\right]^{2/3}$$

$$= 146\ kg/s$$

$$\theta = \frac{Q}{mC_p} = \frac{5000}{146x1.02} = 33.58^{o}C$$

Hole Plugging,1

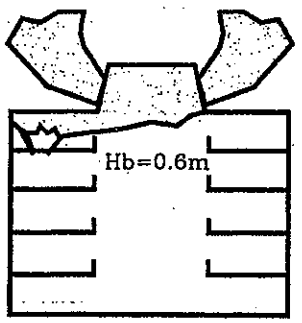
$$m_{max} = 3.13x2x8^{5/2}\left(\frac{33.58}{273+32+33.56}\right)^{1/2}\left(\frac{273+32}{273+32+33.56}\right)^{1/2}$$

$= 338.7\ kg/s$

Therefore

One Exhaust inlet only.

Pg 172 formula



$$m = 0.58x1.2\left[\frac{9.81x5000x10^2}{1.2x1.02x305}\right]^{1/3}(0.6+0.9)\left[1+\frac{0.22(0.6+2x0.9)}{10}\right]^{2/3}$$

$= 0.696x23.58x1.5x1.035$

$= 25.5kg/s$

$$\theta = \frac{5000}{25.5x1.02} = 192K$$

$$m_{max} = 3.13x2x2\left(\frac{192}{273+32+192}\right)^{1/2}\left[\frac{305}{273+32+192}\right]^{1/2}$$

$$= 6.09kg/s$$

$$Number\ of\ exhaust\ inlet = \frac{25.5}{6.09} = 4.2 = 5$$

Therefore , minimum five inlets are used.

The Storage of Flammable Liquids

Feb 2008

SS 532 Code of Practice for the Storage of Flammable Liquids

- Fans are to be connencted to emergency power supply in addition to the normal supply
- The system shall be provided with an airflow warning device

Note : Vapors from flammable liquids are heavier than air, so the ventilation system should be designed to scavenge vapors from the lower parts of the source

SS 532 Code of Practice for the Storage of Flammable Liquids

- Illustrated Example : Flammable/Combustible store.

Store Room located inside building

Room fully enclosed with fire-rated walls & doors

$$Store\ Area = 3.64\ m^2 \quad ; Store\ height = 2.7\ m$$

$$Store\ volume = 3.64 * 2.7 = 9.83 \quad m^3$$

SS 532 Code of Practice for the Storage of Flammable liquids

(A) Exhaust Fan:

$$Fan\ capacity = \left(0.3\ m^3 / \min\right) / m^2 \bullet 3.64\ m^2$$

$$= \ 1.09\ m^3 / \min$$

$$Use\ 5\ m^3 / \min\ as\ 5\ m^3 / \min > 1.09\ m^3 / \min\ (SS532)$$

$$(Check\ CP13 : 20ACH$$

$$20\ x \frac{9.83 m^3}{60\ \min} = 3.28\ m^3 / \min)$$

$$Hence, \min imum\ exhaust\ fan\ capacity$$

$$= 5\ m^3 / \min$$

SS 532 Code of Practice for the Storage of Flammable Liquids

(B) Supply Air Fan

As Exhaust Fan Capacity is 5 m^3/min.

Proposed to use 15% less for supply fan capacity.

$$Supply\ fan\ capacity$$
$$= 0.85 \bullet 5 = 4.25\ m^3 / \min$$

Rooms: Flammable/Explosive Substances

- In addition to CP13, also refer to CP40: 1987, Code of Practice on The Storage of Flammable & Combustible Liquids
- (CP40 now superseded with SS 532: 2007
- Code of Practice for The Storage of Flammable Liquids)

CP532 : 2007 Code of Practice on Storage of Flammable & Combusti

- For Mechanical Ventilation of above code, are summarized as follows :

(1) Both fresh air supply and exhaust are required

(2) The termination points (inside the room) of the supply & exhaust shall be immediately above the upper limit of the spillage compound.

CP532 : 2007 Code of Practice on Storage of Flammable & Combusti

(3) These supply & exhaust points inside the room shall be on opposite walls

(4) The sizes of these points shall be equal

(5) The exhaust fan/s shall be larger than the supply air fan/s so as to create a negative pressure in the room

(6) The MV system shall be capable of exhausting 0.3 m^3 per minute per m^2 of floor area, or 5 m^3 per minute, whichever is greater.

(Note : CP13 requires 20 ACH only)

CP532 : 2007 Code of Practice on Storage of Flammable & Combusti

(7) The air velocity of the supply air outlet & exhaust inlet in the room shall not be lesss than 5.0 m/s

(8) Any intake or exhaust duct terminating in open air shall be at least

(a) 2 m from any opening into a building;

(b) 4 m from the outlet of any chimney or flue (or generator set exhaust)

(c) 3 m above ground

CP532 : 2007 Code of Practice on Storage of Flammable & Combusti

(9) The external exhaust point of any exhaust duct shall be minimum 3 m from the inlets of any inlet ducts (Note : CP 13 requires minimum 5 m)

CP532 : 2007 Code of Practice on Storage of Flammable & Combusti

(10) Any duct that passes through a building shall be fire-rated of minimum of 3 hour

(11) The mechanical ventilation system shall be so designed that it either :

(a) Operates continuously;

or

(b)operates whenever work is being done inside the room

smoke = 2 hrs

CP532: 2007 Code of Practice on Storage of Flammable & Combusti

(12) Fan blades and nearby components shall be made of materials that have minimal potential for giving off sparks when struck

(13) Exhaust fan to be interlocked with supply air fan such that when the exhaust fan is not operating the supply air fan shall automatically off & unable to operate but not vice versa.

(14) Fans are to be connected to emergency power supply in addition to the normal supply.

Room with flammable and explosive substances

Apr 2008

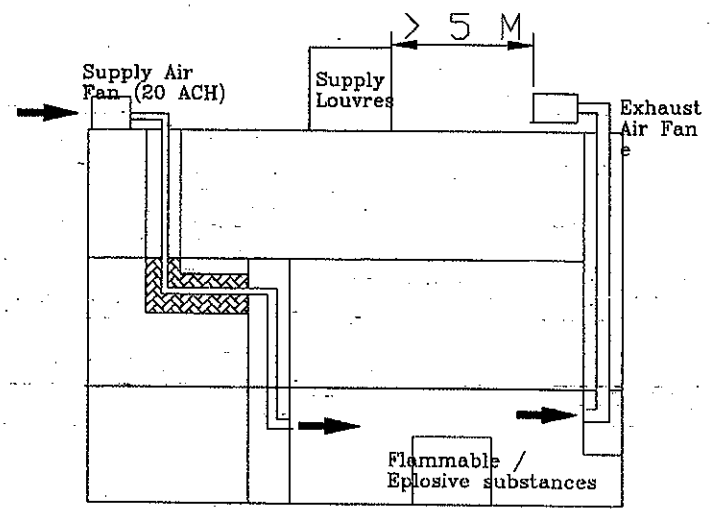
Flammable/Explosive substances

- (a) Mechanical ventilation system where required for rooms which involve the use of flammable and explosive substances shall be independent from those serving other parts of the building. It shall comply with the following requirements :

Flammable/Explosive substances

- (i) Ventilation system shall consist of exhaust and supply part with a rate of 20 ACH or any other rates acceptable to the Relevant Authority. The exhaust shall be direct to the external and shall not be less than 5 m from any air intake openings.
- (ii) where such ducts run outside the room they shall either be enclosed in a structure or be constructed to give at least the same fire rating as the room which they serve or that of the room through which they traverse, whenever is higher. The rating shall apply to fire exposure from both internal and external of the duct or structure. Where the duct risers are required to be enclosed in a masonry shaft, they shall be compartmented from the rest of the shaft space containing other ducts or services installations.

20 ACH



Flammable/Explosive substances

- (iii) No fire damper shall be fitted in either supply or exhaust duct required under this clauses; and
- (iv) Duct serving other areas shall not pass through rooms involving use of flammable and explosive substances.

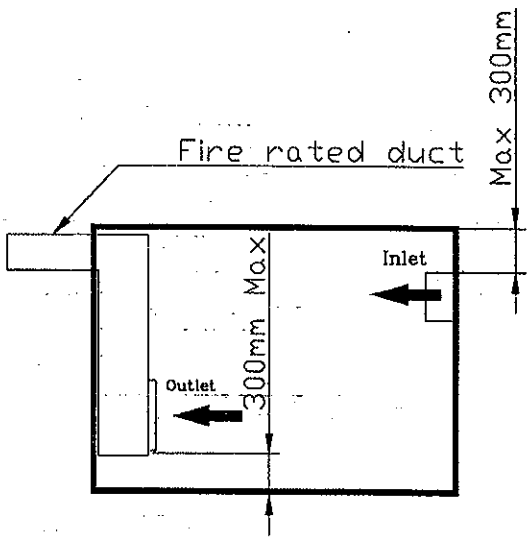
Flammable/Explosive substances

- The exhaust system (without fire damper) is required to operate efficiency to remove any gaseous or flammable vapour from the room.
- Flammable/explosive materials shall not be kept or stored in the building. Small quantity of diesel oil (700 litres) would be permissible in generator room.

(a) Where the flammable vapour being removed is heavier than air

- - (i) at least one air outlet shall be located at a point near a wall, and no higher than 300 mm from the floor , and
- -(ii) at least one air inlet shall be located near the opposite wall, no lower than 300 mm from the ceiling

The flammable vapour is heavier than the air will be discharged via the exhaust system (without damper) as shown below :



(b) where the flammable vapour being removed is lighter than air,

- -(i) At least one air inlet shall `be located at a point near a wall, and no higher than 300 mm from the floor, and
- -(ii) At least one air outlet shall be located near the opposite wall, no lower than 300 mm from the ceiling.

Chapter Air Contaminant Emission Sources and Dilution ventilation

The three common air contaminant control approaches are :

- Dilution of airborne contaminants with " fresh" (or less contaminated) air (usually outdoor air, OA)
- Emission source control (which is closely related to dilution)
- Treatment (filtration /Scrubbing) and subsequent recirculation of indoor air (Treatment is used less often than the others and is covered)

Most buildings often use a combination of these approaches to control air quality but dilution ventilation is always a part of the control strategy.

This chapter covers evaluation of emission sources and the development of appropriate ventilation flowrates.

A. Emission Source Evaluation

Principles :

Cost-effective dilution ventilation rates are directly related to air-contaminant emission rates . Two principles can be stated.

Reduced emissions = reduced need for dilution ventilation and , more specifically

Dilution ventilation rate =f (emission rate, acceptable concentration)

We can minimize dilution ventilation requirements if we :

1. Use specify , and/or small equipment, materials and furnishings with low emission rates, e.g. specify low-emitting carpets
2. Limit or minimize emissions after equipment, materials and furnishings are in place, e.g. cover emitting surfaces with an emission retardant.

Types of Emission Sources

Four major classess :

Wet materials

1. Wet materials emitting into the building air during contruction. These include solvents, paints, adhesives and sealers applied during the construction or remodeling of a building.Most of the emissions occur during the first few hours or days after application but some may continue to emit at low rates for days, weeks, or months after. Exposure may be controlled by applying materials well before the building or space is (re)occupied.
2. Wet materials emitting into the building air during occupancy. These include such materials as cleaners, waxes, disinfectants, and biocides. Because the building may be occupied during application, the potential for exposure and complaints is high. Another sourse is the use of boiler steam for himidification which contains water treatment chemicals.

Dry Materials

3. Dry material emitting into the building air. These include such materials as the SBR latex backing found on many carpets, styrene monomer from plastics, or formaldehyde from press wood. The offending emission is usually some material which was added during the manufacturing process. When the product is unpacked at the site, a large emission may occur, decaying quickly as time passes. Others may emit for months or years (e.g. formaldehyde from wood products).

Sink materials

4. Dry materials emitting adsorbed chemicals. Surfaces can absorb/adsorb and desorb chemicals from the air adjacent to the material. (" Sink ") . The rate of takeup and desorption is a function of the sink material , the surface area , the temperature, the chemical, and the chemical's concentration in the air. Materials with rough or fleecy surfaces (such as cloth or paper) adsorb more readily than smooth materials (such as glass or steel). Sinks can become loaded with chemicals during high air concentrations. (i.e. during construction) and may off load slowly during occupancy.

Specifying Dilution-Friendly Materials

When selecting or specifying building materials, equipment , or furnishings, follow these useful approaches to minimize the need for dilution :

Emission

Select materials with the lowest emission rates, lowest toxicity, the least potential for creating irritation, and the highest odor detection threshold.

Emission rate

Selection materials with no emissions is the ideal. Where not available , choose materials with low long-term emission rates , or materials which have high emission decay factors. Some materials may emit low quantities initially **but may continue to emit over a long period of time.**

Environment

Choose materials which are not adversely effected by temperature or humidity. Materials which significantly increase emissions with increases in temperature or humidity are likely to create IAQ episodes in the future.

Sink potential

Choose materials which are least likely to become sinks, and as such, secondary emission sources. Factors affecting sink ad(b)sorption/desorption include :

1. Roughness : Smooth materials are less likely to become sinkes than fleecy or rough materials
2. Temperature : Affect rate of uptake and desorption, diffusion , and rates of evaporation
3. Humidity : Affects rates of emission of formaldehyde from pressed wood and other carriers
4. Air movement : Affects concentrations of chemicals in air; affect size of boundary layer or air (a tiny layer of 'quiet" air on the surface which controls the rate of gas-phase mass transfer at the sink.)
5. Maintenance : choose 'maintenance free" materials when feasible because they are less likely to become emitter. Materials which can be cleaned with water are less likely to become sources of IAQ complaints.
6. Aging : many specifiers require newly manufactured equipment and furnishings to be "aired out" or 'aged" before installation, or before people occupy the building. This creates an opportunity for initial "burst" of emissions to occur remote from the building and its occupants. For example, carpet can be rolled out for several days or weeks in a warehouse before installation.

Proposed content Maximums for a "low emitter"

Material	Maximum VOC Content , grams/litre of material
Paint (water-based)	250
Paint (solvent-based)	380
Caulks , adhesives and sealants	20

Varnish (water –based)	300
Stain (water-based)	250

Source : Healthy Materials, spring 1, 1994

Emission factors

Emission factor (e.g. the rate of emission expected from a material based on a size parameter and the time elapsed) are often used to estimate appropriate dilution flowrates.

As a reminder, remember that as emission factors (and actual emission rates) increase, greater control is required at the site (e.g. large quantities of OA may be required for dilution)

Emissions guideline

"Permissible Limits " to the amount of emissions allowed (or desirable) are slowly creeping into the literature and , to a lesser extent, into the rules and regulations of society.

The following table, is a summary of "Maximum Emission" factors tentatively recommended by Tucker in 1995.

Emission Rate Recommendations for Specifying Materials

Material	Maximum Emissions, TVOC mg/hr-sq meter	Examples
Floor coatings	0.6	Varnish, paint, wax
Flooring materials	0.6	Carpet, pads, adhesives
Wall coatings	0.4	Paint
Wall materials	0.4	Wallpaper
Moveable wall partitions	0.4	Office partitions
Office furniture	2.5 mg/hr per workstation	Office spaces
Office machines (central)	0.25 mg/hr/m^3 of space	Large copy machines
Office machines (personal)	2.5 mg/hr/m^3 of space	Printers

[See Caveats below]

Caveats for the "Emission Rate Recommendations for Specifying Materials" Table

- Uses emission testing method specified in ASTM D5116-90 "Standard Guide for Small-Scale Environmental Chamber determinations of Organic Emissions from Indoor Materials/products".
- During the first several hours after application, many coatings have emission rates much higher than the table values.

- The recommended values do not take into account any adsorption/desorption potential of fleecy materials in the space
- Tucker assumes the indoor air is well mixed and there are at least N= 0.5 outdoor air chage per hour in the space
- This table has not been officially recognized by any standards authority (as of much 2005).

Carpet Standards

CRI/CCI (The Carpet and Rug Institute and the Canadian Carpet institute) has proposed emission standards for Carpet. These are shown in the next table.

Emission Factor Guidelines for Carpet

Emission Chemical mg/sq-meter per hr	Emission Factor at 24 hours
TVOC	0.5
Styrene	0.4
4-phenylcyclohexene (4-PC)	0.1
Formaldehyde	0.05

Source: (*airfAQS*, see References.)

Particleboard and Fiberboard

Health Canada (HC) " Exposure Guides for Residential IAQ " and ANSI 208.1.1993 suggested the following limits (Based on ASTM E1333-89 Standard Test Method for Determining HCHO Levels from Wood Products).

Emission Factor Guidelines for Particleboard and Fiberboard

		Maximum Concentration Generated in Test Method HC	ANSI
Particleboard	Action Level:	120 µg/m³ or 0.1 ppm	-
	Target Level:	60 ~~120~~ µg/m³ or 0.05 ppm	370 µg/m³ or 0.3 ppm
Fiberboard	Target Level:	180 µg/m³ or 0.15 ppm	-

Flooring	Health Canada (HC) proposed a maximum emission rate of 1.0 mg/m²-hr for vinyl and linoleum flooring when tested by ASTM D5116-90. Hardwood floors have an emission limit of 0.5 mg/m²-hr (when using the same test.)
Office panels and furniture	HC has suggested a proposed maximum airborne concentration of 0.5 mg/m³ when tested according to ASTM D5116-90 and Washington State's standard, "Environmental Chamber Protocol for Measuring Outgassing from Office Furniture."

Washington State

The State of Washington proposed standards for government buildings . The standard was written as " Maximum allowable pollutant concentration s" resulting from building products or materials. Obviously, this approach would be difficult, if not impossible , to monitor or enforce.

Maximum Allowable Pollutant Concentrations

Material	Allowable pollutant concentrations mg/cu-meter	Examples
TVOC	0.5	Varnish, paint, wax
Total Particulates	0.05	Dust from renovations
Formaldehyde	0.06	Pressed wood; fabrics
4-PC	0.0065	Carpet installation

Emissions Factor Limitations

Variability

Emission rate testing can produce a wide range of results. The following table shows test results in one study of materials in new housing. It demonstrates the need for careful, consistent testing and product-specific information.

TVOC Emission Factor Test Results for Various Materials

Material	TVOC Emission Factor, µg/m²-hr Maximum	Minimum	No. of Test Results
Carpet	650	20	7
Carpet pad	360	30	2
Vinyl Flooring	9,400	950	5
Foam Insulation	70	45	2
Composite wood product assemblies	1,380	460	4
Composite structural product assemblies	310	55	3
Low-emitting paint*	40	10	2
Conventional paint*	310	245	2

Source: SRC, as reported in *Healthy Materials*, Summer, 1995; pp 20-21. (*Paint = after 120 hours)

B : Dilution Ventilation

Favorable Conditions

Certain conditions lend themselves to dilution with outdoor air .

- Major indoor air contaminants are of relatively low toxicity
- Contaminant concentrations are not hazardous
- Smoking is not allowed in the occupied space
- Emission sources are difficult or expensive to remove or control
- Emission rates occur uniformly over time
- Emission source are widely dispersed
- Emissions do not occur close to the breathing zone of occupants
- Moderate climatic conditions prevail
- The outside air less contaminated than the inside air
- The MVAC system is capable of conditioning the dilution air required.

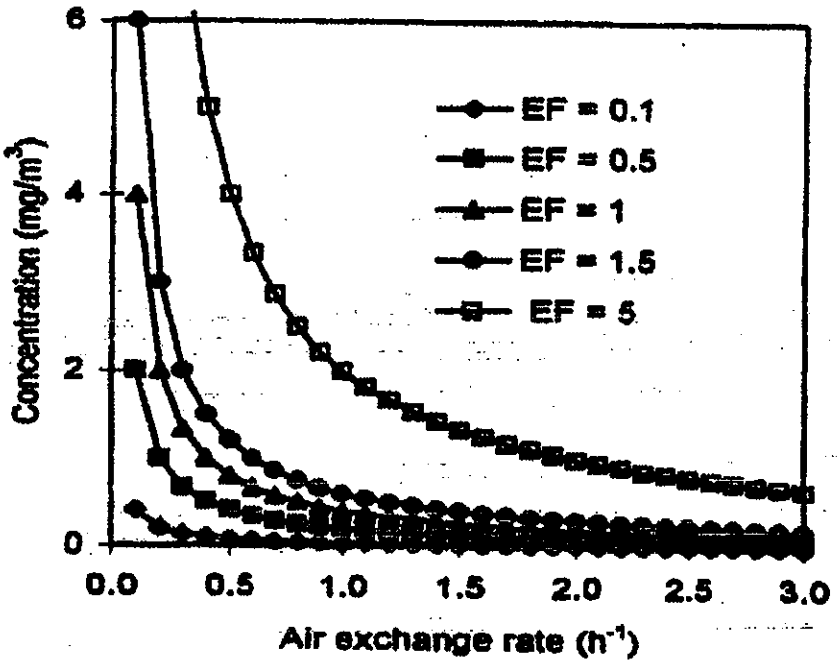
Unfavorable Conditions

This means dilution ventilation is less effective and more expensive when :

- Air contaminants are highly toxic materials
- Contaminant concentrations are hazardous
- Smoking is allowed in the occupied space
- Emission sources are easy to remove or control
- Emission sources consist of large, point sources
- People's breathing zones are in the immediate vicinity of emission sources (2- 3 feet)
- The building is located in severe climates
- The outside air is more contaminated than the inside air
- The existing MVAC system is not capable of treating the air

Determining Dilution Volume FLowrates

The figure shows concentrations for five different emission rates, all other conditions equal and at steady state conditions. The numbers represent ratios of emission rates based on typical emission factors of 0.1 to 5 mg/m^2-hr.



Note that background concentrations increase significantly at air exchange (dilution) rates less than N= 0.5 ACH (Air Change per Hour) . Also , increases above 2 ACH don't offer equivalent reductions in concentrations. This graph suggests some starting points for determining appropriate dilution ventilation rates.

ASHRAE 62 and AIHA Guideline 2

ASHRAE 62 has traditionally suggested about 15 cfm/person as the minimum dilution ventilation (OA) necessary to avoid complaints and odor problems.

AIHA Guideline 2 also recommends a minimum of 15 cfm OA per person as a minimum. These approaches suggest a fixed amount of air for design and operation of buildings. ASHRAE 62 in its "VRS" method, suggests minimum flowrates of OA for many different occupancies (e.g. 4 cfm /person plus 0.06 cfm/sq ft in typical offices).

IAOP

An alternative ASHRAE 62 standard approach is the so-called Indoor Air Quality Procedure which requires evaluation of sources, emission rates. Mixing efficiencies, and actual dilution ventilation flowrates required – the traditional IH approach for determining dilution flowrates in industrial occupancies and the one suggested by AIHA Guideline No. 2 . We will explore this 'traditional IH approach" in the next few page.

Acceptable Concentration

In the traditional IH approach, contaminated air is diluted to some " required" or "acceptable" level. This safe or comfortable level of exposure is called the Acceptable Concentration , C_a (e.g. formaldehyde at C_a = 0.05 ppm, carbon monoxide at 9 ppm , and so forth, depending on occupancy and other factors.)

Systematic Approach

To determine the appropriate rates of dilution ventilation, one should develop the following information :

1. **A profile of contaminant emission sources**
2. **An estimate of generation and emission rates**
3. **A description of the space (e.g. volume, height, air supply registers, windows, population, etc)**
4. **An acceptable concentration for exposure**
5. **An appropriate air dilution flowrate**

1. **Source**

- **Description of emission sources**
- **Chemical description of emissions to include chemical composition, size and shape, temperature**
- **Current airborne concentrations**
- **Rates of emission or evaporation over time**

2. **Emission rates**

Estimating generation and emission rates requires ingenuity, detective work, and skill. Often we must use published emission factors.

For a constant-rate emission source, no sinks, perfect mixing, and a constant air flow rate , the equation below describes the resulting equilibrium concentration , C, in a ventilated space (e.g. chamber, room, building).

$$C = \frac{E}{Q} \quad (ppm)$$

$E = emission\ rate$

$Q = Ventilation\ rate\ in\ the\ same\ unit\ as\ E$

The following equation is often used to provide an estimate for "E".

$$E = \frac{0.0241 \bullet G}{MW}$$

$E = emission\ rate\ of\ vapor\ generated, m^3 / s$

$MW = molecular\ weight\ or\ molecular\ mass$

$G = generation\ or\ generation\ rate, grams / \text{sec}$

3. **The Space**
   Obtain all of the physical parameters of the building and the occupied space – width, height, barriers, obstructions, and so forth. Building plans are often helpful.

   Occupant : Obtain a thorough understanding of the occupants:
   - Physical locations of people
   - Time considerations (How long does a person stay in a location ?)
   - Training , education , cooperativeness

4. Acceptable Concentration
   Someone must determine an acceptable level of exposure (C_a) . It isn't enough to say, "We'll do the best we can", because dilution volume flowrates cannot be estimated.
5. Dilution Rates

   The air volume flowrate required to dilute an air contimatinant can now be approximated as follows :

$$Q_{Dil} = \frac{E \bullet K_{eff} \bullet 10^6}{C_a}$$

$Q_{Dil} = volume\ flowrate\ of\ dilution\ air, m^3/s$

$E = Volume\ flowrate\ of\ vapor, m^3/s$

$C_a = the\ acceptable\ \ \exp osure\ concentration\ \ , ppm$

$K_{eff} = a\ mixing\ factor\ to\ account\ for\ incomplete\ or\ poor\ delivery\ of\ dilution\ air\ to\ occupants, unitless.$

Air Mixing

The mixing of air is called "ventilation efficiency" or "ventilation effectiveness", E_a

Mathematically, K_{eff} can be stated as :

$$K_{eff} = \frac{Actual\ Q_{OA}\ required\ to\ provide\ \min imum\ OA\ to\ occupied\ zone}{Ideal\ Q_{OA}\ required\ to\ provide\ \min imum\ OA\ to\ occupied\ zone}$$

Example 1

What is E, the volume flowrate of toluene vapor, if 1.8 kg of toluene are uniformly evaporated from fresh paint during an 8- hr period ? What volume of flowrate of outdoor air , Q_{OA} , is required for dilution to C_a = 10 ppm, if K_{eff}=1.0. (Assume STP , MW=92.1)

$$E = \frac{0.0241 \bullet G}{MW}$$

$$= \frac{0.0241 x 1800}{8x3600x92.1} = 1.635x10^{-5} m^3/S$$

$$Q_{OA} = \frac{E \bullet K_{eff} \bullet 10^6}{C_a} = \frac{1.635x10^{-5} x\ 1x10^6}{10} = 1.635\ m^3/s$$

if > Air-con flowrate of 6ACH, can dilute

Example 2

Calculate N for the flowrate determine in example 1. Assume a room volume of 850 m^3.

$$N = \frac{1.635}{850} x3600 = 7\ ACH$$

Example 3

What is the estimated volume flowrate of formaldehyde vapor formed , if new particleboard wood shelving is installed in a large office bay ?

What OA volume flowrate is required to keep background concentration s of HCHO at or below 0.05 ppm ?

(assume STP, MW = 30, 250 sq m of particleboard, C_a = 0.05 ppm, K_{eff}= 1.25)

molecular weight

From product literature

Material	HCHO Emissions, micrograms/day/sq m
Particleboard	2,000 to 25,000

Day = 24 hours = 1,440 x 60 sec

25000 µg/day/sq m can be expected to emit.

$$Grams/day = \frac{25{,}000\mu g/m^2}{1{,}000{,}000\mu g/gram} x \frac{250\ m^2}{24x60x60\sec} = 7.23x10^{-5}\ gram/\sec$$

$$E = \frac{0.0241 \bullet G}{MW} = \frac{0.0241x7.23x10^{-5}}{30} = 5.81x10^{-8}\ m^3/s$$

$$Q_{OA} = \frac{E\ x\ K_{eff} \bullet 10^6}{C_a} = \frac{5.81x10^{-8}\ x1.25x10^6}{0.05} = 1.453\ m^3/S$$

Can the existing Ahu provide 1.453 m^3/s OA , if necessary ? If not , some other solution will be required. (emission –retarding coatings, alternative shelving materials).

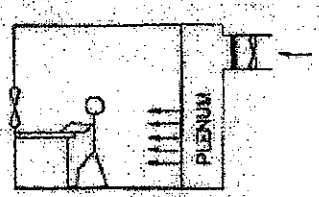
Dilution Ventilation Energy and Cost Control

Costly : Dilution ventilation can be costly and use much energy. Costs and energy consumption are a function of :

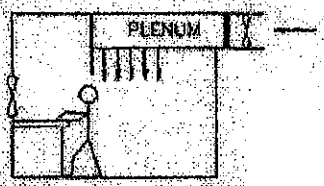
- The volume flowrate of outdoor air introduced to the building
- The temperature of the incoming air
- Fuel costs
- Type of air handler

Costs and energy usage can therefore be minimized by:

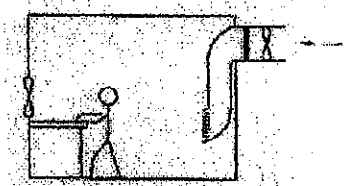
- Reducing the amount of outside air required to be conditioned
- Using low-cost fuels
- Choosing efficient air handling units (AHU)



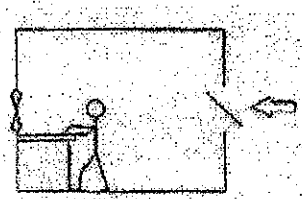
BEST AIR INLET
BEST EXHAUST
K = 1.0 MINIMUM



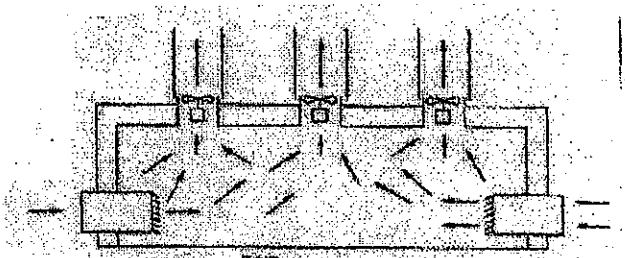
BEST AIR INLET
BEST EXHAUST
K = 1.0 MINIMUM



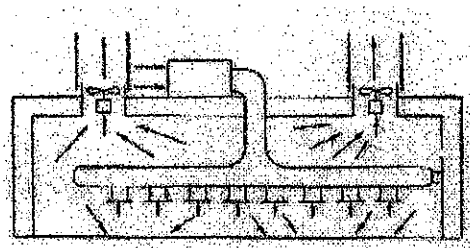
BEST AIR INLET
BEST EXHAUST
K = 1.5 MINIMUM



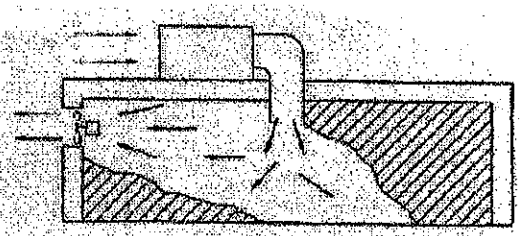
FAIR AIR INLET
BEST EXHAUST
K = 2.5 MINIMUM



FAIR
K = 2 TO 5



GOOD
K = 1.5 TO 2



POOR
K = 5 TO 10

NOTE: THE K FACTORS LISTED HERE CONSIDER ONLY THE INLET AND EXHAUST LOCATIONS AND ARE JUDGEMENTAL. TO SELECT THE K FACTOR USED IN THE EQUATION, THE NUMBER AND LOCATION OF THE EMPLOYEES, THE SOURCE OF THE CONTAMINANT, AND THE TOXICITY OF THE CONTAMINANT MUST ALSO BE CONSIDERED.

Chart No. 2

Major Consensus Standards of Good Practice

Topic	Name (often abbreviated)	Organization	Number (obtain latest)
General	Ventilation for Acceptable Indoor Air Quality	ASHRAE	ASHRAE 62 - latest
	The Ventilation Directory	NCSBCS	Summary of standards
	Various standards for HVAC (to avoid fires)	NFPA	Various (e.g., 90A)
	Guideline for HVAC Commissioning	ASHRAE	ASHRAE Guideline 1
	Fundamentals Governing the Management, Operation, Testing, and Maintenance of HVAC Systems for Maintaining Acceptable Indoor Air Quality in Employee Occupancies	AIHA	AIHA Guideline 2
Comfort	Thermal Comfort	ASHRAE	ASHRAE 55 - latest
	Comfort, Air Quality and Efficiency by Design	ACCA	ACCA Manual RS - 1997
Air filters	Filter testing methods (Arrestance/Dust Spot)	ASHRAE	ASHRAE 52.1 - latest
	Filter testing methods (MERV)	ASHRAE	ASHRAE 52.2 latest
	Test Performance of Air Filter Units	UL	ANSI/UL 900 - latest
	Air Filter Equipment	ARI	ARI 850 - latest
HVAC/ ducts	Rating air flow through outlets	ASHRAE	ASHRAE 70 - latest
	HVAC Duct/Systems Design/Applications	SMACNA	SMACNA (various)
	Guideline for the Commissioning of HVAC Systems	ASHRAE	Guideline 1-latest
	Recommended standards for duct cleaning	NADCA	Various standards
	Standards and practice for TAB	NEBB	Various standards
Air mixing	Methods for Testing Ventilation Effectiveness	ASHRAE	ASHRAE 129 - latest
Fans	Testing fan sound power	ASHRAE	ASHRAE 68 - latest
	Standards Handbook	AMCA	AMCA 99-83 + others
Humidity	Commercial and Industrial Humidifiers	ARI	ARI 640 - latest
Testing	Practices for Measurement and Testing, etc	ASHRAE	ASHRAE 111 - latest
	HVAC Testing, Adjusting, and Balancing	SMACNA	SMACNA 1993
	Method of Testing for Room Air Diffusion	ASHRAE	ASHRAE 113
Chemicals	TLVs	ACGIH	TLV Booklet
Bio-aerosols	Guidelines for the Assessment of Bioaerosols in the Indoor Environment	ACGIH	ACGIH Publication

Standards Associations

ACCA	Air Condition Contractors of America, 1712 New Hampshire Ave, Wash DC 20009
ACGIH	American Conference of Governmental Industrial Hygienists, 1330 Kemper Meadow Road, Cincinnati, Ohio 45240
AIHA	AIHA, 2700 Prosperity Drive, #250, Fairfax, VA 22031
ANSI	American National Standards Institute; Contact AIHA (ANSI Z.9 Secretariat) 2700 Prosperity Ave, Suite 250, Fairfax, VA 22031
ARI	Air Conditioning and Refrigeration Institute, 431 No. Fairfax Dr., #425, Arlington, VA 22203
ASHRAE	American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc., 1791 Tullie Circle, NE., Atlanta, GA 30329 (Best source for various other standards related to HVAC system design, operation, and maintenance.)
NADCA	National Air Duct Cleaners Association, 1518 K St NW Suite 503, Washington, DC 20005
NCSBCS	NCSBCS, 505 Huntmar Park Drive, Suite 210, Herndon, VA 22070
NEBB	National Environmental Balancing Bureau, 4201 LaFayette Center Drive, Chantilly, VA 22021
NFPA	National Fire Protection Association, One Batterymarch Park, Quincy, MA 02269
SMACNA	1385 Piccard Drive, Rockville, MD 20850
UL	Underwriter's Laboratories, 333 Pfingsten Road, Northbrook, IL 60062

Properties of Solvents, Gases, and Vapors Chart No. 4

Solvent/vapor	Molecular Wt MW	Specific Gravity (re: water)	Odor Threshold, ppm* Geo mean	Irritating	TLV** (ppm)	IAQ-LV*** (ppm)
Acetaldehyde	44.1	0.821	0.1d*	50	25	NA
Acetic Acid	60	1.049	0.1d	10	10	NA
Acetone	58.1	0.792	60d 130r	200	750	NA
Benzene	78.1	0.879	60d 100r	2,800	0.1	NA
Ethyl Acetate	88.1	0.901	20d 30r	95	400	NA
Ethyl Alcohol	46.1	0.789	100r	5,000	1,000	NA
Ethylene Glycol	62.1	1.12	na	na	50	NA
Freon TF solvent	187.4	1.57	na	na	1,000	NA
Gasoline	86	0.66	na	na	300	NA
Isopropyl alcohol	60.1	0.785	20r	200	400	NA
Kerosene	180	0.80	0.1d	15	na	NA
Methanol	32	0.792	160d 690r	17,000	200	NA
MEK	72.1	0.806	16d 17r	200	200	NA
MIBK	100.1	0.801	1d 2r	100	50	NA
Mineral spirits	>100	0.80	40d	na	na	NA
Naphtha (solvent)	90	0.7	na	na	200	NA
Perchloroethylene	165.8	1.63	50d 70r	200	25	0.7
Styrene monomer	104.1	0.903	0.15d 0.15r	100	50	0.2
Toluene	92.1	0.866	2d 10r	200	50	2
1,1,1 TC Ethane	133.4	1.31	na	na	350	NA
Trichloroethylene	131.4	1.466	80d 110r	160	50	NA
Xylene	106.2	0.881	20d 40r	na	100	NA

Gas	Molecular Wt MW	Specific Gravity (re: air)	Odor Threshold* Geo mean	Irritating(ppm)	TLV** (ppm)	IAQ-LV*** (ppm)
Air	29	1.0	-	-	na	"no odors"
Butane	58.1	2.085	2,500^	na	800	NA
Carbon Dioxide	44	1.53	-	-	5,000	700 + Bkgrnd
Carbon Monoxide	28.1	0.968	-	-	25	9
Chlorine	70.9	3.21	0.1d	3	0.5	NA
Formaldehyde	30	1.08	0.5^	1	0.3	0.1
Hydrogen Chloride	36.5	1.27	0.3^	30	5	NA
Hydrogen Sulfide	34.1	1.19	0.005r	10	10	0.1
Nitrogen Dioxide	46.0	1.448	0.1^	10	3	0.1
Ozone	48	1.658	0.008^	1	0.05	0.05
Sulfur Dioxide	64.1	2.264	0.3^	2	2	0.14

* "Odor Thresholds" adopted from Ruth and AIHA; See References. d = detection, r = recognition, ^ = lowest value reported; values rounded. Individual responses vary over several orders of magnitude. Odor thresholds are generally for single compounds in test air.

** "TLV" is ACGIH airborne limits for places of employment.

*** "IAQ-Limiting Value" is ASHRAE, WHO, Canadian, OSHA, EPA or other typical/proposed limiting *targets* for general indoor environments. See also tables on page 3-11. See also Checklist 8 on page A-27. Total VOC mixtures in excess of 5 ppm is unusual in non-industrial environments and may cause discomfort, complaints, and adverse health effects in compromised or hypersensitive individuals. "NA" not available/not applicable. All values at STP.

Emission Factors Sampler — Chart No. 6

Materials	mg/hr-sq meter	VOC	Time at testing
Med. density fiberboard	0.7-2.3	HCHO	Unknown, higher values = newer materials
Hardwood plywood paneling	0.06-1.4	"	"
Particleboard	0.08-2.0	"	"
Urea-form. foam insulation	0.05-0.8	"	"
Softwood plywood	0.01-0.03	"	"
Paper products	0.01-0.03	"	"
Clothing	0.015-0.02	"	"
Plywood	1.0	"	"new"
Room partition	0.2-1.5	"	4 hours, different brands
Room partition	0.2-1.5	"	48 hours, different brands
Silicone Caulk	13	TVOC	< 10 hours
	< 2	"	10-100 hours
Floor adhesive	220	"	< 10 hours
	< 5	"	10-100 hours
Floor wax	80	"	< 10 hours
	< 5	"	10-100 hours
Wood stain	10	"	< 10 hours
	< 0.1	"	10-100 hours
Polyurethane wood finish	9	"	< 10 hours
	< 0.1	"	10-100 hours
Floor varnish or lacquer	1	"	< 10 hours
Particle board	0.2	"	2 years old
Chipboard	0.1	"	unknown
Gypsum board	0.03	"	unknown
Wallpaper	0.1	"	unknown
Room partition	0-0.2	"	4 hours, three different brands
Latex-backed carpet	0.1-3.3	"	24 hours
	0.15	4-PC	one week old
	0.08	4-PC	two weeks old
Carpet adhesives	1-90	TVOC	24 hours, range of four sources
Carpet adhesives	0.1-12	TVOC	144 hours, range of four sources
Dry-cleaned clothes	1	PCE	0-1 day
	0.5	PCE	1-2 days

Sealants and caulks

Material	Weight loss, fraction of original sample 48 hrs	fully dry	Hours to dry completely	Primary VOC
Butyl rubber	0.05	0.18	430	Aliphatic HC
Acrylic emulsion latex	0.05	0.12	--	TVOC
Silicone	0.02	0.04	480	Xylene
Styrene butadiene rubber	0.14-0.16	0.19-0.35	100-250	Aliph HC, Xylene
Neoprene blend	0.17	0.23	100	MEK, Xylene
Oleoresin	0.01	0.04	2,000	Aliphatic HC
Polysulfide, one-part	0.01	0.06	4,900	Toluene
Acrylic solvent-based	0.03	0.14	1,000	Xylene
Asphaltic, one part	0.01	0.08	4,500	Petrol. HC
Neoprene, one part	0.18	0.33	214	Xylene
Polyurethane, one part	0.01	0.15	8,300	Xylene

Performance Characteristics of Carbon Dioxide Monitors

Carbon Dioxide Sensors[1]

Sensor Type/Initial Cost	Approach	Analysis	Pros	Cons
Mass Spectroscopy /Up to $25,000	Atomic weight, mass	In lab	Excellent accuracy at 5-10 ppm	High costs, no real time results, high tech training
Amperometric Electrochemical /$5,000	Oxidation/reduction potential	In field	Good accuracy, long battery life, fast response, datalogging	Limited sensor life, requires low RH, limited temperature range
Non-dispersive Infrared Technology /$2,500	CO2 wavelength absorption (infrared)	In field	Good accuracy, wide temp and RH range, datalogging capability	Short battery life, sensitive to pressure, dust, condensation, slow response, warmup req'd
Ion Selective Electrochemical /$600	Diffusion/pH	In field	Inexpensive	Calibration drift over time, very slow response time
Detector Tubes /$350	Chemical reaction with color stain	In field	Inexpensive, simple	Slow response, low accuracy, no datalogging, can be expensive over the long term