Comparison Study for Estimation Methods of the Required Air Volume Flow Rates for Air Conditioning of Buildings

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Abstract

A rehabilitation center of 2 floors, 35 spaces and 300 persons was used to check the merit of available methods, which may be used to calculate the supply air volume flow rates for air conditioning systems. The available methods were five, which are: - the method depending on sensible or latent space load, the method depending on effective room sensible heat load, the method which depends on specifying certain air flow rate for each calculated ton of refrigeration, the method which depends on number of air change through the space, and finally the method depending on knowledge of contaminants concentrations and generation within the space. The merit of each method was checked by determining the energy consumption, HVAC equipments size, noise level and the influence of occupation density on body odour intensity for all building spaces of ground floor. The efficiency of each method to calculate the ducting system design was calculated by determining a factor (which was used for the first time in this research) called Duct Volume Ratio (DVR). The study shows that the distribution of ducts system volume to the volumes of coffee shop and small kitchen secondary ceiling for methods 1, 2 and 3 is very good (about 0.07) since the values of DVR are acceptable.

The study of the present work revealed that the first three methods are converging in their calculated supply air volume rates results and the fourth method is slightly different but the last one is very different so that it should not be depended to calculate space load. The more precise one of these methods is the second in spite of its tedious calculations because it is comprehensive for all the effective variables, which affect the building HVAC design. The third method is the simplest and fastest method with reliable results. In addition, the method four was so effective in required mechanical ventilation volumes calculations. The method five is appropriate for knowing the minimum airflow rates, which are required to maintain specified rate of allowable contaminant concentration within the space and in the supplied incoming conditioned air.

الخلاصة

إستُخدم مركز تأهيل ذي طابقين, 35 فضاء و 300 شخص لفحص جدارة الطرق المتوفرة التي يمكن أن تُستخدم لحساب معدلات إنسياب حجم الهواء المجهّز لأنظمة تكييف الهواء. كانت الطرق المتوفرة خمس هي:- الطريقة التي تعتمد على الحمل المحسوس أو الكامن للحيّز, الطريقة المعتمدة على حساب حمل الغرفة المحسوس الفعّال, الطريقة المعتمدة على حساب معدل معيَّن من إنسياب الهواء لكل طن تبريد محسوب, الطريقة المعتمدة على عدد مرات تبديل الهواء داخل الحيز, وأخيراً الطريقة المعتمدة على معرفة معدّل تركيز الملوثات ومعدل توليدها داخل الحيز. فُحصت الجدارة بواسطة حساب إستهلاك الطاقة, حجم أجهزة التدفئة, التهوية وتكييف الهواء, مستوى الضوضاء, ودراسة تأثير كثافة إشغال الحيز على شدة رائحة الجسم لجميع فضاءات الطابق الأرضي. تم تحديد كفاءة كل طريقة في تصميم منظومة مجاري التكييف عن طريق حساب معامل (إستُخدم لأوّل مرة في هذا البحث) يُدعى معامل نسبة الحجم. بينت الدراسة بأن نسبة توزيع حجم منظومة المجاري إلى حجمي السقف الثانوي للمطبخ الصغير والمقهى للطرق 1, 2 و 3 كانت جيدة جداً (حوالي 0,07) لأن قيم **DVR** كانت مقبولة.

كشفت الدراسة الحالية بأن الطرق الثلاثة الأولى متقاربة في نتائج معدلات حجم تجهيز الهواء المحسوبة ونتائج الطريقة الرابعة تختلف عن هذه الثلاثة بصورة طفيفة ولكن نتائج الطريقة الأخيرة مختلفة جداً ولذلك لم تُعتمد هذه الطريقة في حساب حمل الحيز. الطريقة الأكثر دقة من هذه الطرق هي الطريقة الثانية على الرغم من كون حساباتها شاقة لأنها شاملة لجميع المتغيرات الفعالة التي تؤثر على تصميم منظومة التسخين, التهوية وتكييف الهواء للبناية. أبسط وأسرع طريقة للحساب وبنتائج موثوقة هي الطريقة الثالثة. كما كانت الطريقة الرابعة فعالة جداً في حساب الحجوم المطلوبة للتهوية الميكانيكية. الطريقة الخامسة ملائمة لمعرفة أقل معدَّل لتدفق الهواء المطلوب لحفظ معدّل معيَّن من تركيز الملوثات المسموح به خلال الحيز وفي الهواء المكيَّف المجهز القادم للبناية.

Nomenclature

Symbol	Meaning	Unit
ACH	The Number of Air Changes Per Hour	——
ADP	Apparatus Dew Point	$^{\circ}c$
BF	Bypass Factor	——
C_l	Air Latent Heat Factor, (3010 at sea level)	$W/(m^3{\cdot}s)$
C_e	Concentration of a Contaminant of Interest in The Entering Air	ppm
CFM	Cubic Foot per Minute	Ft^3/min
C_{space}	Average Concentration of a Contaminant within The Space	ppm
C_s	Air Sensible Heat Factor, (1.23 at sea level)	$W/(m^3{\cdot}s{\cdot}^{\circ}c)$
C_t	Air Total Heat Factor, per kJ/kg Enthalpy h (1.2 at sea level)	$W/(m^3{\cdot}s)$
ERSH	Effective Room Sensible Heat	W
DVR	Duct Volume Ratio	——
GLH	Grand Latent Heat	W
GSH	Grand Sensible Heat	W
GTH	Grand Total Heat	W
$\mathbf{N}^{\cdot}$	Contaminant Generation Rate within the Space	m/min
Q	Air Volumetric Flow Rate	m^3/s
Q_o	Outdoor Airflow Rate Required for Ventilation	m^3/s
qs, ql, qt	Sensible, Latent, Total Heat Transfer Rates	W
OALH	Outdoor Air Latent Heat	W
OASH	Outdoor Air Sensible Heat	W
T[illegible]	Outdoor Air Temperature	$^{\circ}c$
T[illegible]	Room Air Temperature	$^{\circ}c$
$\mathbf{W}_o$	Outdoor Air Humidity Ratio	kgw/kgda
W[illegible]	Room Air Humidity Ratio	kgw/kgda
Δt	Air Temperature Difference across Process	$^{\circ}c$
ΔW	Air Humidity Ratio Difference across Process	kgw/kgda
Δh	Air Enthalpy Difference across Process	kJ/kg
V	Volume	m^3

1.1 Introduction:-

Providing a comfortable and healthy indoor environment for building occupants is the primary concern of HVAC engineers. Comfort and indoor air quality (IAQ) depend on many factors, including thermal regulation, control of internal and external sources of pollutants, supply of acceptable air, removal of unacceptable air, occupants' activities and preferences, and proper operationand maintenance of building systems. Ventilation and infiltrationare only part of the acceptable indoor air quality and thermal comfort problem. Figure 1 shows a simple air-handling unit (AHU) or air handler that conditions air for a building [**ASHRAE Handbook fundamental, 2009**].

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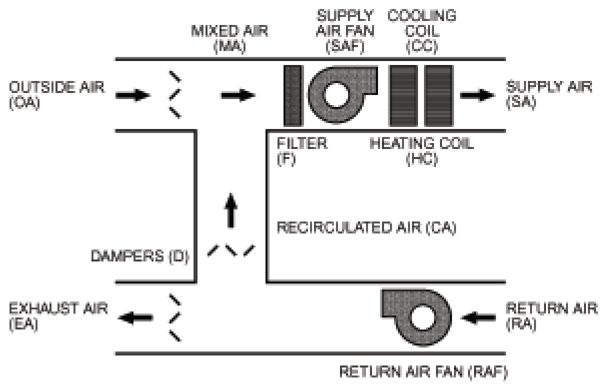

Figure (1) Simple All-Air Air-Handling Unit with Associated Airflows [ASHRAE Handbook fundamental, 2009]

All the methods, which calculate the cooling load, are investigation methods, i.e. they are not calculating the space load exactly. There are a large number of calculation methods available in Europe. These methods generally are based on different solution techniques that include simplifications of the real phenomena. According to those simplifications, they are able to consider specific or general situations. One specific situation is represented by the calculation of the maximum peak load of a single zone for convective source with the control of the air temperature [**Thermal Performance of Buildings, 2006**]. Surely, there is one of these methods is the more accurate one and closest one to the fact (i.e. the closest in calculation the exact load).

The air flow quantity supplied to a conditioned space is calculated from the heating and cooling loads on the building. This form the basis for air-conditioning design [**David V. Chatterton E. & FN SPON, 1997**]. For each load calculation method, there are many accompanied supply airflow rate calculation method, but inevitably there is one of them is the best one. This paper was prepared for checking the merit of the required airflow rate calculation method, in designing an efficient and economic airflow rates. One of the most popular load designing methods, grand total heat method, was used to estimate the required load and related flow rates and the design of the ducting layouts and sizes was also calculated and drawn (Appendices A and B respectively) for rehabilitation center of two floors, 35 spaces and 300 persons. Then, all the possible airflow rate calculation methods results will be compared with grand total heat method in a way that makes the grand total heat method is one of these methods.

1.2 Literature Survey

Mats Sandberg and Claes Blomqvist presented a quantitative estimate of the error of the decay and constant concentration method. A number of tests were carried out in an indoor test house located in the laboratory hall at the National Swedish Institute for Building Research. Apart from the accuracy of the tracer gas methods, some results of studies of the effect on the infiltration rate due to different operation modes of mechanical ventilation systems were also presented [**Mats Sandberg and Claes Blomqvist, 1985**].

C. Karakatsanis et.al determined wind pressure coefficients at various openings of a wind tower by testing a scale model of the building in a boundary layer wind tunnel. Tests were conducted on an isolated tower, the tower and adjoining house, and the tower and house surrounded by a courtyard. The wind pressure coefficients at the tower and house openings were determined at various wind angles for two types of terrain: suburban and open country. The air flow rates were then estimated from a knowledge of the wind pressure coefficients at the building apertures. If leeward openings of the tower can be closed restricting the air leaving these apertures, the air flow rate from the tower to the house can be greatly increased [**C. Karakatsanis et.al, 1986**].

Qingyan Chen et.al studied the field distributions of air velocity, temperature, contaminant concentration, and thermal comfort in an office with displacement ventilation for different air supply parameters such as the effective area, shape, and dimension of the diffuser and the turbulence intensity, flow rate, and temperature of the air supplied. The research was conducted numerically by using an airflow computer program based on a low-Reynolds-number k-ε model of turbulence. The influence of the flow rate and temperature of the air supplied is very significant on the air diffusion as well as on the thermal comfort and indoor air quality [**Qingyan et.al, 1991**]

Volume controller supplied separately from the base unit shall be capable of being field set for maximum and minimum flow (cfm). The volume controller will maintain the maximum set airflow quantity independent of variations in system static pressure.The basic moduline terminal control system adjusts the unit valve to maintain flow proportional to the room load [**Carrier Product Data, 1991**].

Because the amount of air supplied to each room is based on the load for that room, numerous indoor air quality investigations over the last decade by the National Institute for Occupational Safety & Health (NIOSH) have found that the inadequate ventilations are the source of indoor air quality problems with percent of 52% [**Fundamentals of Heating, Ventilating, and Air-Conditioning, 1992**].

All duct systems must be constructed in a manner to be considered a sealed duct with a leakage rate of less than 2 % of design cfm, measured at test pressure of 1.5 times maximum operating pressure [**Air Handling and Distribution Equipment, October 15 2000**].

With special attention to air distribution in the conditioned room, supply air temperatures as much as 15 ^{o}c less than the room air temperature have been successfully used in order to minimise the airflow rate choosing a lower supply air temperature gives a smaller airflow rate, smaller air distribution system and smaller air handling plant. Fan power will be less with lower running costs, less space will be occupied by ducts and the system will be easier to install. [**W.P. Jones, 2001**].

The relation between mass flow and volume flow depends on the density ρ, which varies quite significantly with temperature and pressure [**Jan F. Kreider, 2001**].

The supply air temperature is the temperature of the air as it leaves the air handling unit and enters the ductwork – not as it leaves the coil. As the supply air temperature is reduced, the supply air volume is reduced proportionally. That is, a 10 % increase in supply air temperature difference (space set point minus the supply air temperature) will result in a 10 % drop in required supply air volume.This allows a 10 % reduction in duct and air handler face areas, and up to a 23% reduction in supply fan motor BHP [**Optimal Air Systems – Benefits And Design Tips, July 2002**].

The airflow calculation for an underfloor system needs to be broken into two parts instead of one because of two distinct room regions that develop. For simplicity, assume that both regions are fully mixed but separated by a “stratification layer” defined as a point in the path of air above which the air never returns to the lower occupied zone. A separate energy balance equation can then be written for each of the distinct zones (occupied and unoccupied and the equations are written in the reference). The key to using these equations properly is correctly assigning the load in a room to the occupied and unoccupied zones in the [**Allan Daly, May 2002**].

In a paper submitted by **Fredrik Engdahl and Dennis Johansso**, the theory for an optimal supply air temperature was presented and HVAC energy use was calculated depending on supply air temperature control strategy, average U-value of the building envelope and two outdoor climates. The analyses showed that controlling the supply air temperature optimally results in a significantly lower HVAC energy use than with a constant supply air temperature. The optimal average U-value of the building envelope was in practice mostly zero [**Fredrik Engdahl and Dennis Johansso, 2004**]

Each degree the space temperature set point is raised, reduces the supply air volume by almost 4 %. The result is smaller ducts, air handling units and fan motors [**Optimal Air Design Manual, 2005**].

The design and accuracy of simple airflow estimators that are based on actuator control signals were investigated by **Huiling Tan and Arthur Dexter**. A computer simulation of the air-circuits of a variable-air-volume air-conditioning system was developed and validated experimentally. The simulation was used to examine the relationship between the supply airflow, the extract airflow and the inlet airflow, and the control signals for the fans and the mixing-box dampers in the air-handling unit (AHU). The results showed that the estimation errors are less than 8% of full-scale [**Huiling Tan and Arthur Dexter, 2006**].

The term overventilation refers to excessive air exchange, and the term underventilation refers to inadequate air exchange, regardless of the delivery mechanism. IAQ does depend on occupant behavior, as well as on airflows. Bathrooms and kitchen ranges are often strong pollutant sources. However, these locations are generally treated with spot exhaust ventilation. The ventilation system should be designed so these pollutants do not flow to other portions of the home [**C. Dennis Barley et.al, 2007**].

The indoor environment in the classroom was investigated experimentally using confluent jet ventilation by **T. Karimipanah et.al**. Measurements of air speed, air temperature and tracer gas concentrations had been carried out for different thermal conditions. In addition, 56 cases of CFD simulations have been carried to provide additional information on the indoor air quality and comfort conditions throughout the classroom, such as ventilation effectiveness, air exchange effectiveness, effect of flow rate, effect of radiation, effect of supply temperature, etc., and these are compared with measured data **[T. Karimipanah et.al, 2007**].

Calculated supply air volume shall be rounded off to the next 100 CFM or Liters/Second and increased by 4 % to account for the ductwork and the system component air leakage. Increase the supply air volume by an additional 5 % safety factor. Thus, the calculated supply air volume shall be increased by calculated supply air x 1.04 x 1.05 = 1.092, that is, 9.2 % more than the calculated air volume [**HVAC Design Manual, February 2008**].

Cleanroom HVAC systems can consume up to 50 times more energy than those used in commercial spaces of the same size. The airflow change rate in cleanrooms (15 to 600 ACH) is typically higher than in general purpose buildings (6 to 25 ACH). The airflow rate in cleanrooms needs to meet not only the heating and cooling loads, but also the dilution requirement to reduce room particle concentration [**Wei Sun, July 2008**].

It is not prudent to specify an air change rate for a building by policy or guideline. Instead, this decision must be based on site-specific information about the various spaces and intended uses of the areas. A more appropriate design goal is: What ventilation considerations will be necessary to enable current and future users of this space to work safely? That question may be answered only after extensive programming and inter-disciplinary collaboration among users, building designers, building operators, and environmental health and safety professionals.[**EH&S, 26/8/2008**].

The higher minimum airflow rates warrant a smaller temperature difference, to avoid excessive reheat [**Laboratory Modeling Guideline, 27 August 2008**].

The calculation of supply air quantity by using sensible heat gain depends mainly on the calculated capacity, hence proper load calculation procedures must be used for the reason that excessive sensible capacity results in short-cycling and severely degraded dehumidification performance [**ASHRAE Handbook Fundamental, 2009**].

Tao Lu et.al developed a new method for accurately quantifying ventilation rates (i.e. space air change rate) and CO2 generation rates from measured CO2 concentrations for individual spaces. The proposed method firstly determined space air change rate using Maximum Likelihood Estimation (MLE). Additionally, a novel coupled-method was initiated for further estimating CO2 generation rates. Both simulated and experimental data were used to validate the model. Experiments were conducted in a school office by measuring indoor CO2 concentrations and pressure differences between the return air vent and space. Excellent agreement was obtained. The estimated numbers of occupants were same as the actual ones. Furthermore, the predicted space air change rates showed great consistencies with those from CO2 equilibrium analysis. The model was simple, handy and effective for practical use. Moreover, the model was also capable for dealing with time-varying space air change rates [**Tao Lu et.al, 2010**].

The capacity of a single air-handling unit shall not exceed 50,000 cfm (23,600 L/s) [**HVAC Design Manual, March, 2011**].

1.3 Objective of the present work

There is no any study established to review and compare the methods, which may be used to estimate the required supply flow rate for any space during design of HVAC systems. The aim of this work is to list all the available methods, which may be used to calculate the supply air volume flow rates for air conditioning systems, and then to make comparison study between these methods to discover the features and drawbacks of each one.

2. Modeling of Air Volume Flow Rate Calculation Methods

For proper desig of the distribution system, the conditioned airflow required by each room must be known. The required airflow rate for each space can be calculated by using one of the following five methods:-

I. Using Sensible or Latent Space Load:- Common air-conditioning processes involve transferring heat via air transport or leakage. The sensible, latent, and total heat conveyed by air on a volumetric basis is

$q_s = C_s Q \Delta t$ ------------ (1)

$q_l = C_l Q \Delta W$ ------------ (2)

$q_t = C_t Q \Delta h$ ------------ (3)

$q_t = q_s + q_l$------------ (4)

The heat factors Cs, Cl, and Ct are elevation dependent. The sea level values in the preceding definitions are appropriate for elevations up to about 300 m. Procedures are provided in Chapter 18 of ref.[**ASHRAE Handbook Fundamental, 2009**] for calculating adjusted values for higher elevations. The required air flow rates of the ground and 1st floors rehabilitation center regions (shown in appendices A and B respectively) according to this method were tabulated in tables (1) and (2) respectively.

Table (1) Air Flow Rates of Ground Floor Regions According to Method I.

Space(s)	Name	No. of Rooms	No. of Persons	qs (kW)	ql (kW)	Q (m^3/s)
1	**Small Kitchen**	1	5	8.661	2.771	0.698557878
2	**Coffee shop**	1	15	13.045	3.865	1.052151891
3	**(Secretary + Manager +Meetings) Rooms**	1	5	6.0827	1.054	0.490603626
4-6	**Accountant room = Small Storage-room = Workers room**	3	4	4.047	1.17	0.326413086
7-8	**Waiting Rooms**	2	4	3.584	1.038	0.289069557
9	**Long Passage**	1	2	2.97713	0.58643	0.240122113
10	**Around Circular Passage**	1	5	6.699	1.8361	0.540311653
11-18	**Physiotherapy Room**	8	6	33.64	5.8016	2.713253323

Table (2) Air flow Rates of 1st Floor Regions According to Method I.

Space(s)	Name	No. of Rooms	No. of Persons	qs (kW)	ql (kW)	Q (m^3/s)
19	**Small Kitchen**	1	5	9.468	2.771	0.763646922
20	**Cafeteria**	1	20	20.7892	8.1126	1.676764744
21	**Main Entrance Hall with Dome**	1	10	8.2326	1.2563	0.664005033
22	**Large Storage Room**	1	6	4.205	0.7252	0.339156665
23-30	**Workers = small storage-room = lockers men/women and Physiotherapy Rooms**	8	4	15.76	3.12	1.271131759
31-32	**Waiting Rooms**	2	4	5.1875	1.038	0.418400761
33	**Long Passage**	1	2	3.01332	0.48773	0.243041038
34	**Around Circular Passage**	1	5	11.548	1.8361	0.931410505
35	**Fitness Hall**	1	30	19.6246	11.763	1.582833269

II. Using Effective Room Sensible Heat Load:-

This method is used when the calculated ventilation rate is so large compared to the infiltration rate. The procedure of this method will be written according to ref.[**C. Arora, 1981**]. Airflow rates relying on this procedure will be calculated as follows:-

$$Q = \frac{ERSH}{1.22(T_R - ADP)(1 - BF)} \quad \text{------------ (5)}$$

$$ERSH = qs + BF\ OASH \quad \text{------------ (6)}$$

$$OASH = 1.22\ Q_o(T_o - T_R) \quad \text{------------ (7)}$$

It is appropriate to mention that the total load according to this method is calculated according to the following equation:-

$$GTH = GSH + GLH \quad \text{------------ (8)}$$

In which:-

$$GSH = qs + OASH \quad \text{-------------- (9)}$$

$$GLH = ql + OALH \quad \text{-------------- (10)}$$

$$OALH = 2940\ Q_o(W_o - W_R) \quad \text{-------------- (11)}$$

For rehabilitation center and according to method II; the airflow rate of each space is inserted in table (3) for ground floor and table (4) for first floor.

Table (3) Air Flow Rates of Ground Floor Regions According to Method II .

Space(s)	Q (m^3/s)
1	0.676112412
2	1.018345043
3	0.474839969
4-6	0.315925059
7-8	0.279781421
9	0.232406714
10	0.52295082
11-18	2.62607338

Table (4) Air flow Rates of 1st Floor Regions According to Method II.

Space(s)	Q (m^3/s)
19	0.73911007
20	1.622888368
21	0.642669789
22	0.328259173
23-30	1.230288837
31-32	0.404957065
33	0.23523185
34	0.901483216
35	1.53197502

III. Depending on Experience and Making a Certain Airflow Rate Against Each Ton of Refrigeration of The Total Calculated Space Load:-

The minimum airflow for cooling is 300 cfm/nominal ton. Operation Air Quantity Limits table on page 7 for minimum airflow cfm for heating [**Carrier Product Data, 2003**].

System air flow should be between 400 (680 m^3/hr) and 425 cfm (722.5 m^3/hr) per ton of cooling for dry climates, and between 350 (595 m^3/hr) to 400 cfm per ton for humid climates. Dry climates are defined as those with 20 inches or less of annual rainfall, and where the evaporator coil rarely removes much moisture from the air. These are also referred to as dry-coil climates. Humid climates, or wet-coil climates, have more than 20 inches of annual rainfall [**Armin Rudd, 2006**]. These limitations are valid only if the AHU and supply air ductwork are both located within the conditioned space, else the cfm per each ton will be minimized.

Reducing the supply air flow rate through a cooling system (e.g., reducing the fan motor speed) will increase its dehumidification performance. This will typically result in a lower energy efficiency ratio (EER), but the efficiency decrease may be modest in some cases depending on the supply air fan and compressor characteristics. Operating at too low of a supply air flow rate could cause coil icing and/or sweating ductwork (i.e., moisture condensing on the outside of the ductwork due to cold surface temperatures). Placing ductwork and the air handler in conditioned space or increasing duct insulation levels can help alleviate the sweating ductwork issue, but the potential for coil icing remains so a lower limit on supply air flow rate is required. On the other hand, reducing the air flow rate across the cooling coil lowers the sensible heat ratio of the cooling equipment, and the lower SHR means the unit will remove more moisture from the air when it operates. In addition, the lower sensible capacity means the system will run a little longer to achieve the same dry-bulb temperature set point resulting in additional dehumidification [**Don B. Shirey, 2008]**.

Since Iraq has wet climate in summer [**Iraqi Meteorological Directorate, 2011**], hence 400 cfm (0.189 m^3/sec) per ton will be took as a guide to complete the rehabilitation airflow calculations according to this mehod and the results were inserted in tables (5) and (6) for ground and first floor respectively.

Table (5) Air Flow Rates of Ground Floor Regions According to Method III.

Space(s)	Q (m^3/s)
1	0.632321458
2	0.935318041
3	0.394741825
4-6	0.288560273
7-8	0.255649911
9	0.19710597
10	0.472089475
11-18	2.181575403

Table (6) Air flow Rates of 1st Floor Regions According to Method III.

Space(s)	Q (m^3/s)
19	0.676957866
20	1.598602896
21	0.524845616
22	0.272696925
23-30	1.044281763
31-32	0.344341955
33	0.193648446
34	0.740295103
35	1.736096307

IV. Knowing The Number of Air Change (ACH) for The Relevant Space:-

The required airflow rate can be calculated according to the ACH, as follows [**David V. Chatterton E. & FN SPON, 1997**]:-

Q= ACH / hour x V m^3x 1 hour / 3600 s ------------ (12)

Calculating the supply air volume is based on the recommended air changes per hour by the applicable code or criteria and the project-specific parameters, such as, ceiling height and air distribution mode. The minimum supply air volume is generally maintained at 4 air changes per hour. To account for the internal heat gain due to lighting, equipment load, solar and other envelope heat gain, the supply air volume can also be estimated by assuming 10.0 F [5.6 C]edmm difference between the space and ambient air temperatures [**HVAC Design Manual, 2011**].

. Normally the ACH value varies from 0.5 ACH for tight and well-sealed buildings to about 2.0 for loose and poorly sealed buildings. For modern buildings, the ACH value may be as low as 0.2 ACH. Thus depending upon the age and condition of the building an appropriate ACH value has to be chose, using which the infiltration rate can be calculated [**IIT Kharagpur, 2005**].

Air changes per hour is the volume of air that needs to be replaced with fresh air every hour.One air change occurs in a room when a quantity of air equal to the volume of the room is supplied and/or exhausted. Air change rates are units of ventilation that compare the amount of air moving through a space to the volume of the space. Air change rates are calculated to determine how well a space is ventilated compared to published standards, codes, or recommendations. Air changes per hour (ACH) is the most common unit used. This is the volume of air (usually expressed in cubic feet) exhausted or supplied every hour divided by the room volume (also usually expressed in cubic feet). Airflow is usually measured in cubic feet per minute (CFM). This is multiplied by 60 minutes to determine the volume of air delivered per hour (in cubic feet).

A room may have two airflow values, one for supply and another for exhaust. (The airflow difference between these two values is called the offset.) To calculate the air change rate, use the greater of the two airflow values. For AIIRs, the exhaust should be greater than the supply [**HVAC Design Manual, 2011**]

For rehabilitation center and according to method IV, the airflow rate of each space was inserted in table (7) for ground floor and table (8) for first floor.

Table (7) Air Flow Rates of Ground Floor RegionsAccording to Method IV.

Space(s)	Volume (m^3)	ACH	Q (m^3/s)
1	75.6	15	0.315
2	213.66	10	0.5935
3	62.9412, 89.604, 35.2821	4,4,10	0.267501
4-6	49.506, 56.202, 51.033	6	0.261235
7-8	150.6	4	0.16733
9	116.166	6	0.19361
10	431.493	8	0.95887
11-18	83.727	6	0.139545

Table (8) Air flow Rates of 1st Floor RegionsAccording to Method IV.

Space(s)	Volume (m^3)	ACH	Q (m^3/s)
19	75.6	15	0.315
20	244.491	13	0.88288
21	127.388	6	0.21231
22	108.985	5	0.15137
23-30	56.676, 51.861,53.76	6	0.270495
31-32	150.6	4	0.16733
33	116.166	6	0.19361
34	449.379	8	0.99862
35	465.09	8	1.03353

V. Knowing average concentration of a contaminant and the rate of contaminant generation within a space:-

The basic equation for contaminant concentration in a space is obtained by making a balance on the concentrations entering and leaving the conditioned space assuming complete mixing, a uniform rate of generation of the contaminant, and uniform concentration of the contaminant within the space and in the entering air. All balances should be on a mass basis; however, if are assumed constant, then volume flow rates may be used. For the steady state case [**Faye C. McQuiston, et.al, 2005**] :-

$$Q = \frac{N^{\cdot}}{C_{space} - C_e} \quad \text{------------ (13)}$$

Assume that each person in the building breath out carbon dioxide at the rate 0.3 L/min, and the concentration of CO_2 in the incoming ventilation air is 300 ppm (0.03 percent for each space of the building and let the desire is to hold the CO_2 concentration in each space below 1000 ppm (0.1 percent)[**Stanley A. Mumma, 2004**]. The air flow rates of rehabilitation center spaces were inserted in table (7) for ground floor and table (8) for first floor, where these rates are calculated according to method 5.

Table (9) Air Flow Rates of Ground Floor Regions According to Method V.

Space(s)	No. of Rooms	No. of Persons	Q (m^3/s)
1	1	5	0.035714286
2	1	15	0.107142857
3	1	5	0.035714286
4-6	3	4	0.028571429
7-8	1	4	0.028571429
9	1	2	0.014285714
10	1	5	0.035714286
11-18	8	6	0.042857143

Table (10) Air flow Rates of 1st Floor Regions According to Method V.

Space(s)	No. of Rooms	No. of Persons	Q (m^3/s)
19	1	5	0.035714286
20	1	20	0.142857143
21	1	10	0.071428571
22	1	6	0.042857143
23-30	8	4	0.028571429
31-32	1	4	0.028571429
33	1	2	0.014285714
34	1	5	0.035714286
35	1	30	0.214285714

The ducting system design efficiency for each one of calculation methods can be checked for any space by a factor named by the present work as "Duct Volume Ratio" and denoted as "**DVR**" which can be calculated as:-

$$DVR = \frac{\text{The volume of Ducting System in The Space}}{\text{The volume of the Secondary Ceiling in that Space}} \quad \text{------------ (14)}$$

3. Results and Discussion:-

The values of air flow rates for two floors regions of rehabilitation center according to five estimation methods, were sumerized in tables (11) and (12) respectively.

Table (11) Summary of Air Flow Rates Calculation Methods for Ground Floor Regions.

Method / Space(s)	I	II	III	IV	V
1	0.698557878	0.676112412	0.632321458	0.315	0.035714286
2	1.052151891	1.018345043	0.935318041	0.5935	0.107142857
3	0.490604	0.47484	0.394742	0.267501	0.035714
4-6	0.326413	0.315925	0.28856	0.261235	0.028571
7-8	0.289069557	0.279781421	0.255649911	0.16733	0.028571429
9	0.240122113	0.232406714	0.19710597	0.19361	0.014285714
10	0.540311653	0.52295082	0.472089475	0.95887	0.035714286
11-18	2.713253323	2.62607338	2.181575403	0.139545	0.042857143

Table (12) Summary of Air Flow Rates Calculation Methods for First Floor Regions.

Method / Space(s)	I	II	III	IV	V
19	0.763646922	0.73911007	0.676957866	0.315	0.035714286
20	1.676764744	1.622888368	1.598602896	0.88288	0.142857143
21	0.664005033	0.642669789	0.524845616	0.21231	0.071428571
22	0.339156665	0.328259173	0.272696925	0.15137	0.042857143
23-30	1.271131759	1.230288837	1.044281763	0.270495	0.028571429
31-32	0.418400761	0.404957065	0.344341955	0.16733	0.028571429
33	0.243041038	0.23523185	0.193648446	0.19361	0.014285714
34	0.931410505	0.901483216	0.740295103	0.99862	0.035714286
35	1.582833269	1.53197502	1.736096307	1.03353	0.214285714

From results of these tables and from review of mathematical model for all methods, it is clear that the best method for Ac system design is the second because it takes in the account all the factors which can affect the building load like building envelope, persons, …etc, but the calculations of this method are tedious and the designer can replace this method efficiently by method one when there is no importance for ventilation. Besides, the method one is considered as an effective method for the teaching purposes unlike method two which is an excellent designing method inspite of it's hard working calculations .

Methods 3 and 4 are commercial methods and the load calculation is easy and of good acceptance. Method 3 is most efficient than 4 in designing air flow rates required volumes for AC systems but method 4 is more efficient than 3 in ventilation rates requirements design especially for mechanical ventilation of spaces like bathrooms.

The air flow rates of method five are so small because it take care of contaminants only regardles of the other factors, hence it is appropriate for preliminary estimation of the minimum required air flow rates which valid for maintaining specified contaminant level and this is the main cause of undepending this method in determining the space load calculation.

The load calculated for two floors of rehabilitation center according to all methods are plotted in figures (1) and (2) respectively. Since method 5 should not be depended to calculate space load, therefore it will not entered in the present load comparison. These two figures show that, the maximum load is calculated when method 2 is depended and the minimum load is calculated when method 4 is used and the load calculated by depending on the other methods is ranging between these two values. The load values give a good impression about HVAC equipment size and energy consumption, where the larger space load is the larger HVAC equipment size and larger energy consumption; hence the equipments which are calculated relying on the method 2 calculations are the larger and those calculated relying on the method 4 are the smalest in size.

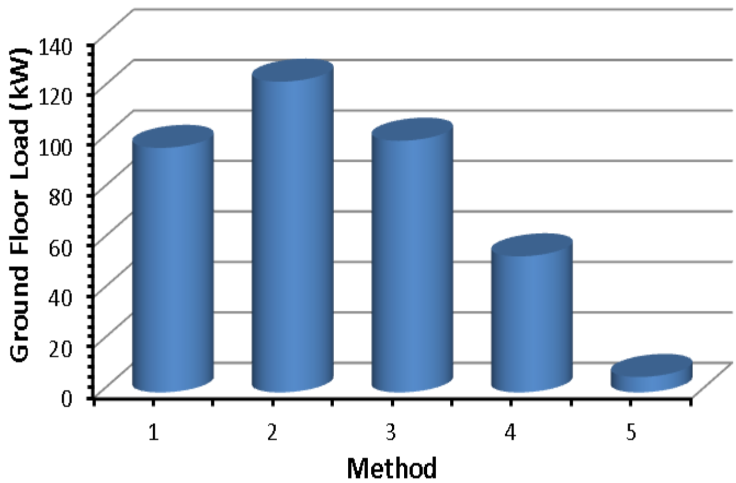

Figure (1):- Ground Floor Loads Comparison (kW)

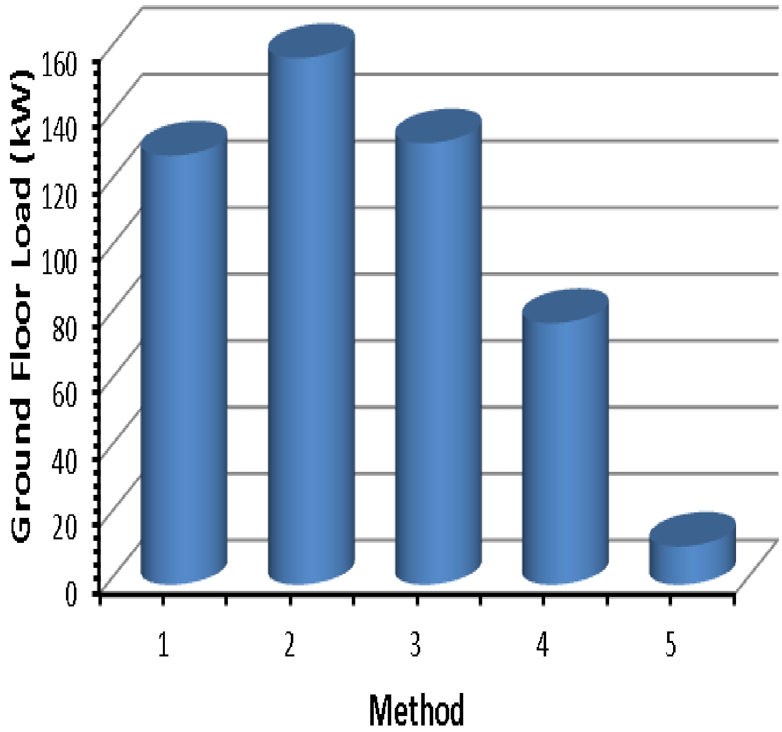

Figure (2):- First Floor Loads Comparison (kW)

The sound levels for two floors of rehabilitation center according all methods are tabulated in tables (13) and (14) respectively. The sound levels calculations are estimated according to website (www.gammaline.com). These two tables show that all methods can result a good noise level of air flow rates except method five which have no values for sound levels since there is no available values for the range of volume flow rates of this method.

Table (13) Sound Levels (dB) for Ground Floor Regions according to All Calculation Methods.

Method / Space(s)	I	II	III	IV	V
1	25	25	35	20	——
2	30	30	35	20	——
3	35	30	25	30	——
4-6	35	25	20	30	——
7-8	30	30	25	30	——
9	30	30	20	20	——
10	25	25	35	30	——
11-18	30	35	25	20	——

Table (14) Sound Levels (dB) for 1st Floor Regions according to All Calculation Methods.

Method / Space(s)	I	II	III	IV	V
19	40	35	25	25	——
20	35	35	35	25	20
21	25	20	40	35	——
22	25	25	35	25	——
23-30	40	25	35	20	——
31-32	30	30	25	40	——
33	40	40	30	35	——
34	25	35	25	40	——
35	25	25	25	30	20

The DVR for coffee shop and small kitchen of the ground floor was calculated and drawn as a function of each one of the five estimation methods in figure (3) which shows that the distribution of ducts system volume to the volumes of coffee shop and small kitchen secondary ceiling for methods 1, 2 and 3 is very good (about 0.07) since the values of DVR are acceptable.The very high values for DVR are not desirable nor very small one since the very high values means that the plenum is full by the duct system which means high cost , very difficult maintainance, and hard noise attenuation while the very small values indicates that the ducts system is clumsy. Many tests and calculations for many different types of buildings should be checked to decide the very high and very low values ranges of DVR. The decision of considering the DVR of methods 1, 2 and 3 was very good relied on the inspection of load values and noise levels of these methods as discussed before.

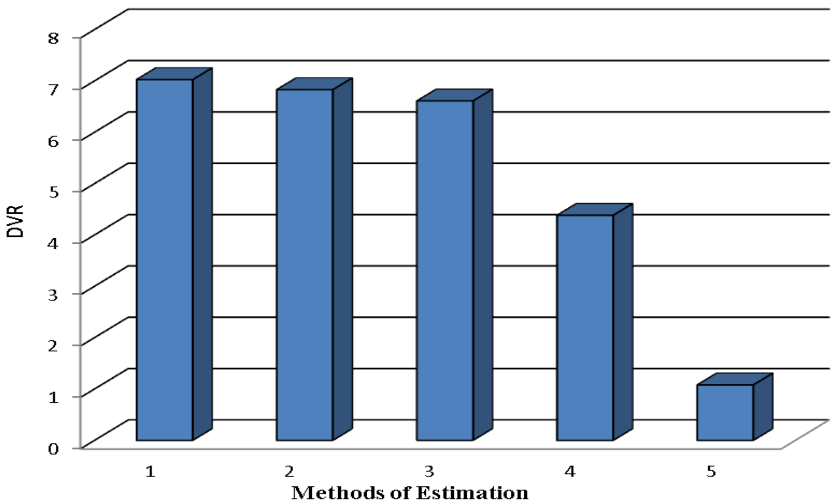

Figure (3) DVR for Coffee Shop and Small Kitchen of Ground Floor.

The occupation density is an important parameter affecting odour intensity [**H.B. Awbi.,1991**]. Figures (4) shows the influence of occapation density on body odour intensity for spaces of 1st floor according to five estimation methods. From this figure, one can see that for very small density values, the supply flow rates values are very high which means that when the persons' number increases, the required supply flow rate will also increases to maintain an acceptable concentration of body odour and the supply flow rate is inversly proportional to the occupation density for all methods. It is also noted that the method of minimum air flow rates ranges has the upper curve and that of maximum has the lower one and this express that the air flow rates of method of upper curve shoulde be larger to dillute the undesired body odour .

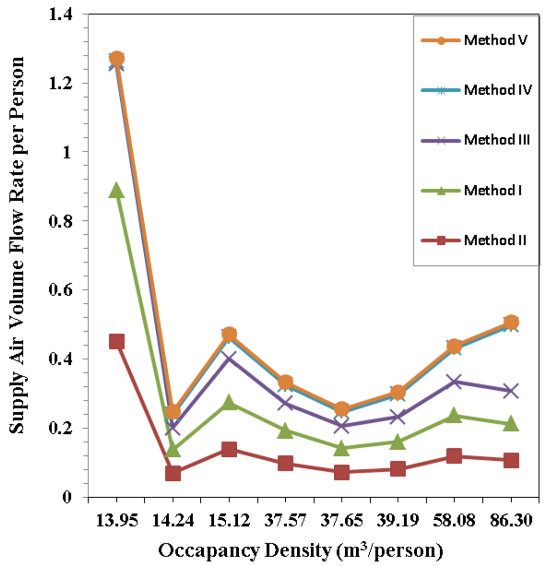

Figure (4) Influence of Occapation Density on Body Odor Intensity for Spaces of 1st Floor.

The table (15) shows the scenario of comparison for all methods.

Table (15) All Methods Comparison Scenario.

Merit Factor / Method	**Energy Consumption and Equipment Size**	**Noise Level**	**DVR (%)**	**Needing for Air Flow Rate to Dilute Body Odor Intensity**
I	Large energy consumption and equ. size	Good	7.024534156	Low
II	The Largest energy consumption and equ. size	Excellen t	6.828034467	Very low
III	Large energy consumption and equ. size	Good	6.609445006	medium
IV	medium energy consumption and eq u. size	Good	4.381666796	Very high
V	The smallest energy consumption and equ. size	——	1.078543957	Very high

4. Conclusions:-

1. The best efficient, more reliable method to estimate the required air volume flow rates is the second method, since it is comprehensive for all the parameters, which can affect the occupants comfort but the calculations of this method are tedious.

2. The second method can be substituted effectively with the first method when the infiltration rate is enough to meet the building ventilation rate, hence the required rate of building ventilation should be calculated and compared with the available infiltration to decide which method should be selected.

3. The fifth method cannot be depended as reliable method to calculate air volume flow rates rather than it can check the minimum allowable airflow rates which maintain the specified rate of contaminants. This method can be more efficient if it is used to calculate airflow rates by depending on different contaminants and then taking the highest airflow rate result of these contaminants.

4. The airflow rate can be estimated confidently by depending on method three merely by knowing the required space load and inversely it can calculate this load by knowing the estimated airflow rate. Attention should be taken to determine precisely the climate state whence it is dry or wet.

5. The fourth method calculations based on the knowing of the space volume and the required times of air change through this space to replace the room air volume by fresh outside air, hence this method should not be used to calculate the volume flow rates for spaces which are neighboring to contaminant sources like factories, sewage treatment stations,…etc.

6. Method five should not be depended to calculate space load.

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