# Part (I)
# Design of Central Air Conditioning System for Surman Medicine Faculty -Building (A)

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## ABSTRACT

In recent years, air-conditioning systems are widely used in residential, commercial and industry buildings. The purpose of the system involves comfortable environment of the occupants in terms of desired temperature, relative humidity, indoor air quality, airflow, air filtration, ventilation, maintain a certain level of noise, and other environmental for the occupants, equipments as well as to save energy. The aim of this study is to design a central air conditioning system for cooling purpose of Surman Medicine faculty Building. It has two floors; the building divided to three zones (A, B, and C). So the inlet and outlet designing temperature and the inside and outside relative humidity are selected. The main objective of this research is to calculate the total cooling loads that the building receives through walls, roofs, windows, occupants, and equipments using cooling load temperature differences (CLTD) method. The calculation of the parameters such as supply air, flow rates, return air, and the capacity of the required cooling coil of each system are presented. The calculation is made to design air supply ducts, return air ducts, and pressure losses, so according to this the ducts are drawn in detail to every part of the system to demonstrate how the air distribution system delivers the proper amount of conditioned air to a space. The appropriate air-conditioning systems such as (Chillers and Pumps) can be selected according to the cooling coil loads and the air distribution design. For simplification of theoretical calculations an Excel program was developed which is useful for calculating the total cooling loads of the building and the other parameters.

**Keywords:** Central air-conditioning, cooling loads, air distribution, cooling coil, Chillers, Excel program.

## 1. Introduction

Air conditioning systems have been used in many parts of institution in the world. The purpose of most systems is to provide thermal comfort and an acceptable indoor air quality for occupants. With the environment of standard of living occupants require more comfortable and healthful indoor environment. Most countries are experienced with increasing in the average summer temperatures every year and consequently the energy needs to provide air-conditioning is also increasing annually. The air-conditioning industry is required to providing energy efficient technologies to balance this growing demand with a minimum impact on global warming and ozone depletion. The air-conditioning system efficiency is very important as it determines the amount of energy that is being consumed for heating and cooling. Many countries are creating minimum efficiency grades, to ensure that the HVAC industry continually strives towards the development of more efficient systems, thereby reducing the demands of energy. The increasing buying power of consumers globally is also generating a large demand for the development of air conditioning systems that provide a higher level of comfort than that provided by the standard fixed capacity systems. The comfort air-conditioning systems are divided into three groups:

**1) Summer-Air Conditioning System:**

The main objective of this system is to reduce the sensible heat (cooling air) and to reduce water vapor content of the air by cooling and dehumidifying.

**2) Winter Air Conditioning System:**

The main objective of this system is to increase the sensible heat and water vapor content of air by heating and humidification. The addition of water vapor to the air is termed as a humidification of air.

**3) Year Round Air Conditioning System:**

This system assures the control of temperature and humidity of air in an enclosed surface throughout of the year when the atmospheric conditions are changing as per season. In most of the A/C applications for industry the common problem is to control the temperatures, humidity and air motion for maintaining the quantity of the product to perform a specific industrial process, successfully [1].

Air conditioning systems are classified into different types of air conditioners for various environmental applications. Air conditioning systems can be modified such as window air conditioner, central air conditioner, packet air conditioner, and split air conditioner. The main objective of this study is to design a central Air Conditioning System for cooling purpose of Surman Medicine faculty Building. The calculation of the parameters such as total cooling loads, supply air, flow rates, return air, and the capacity of the required cooling coil of system are presented. The calculation is also made to design air supply ducts, return air ducts, and pressure losses, so according to this the appropriate air-conditioning systems can be selected.

## 2. Materials and Methods

The Air conditioning system is designed as per the latest American Society of Heating, Air Conditioning and Refrigeration Engineers Standards (ASHRAE) Handbook of Fundamentals. Medicine faculty Building is located in Surman town of Libya Country on N 32º latitude on the Mediterranean Sea. It has two floors; the building divided to three zones (A, B, and C) and three central air-conditioning systems were designed for each zone. In this paper a central air conditioning system was designed for zone A. however; central air conditioning system for zones B and C will be designed in another paper. To design the Air conditioning systems the following steps should be considered, first step is to select inside and outside design conditions, the recommendation inside design conditions for comfort in auditorium allowing for present practice is given as: dry bulb temperature $T_R$ = 24ºC, relative humidity 45%. The outside design conditions were chosen as: dry bulb temperature $T_o$= 40ºC, relative humidity 65% and the daily range of external temperature change is 11ºC. Second step is to measure various dimensions of the space which is to be cooled, including that of walls and roofs, and find out the heat gained by them. We will also consider the number of windows, type of windows, blinds and their exposure to sun and accordingly decide on the heat gained by them. The heat emitted by lights, air filtration, ventilation and other electrical appliances is also considered. One of the most important parameters to consider is the number of people that will occupy the room or the office. After measuring the total amount of heat generated in the room or office per hour, the HVAC designer will calculate the capacity of the required cooling coil of the system. The calculation to design air supply ducts, return air ducts, and pressure losses are considered. The HVAC designer will suggest the HVAC system of proper tonnage for comfort in room or office without excessive burden of electricity bills [2].

## 3. Theory and Calculation

Most air conditioning systems operate at their design loads for only a small part of their life and it follows, therefore, that the designer should be concerned not only with the maximum heat gains and cooling loads but also with the way these change throughout the day and over the year. The main factor to determine the time of maximum space cooling load is the direction of the wall and window also the space activity. Peak time was chosen to obtain maximum heat gain for each room and was selected at 2 pm (14:00 hours) for all walls and roofs of these rooms [3].

### 3.1 Heat gain through exterior structure

Solar radiation forms the greatest single factor of cooling load in buildings. Radiant energy from the sun is absorbed by the room materials, both the structure and furnishings.

#### 3.1.1 Conduction heat gain through exterior structure

The conduction heat gain through the exterior roof, walls and glass are each determined by the following equations [2]:-

For walls, roof, and glass

$$Q = U \times A \times (CLTD)_C \quad (1)$$

Q = net room conduction heat gain through roof, wall or glass, KW

U = overall heat transfer coefficient for roof, wall or glass, W/Kg ºC

A = area of wall, roof or glass, m²

$(CLTD)_C$ = corrected value of cooling load temperature difference ºC.

For wall: $(CLTD)_C = (CLTD + LM) \times K + (25.5 - T_R) + (T_o - 29.4)$ (2)

For roof: $(CLTD)_C = [(CLTD + LM) \times K + (25.5 - T_R) + (T_o - 29.4)] \times F$ (3)

For glass: $(CLTD)_C = CLTD + (25.5 - T_R) + (T_o - 29.4)$ (4)

LM = correction factor for latitude and month

K= color adjustment factor

F = Ventilation factor

#### 3.1.2 Heat gain by solar radiation through glass

The net heat gain by solar radiation through glass can be calculated by the following equation:-

$$Q = (SHG)_{Max} \times A \times SC \times CLF \quad (5)$$

$(SHG)_{Max}$= maximum solar heat gain factor, w/m²

A = area of glass, m²

SC = shading coefficient

CLF = cooling load factor for glass

#### 3.1.3 Transmission gain through interior structure

The heat flows from interior unconditioned spaces to the conditioned spaces can be calculated by the following equation:-

$$Q = U \times A \times (T_b - T_R) \quad (6)$$

Where: $T_b = T_R + \frac{2}{3} \times (T_O - T_R)$

(7) $T_b$ = temperature of unconditioned spaces

#### 3.1.4 Heat transfer due to infiltration

Heat transfer due to infiltration consists of both sensible as well as latent components, the equations for determining the sensible and latent loads are as follow [3].

$Q_s = 1.22 \times \dot{V}(T_O - T_R)$

(8)

$Q_L = 2940 \times \dot{V}(W_O - W_R)$

(9)

$Q_s$,$Q_L$ = sensible and latent cooling load from infiltration air, W

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

$W_O$, $W_R$ = specific humidity for outside and inside conditions

The infiltration rate is obtained by using either the air change method or the crack method, in this paper the infiltration rate will be calculated by air change method which is given by:

$\dot{V} = \frac{nV}{3600}$

(10)

Where:

n is the number of air changes per hour and V is the gross volume of the conditioned space in $m^3$

**3.1.5 Heat gain due to Ventilation Air**

Ventilation air is the amount of outdoor air required to makeup for air leaving the space due to equipment exhaust, infiltration as required to maintain Indoor air quality for the occupants. The heat is usually added to the air stream before the cooling coil and has no direct impact on the space conditions. The additional cooling coil load is calculated as follows [4]:

$Q_s = 1.22 \times \dot{V}(T_O - T_R)$

(11)

$Q_L = 2940 \times \dot{V}(W_O - W_R)$

(12)

$\dot{V} = N_p \times q$

(13)

Where

$\dot{V}$= Ventilation airflow rate.

$N_p$ = number of person

q = the amount of air each person needs to ventilate.

**3.2 Estimation of internal loads**

The internal loads consist of load due to occupants, due to lighting, due to equipment and appliances and due to products stored or processes being performed in the conditioned space [5].

**3.2.1 Load due to occupants:**

The internal cooling load due to occupants consists of both sensible and latent heat components. The rate at which the sensible and latent heat transfer take place depends mainly on the population and activity level of the occupants. So the sensible heat transfer to the conditioned space due to the occupants is given by the equation:

$Q_{ps} = q_s \times CLF \times N_p$

(14)

$Q_{pl} = q_l \times N_p$

(15)

$q_s$, $q_l$ = sensible and latent heat gain per person, w

CLF = cooling load factor by hour of occupancy

**3.2.2 Load due to lighting:**

Lighting adds sensible heat to the conditioned space. Since the heat transferred from the lighting system consists of both radiation and convection, a Cooling Load Factor is used to account for the time lag. Thus the cooling load due to lighting system is given by:

$Q_{light} = N \times W \times F_U \times CLF$

(16)

Where

W = Watt input from electrical lighting

$F_U$= Lighting use factor

CLF = cooling load factor for lighting

N = number of lighting

**3.2.3 Internal loads due to equipment and appliances:**

The equipment and appliances -electrical, gas, or steam used in the conditioned space may add both sensible as well as latent loads to the conditioned space. Thus the internal sensible load due to equipment and appliances is given by [6]:

$Q_{eqs} = q_s \times N_{eq}$

(17)

$Q_{eql} = q_l \times N_{eq}$

(18)

$q_s$, $q_l$ = sensible and latent heat gain per equipment, W

$N_{eq}$ = number of the equipment

**3.3 Estimation of the cooling coil capacity of the system:**

The required cooling coil capacity of the system can be obtained by summing up the total of the sensible and latent loads due to the building ($Q_{cs}$), leakage losses in the return and supply air ducts ($Q_{return\_supply}$), and heat added due to return air fan ($Q_{return\_airfan}$).

$Q_{cc} = Q_{cs} + Q_{return\_supply} + Q_{return\_airfan}$ (19)

One can also calculate the sensible and latent for the coil and draw the process line on the psychometric chart and find the required cooling coil capacity as following [7]:

$Q_{cc} = \dot{m}_M \times (h_M - h_{s'})$

(20)

Where

$\dot{m}_M$ = mixture mass flow rate

$h_M$, $h_{s'}$ = Enthalpy of mixture air and Enthalpy of air leaving the cooling coil

**3.4 Supply air and return air calculation**

In general, air distribution systems should be designed to supply air to the lower levels of the plant and return air from the upper levels. This will allow the warm air to pre-cool as it flows over exposed cold surfaces, such as pipes, discharge lines and penstocks. Calculation for Supply air and return air flow rate to a space is based only on the total space sensible heat load as following:

$\dot{V}_s = \frac{Q_s}{1.22 \times (T_R - T_s)}$ the rate variable, $\dot{V}_s$ , need to be written this way: $\dot{V}_s = \frac{Q_s}{1.22 \times (T_R - T_s)}$

(21)

$$\dot{V}_R = \frac{Q_R}{1.22 \times (T_R - T_S)}$$

(22)

Where:

$\dot{V}_S, \dot{V}_R$= Supply air and return air flow rate in cubic meter per second $\dot{V}_S$ , $\dot{V}_R$

$Q_S, Q_R$ = total sensible heat gain for supply air and return air

$T_S$ = supply air dry bulb temperature

### 3.5 Air ducts design

There are three methods of designing an air conditioning duct system: velocity method, equal friction method and static regain method. In this paper the equal friction method was selected for sizing duct. It is simple and is most widely used conventional method. This method usually yields a better design than the velocity method as most of the available pressure drop is dissipated as friction in the duct runs, rather than in the balancing dampers. The total pressure drop in each duct run is obtained by summing up the frictional and dynamic losses of that run as following [7]:

$$\Delta P = {}^{\Delta P_f}/_{L} \times \left(\sum L\right) + \sum Pd$$

(23) $\Delta P_f/L$ = frictional pressure drop per unit length

$\sum L$ = summation of the main duct length

$\sum Pd$ = dynamic pressure drop

For Rectangular Ducts [9]:

$$D_{eq} = \frac{1.3 * (H * W)^{0.625}}{(H + W)^{0..250}}$$

(24)

H = Height of the duct

W = Width of the duct

$D_{eq}$ = circular equivalent of rectangular duct

$$V = \frac{Q}{A}$$

(25)

$$A = W * H$$ [10]

(26)

### Results and Discussion

Table (1) summarizes the total heat gain, total return heat gain, Supply air and return air flow rate each room of the building (A)

| Room Number | Room Name | $Q_T$ KW | $Q_R$ KW | $\dot{V}_s$ m³/s | $\dot{V}_R$ m³/s |
|---|---|---|---|---|---|
| 1-1 | Office | 12.014 | 8.579 | 0.820 | 0.586 |
| 1-2 | Office | 4.065 | 2.953 | 0.277 | 0.201 |
| 1-3 | Office | 3.735 | 2.623 | 0.255 | 0.179 |
| 1-4 | Coffee | 10.979 | 6.497 | 0.749 | 0.443 |
| 1-5 | Office | 6.074 | 3.939 | 0.414 | 0.269 |
| 1-6 | Office | 5.435 | 3.301 | 0.371 | 0.225 |
| 1-7 | Office | 5.435 | 3.301 | 0.371 | 0.225 |
| 1-8 | Office | 4.193 | 3.125 | 0.286 | 0.213 |
| 1-9 | Pass | 6.672 | 5.284 | 0.455 | 0.361 |
| 1-10 | Entrance | 50.1107 | 22.955 | 3.422 | 1.567 |
| 1-11 | Office | 3.177 | 2.146 | 0.217 | 0.146 |
| 1-12 | Room | 2.898 | 1.918 | 0.198 | 0.131 |
| 1-13 | Room | 19.355 | 10.818 | 1.322 | 0.738 |
| 1-14 | Room | 18.777 | 10.240 | 1.282 | 0.699 |
| 1-15 | Room | 19.355 | 10.818 | 1.322 | 0.738 |
| 1-16 | Room | 18.777 | 10.240 | 1.282 | 0.699 |
| 1-17 | Pass | 53.674 | 25.074 | 3.666 | 1.712 |
| 1-18 | Room | 19.355 | 10.818 | 1.322 | 0.738 |
| 1-19 | store | 3.771 | 2.520 | 0.257 | 0.172 |

Table (2) shows Ducting Measurement, velocity, air flow rate of the main duct (2A)

| **Section No** | **W (m)** | **H (m)** | **Deq (m)** | **V (m/s)** | **Q (m³/s)** |
|---|---|---|---|---|---|
| 2A | 1.3 | 0.75 | 1.069 | 18.756 | 18.288 |

Table (3) shows Ducting Measurement, velocity, air flow rate for section (A-I)

| Section No | W (m) | H (m) | $D_{eq}$ (m) | V (m/s) | Q ($m^3/s$) |
|---|---|---|---|---|---|
| 1 | 0.8 | 0.55 | 0.722 | 17.213 | 7.574 |
| 3 | 0.35 | 0.275 | 0.339 | 7.781 | 0.749 |
| 5 | 0.75 | 0.55 | 0.700 | 16.545 | 6.825 |
| 7 | 0.25 | 0.175 | 0.228 | 5.828 | 0.255 |
| 9 | 0.75 | 0.55 | 0.700 | 15.927 | 6.570 |
| 11 | 0.25 | 0.175 | 0.228 | 6.331 | 0.277 |
| 13 | 0.75 | 0.55 | 0.700 | 15.255 | 6.293 |
| 15 | 0.225 | 0.2 | 0.232 | 6.355 | 0.286 |
| 17 | 0.75 | 0.55 | 0.700 | 14.562 | 6.007 |
| 19 | 0.4 | 0.275 | 0.361 | 12.036 | 1.324 |
| 21 | 0.65 | 0.5 | 0.622 | 14.409 | 4.683 |
| 23 | 0.35 | 0.25 | 0.322 | 12.617 | 1.104 |
| 25 | 0.55 | 0.5 | 0.573 | 13.014 | 3.579 |
| 27 | 0.35 | 0.25 | 0.322 | 12.617 | 1.104 |
| 29 | 0.45 | 0.45 | 0.492 | 12.222 | 2.475 |
| 31 | 0.4 | 0.275 | 0.361 | 12.036 | 1.324 |
| 33 | 0.35 | 0.275 | 0.339 | 11.927 | 1.148 |

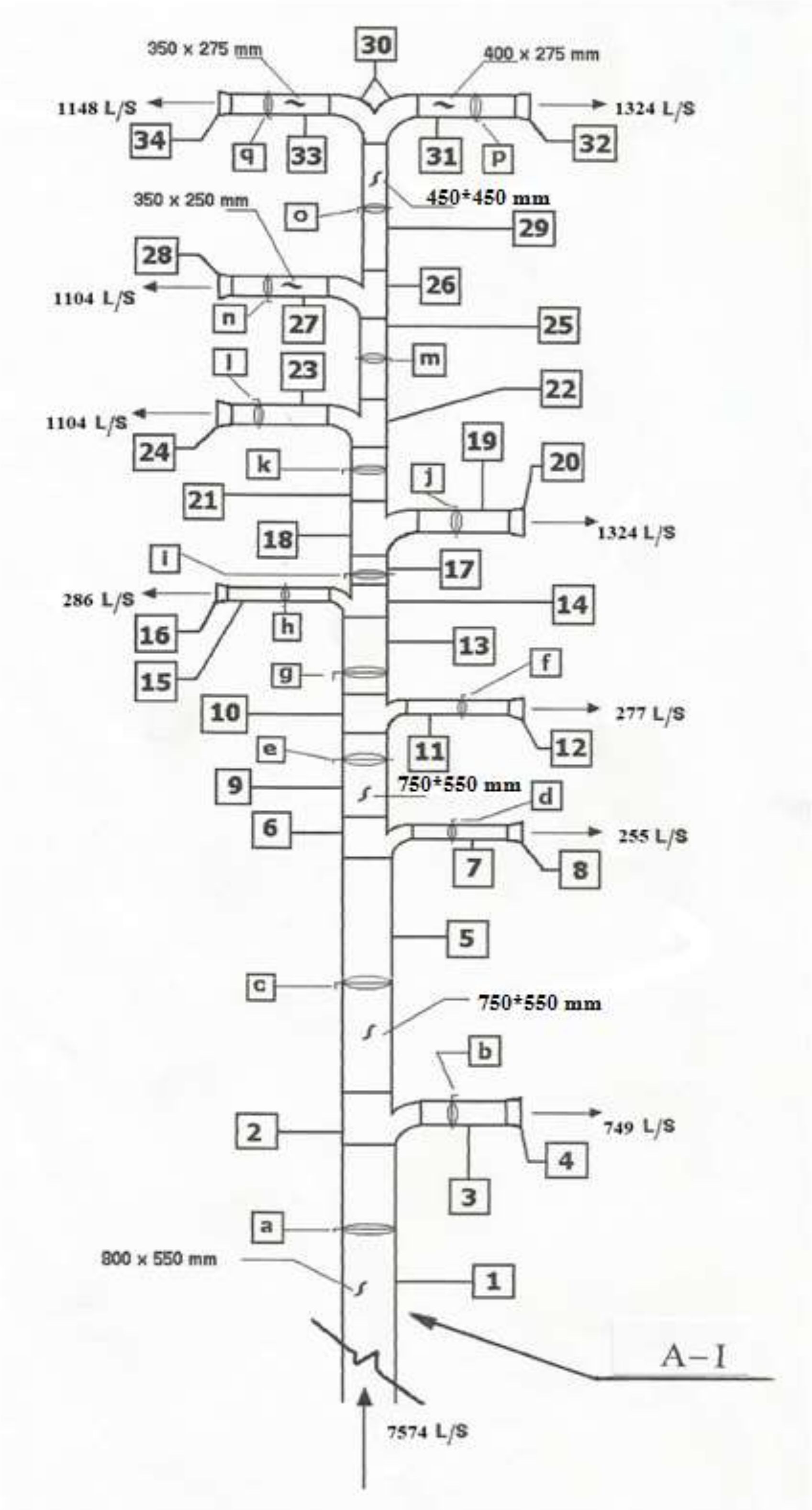


Figure (1) shows duct layout for section A-I

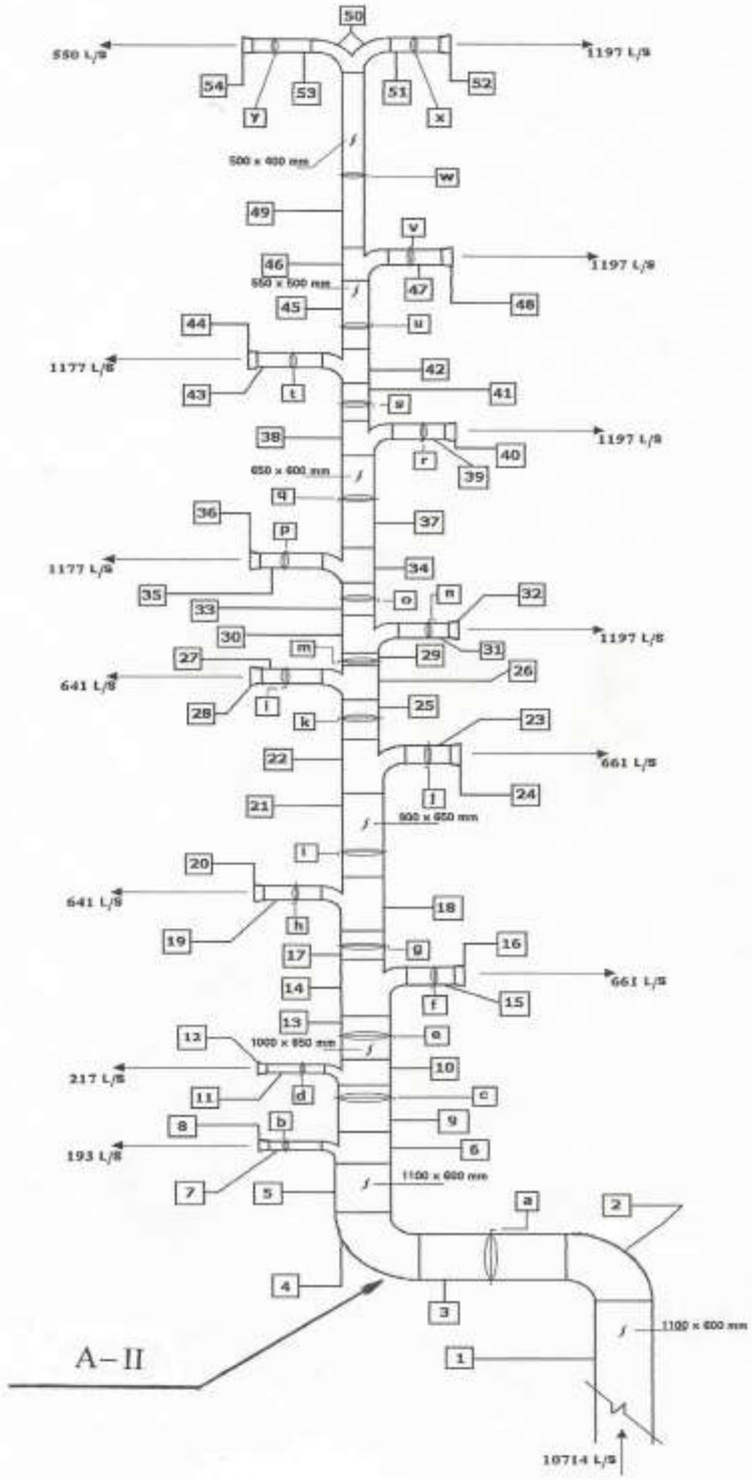

Figure (2) shows duct layout for section A-II

Table (4) shows Ducting Measurement, velocity, air flow rate for section (A-II)

| Section No | W (m) | H (m) | $D_{eq}$ (m) | V (m/s) | Q ($m^3/s$) |
|---|---|---|---|---|---|
| 1 | 1.100 | 0.600 | 0.878 | 16.233 | 10.714 |
| 7 | 0.225 | 0.150 | 0.200 | 5.718 | 0.193 |
| 9 | 1.100 | 0.600 | 0.878 | 15.933 | 10.516 |
| 11 | 0.225 | 0.175 | 0.216 | 5.511 | 0.217 |
| 13 | 1.000 | 0.650 | 0.876 | 15.844 | 10.299 |
| 15 | 0.450 | 0.200 | 0.321 | 7.344 | 0.661 |
| 17 | 1.000 | 0.650 | 0.876 | 14.827 | 9.638 |
| 19 | 0.400 | 0.200 | 0.305 | 8.012 | 0.641 |
| 21 | 0.900 | 0.650 | 0.833 | 15.377 | 8.996 |
| 23 | 0.450 | 0.200 | 0.321 | 7.344 | 0.661 |
| 25 | 0.800 | 0.700 | 0.818 | 14.883 | 8.335 |
| 27 | 0.400 | 0.200 | 0.305 | 8.012 | 0.641 |
| 29 | 0.750 | 0.700 | 0.792 | 14.655 | 7.694 |
| 31 | 0.400 | 0.350 | 0.409 | 8.55 | 1.197 |
| 33 | 0.700 | 0.650 | 0.737 | 14.279 | 6.497 |
| 35 | 0.400 | 0.350 | 0.409 | 8.407 | 1.177 |
| 37 | 0.650 | 0.600 | 0.683 | 13.638 | 5.319 |
| 39 | 0.400 | 0.350 | 0.409 | 8.55 | 1.197 |
| 41 | 0.600 | 0.550 | 0.628 | 12.490 | 4.122 |
| 43 | 0.400 | 0.350 | 0.409 | 8.407 | 1.177 |
| **Section No** | **W (m)** | **H (m)** | **$D_{eq}$ (m)** | **V (m/s)** | **Q ($m^3/s$)** |
| 45 | 0.550 | 0.500 | 0.573 | 10.705 | 2.944 |
| 47 | 0.400 | 0.350 | 0.409 | 8.55 | 1.197 |
| 49 | 0.500 | 0.400 | 0.488 | 8.735 | 1.747 |
| 51 | 0.400 | 0.350 | 0.409 | 8.55 | 1.197 |
| 53 | 0.350 | 0.200 | 0.286 | 7.857 | 0.550 |

## Conclusions

The total cooling load for the building depends on external and internal design condition, overall heat transfer, temperature difference, and total heat transfer area. For this system total estimated cooling load required for the building is 60.105 tons and the capacity of cooling coil is 67.551 tons. The Supply air flow rate for this system is 18.288 $m^3$; however the return air flow rate is 10.051 $m^3$. According to the cooling coil loads and the air distributions design so the appropriate air-conditioning systems such as (Chillers and Pumps) can be selected. Type of equipment chosen is TRANE packaged rooftop air conditioner which is integrated unit consisting of the evaporator, condenser, compressor, cooling tower and working performance is illustrated. The total pressure drop which is calculated for the system is 1078.533 Pa, so the power of the fan for the system which is selected is 35.793 KW (48 hP). The supply air flow rates for the corridors and entrance were added to the rooms farther away from the system to compensate for the loss of pressure; however the corridors are adapted through the return air where the air returns from the rooms through the doors to the corridors.

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