RESEARCH ARTICLE

EXPERIMENTAL STUDY TO THE PERFORMANCE OF THE DISPLACEMENT VENTILATION UNDER DIFFERENT OPERATION CONDITIONS.

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Manuscript Info	*Abstract*
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The present study presents the experimental study of performance of the displacement ventilation . The experimental study included the effect of different internal loads of (1600,1300and 1000)W , different supplied air temperatures of (18,20,22and 24)°C and different supplied air velocities (0.75,1,1.25 and 1.45)m/s on the performance of displacement ventilation . The experimental results show that the cooling capacity of air increases as the temperature of supplied air decreases. At a increases in temperature of supplied air by (33.33)%,the cooling capacity of air decreases by (23.52, 24.05 and 20.63)% for internal load of (1600,1300 and 1000)W respectively .While as supplied air velocity increases ,the cooling capacity of air increases .At supplied air increases by (93.33)% ,the cooling capacity increase by (33.61, 39.23 and 36.8)% for internal load of (1600, 1300 and 1000)W respectively.

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Introduction:-

To enhance air quality and lowering energy consumption displacement ventilation is used . For many years displacement ventilation has been used in industrial buildings with high thermal loads . Also it has been used in non industrial buildings since mid1e 80's , especially in the Scandinavian countries. All over the world attention is focused on displacement ventilation increase at the recent years. Temperature effectiveness and the ventilation effectiveness could be enhanced by using displacement ventilation . [1] **Nielsen [2]** described a temperature distribution model in a room with displacement ventilation .The various types of heat sources in occupied zone were used ,characteristic Archimedes number of flow in the account, have been used with full scale experimental in the developed new model. Measurements show that the distribution of temperature can be linear function of elevation of room. The work show that the air flow , heat load and type of heat source had strongly affected the vertical temperature distribution. **Chen et. al . [3]** compared between efficiency of displacement and mixing ventilation experimentally . The study achieved in environmental chamber used as patient ward with one person that dimensions are (4.90m(L)×4.32m(W)×2.72m(H)) .Also it is equipped with well insulation on the walls ,ceiling and floor and supplying with different air flow rate ,temperature and humidifies. The experimental results show that worse air quality produced by displacement ventilation .When auxiliary exhaust is placed in lower part of restroom door compared with mixing ventilation . when auxiliary exhaust of displacement ventilation at upper part . **da Graça and Mateus (2017),[4]** assesses the thermal performance and indoor air quality of displacement ventilation by measuring the CO_2 in two occupied large rooms and used these measurement to validate three-node DV model

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implemented in Energy Plus. The results show that the DV achieved thermal performance in both rooms with high efficiency for the removal of CO_2.

Experimental Work:-
In experimental work DV unite was made that used as a cooling system, installed at a room with dimensions of 3m(H)x4m(W)x3m(L) located at fourth floor on surface of Mechanical Engineering Department building , Al Mustansiriya University , Collage of Engineering .Its location is at south-east corner where both south and east walls are exposed to a direct solar radiation while north and west walls are internal walls. The north wall nearby conditioned space while the west wall nearby unconditioned space . There are two windows one of them has dimensions ,1.4m(w)x1.5m(H), on south wall and another has dimensions , 1.85m(w) x 1.5 m (H) locate on east wall . Both windows have single glass with iron frame. All four walls are of 30 cm thickness and consist of following layers from inside to outside , Juss plaster 20mm , Cement plaster 20 mm, Common Brick 240 mm , cement plaster 20mm . Figure (1) show the diagram sketch of the room. The component of DV as shown in figure (2) consist of air washer box that sides are closed by aluminum plates and then all sides of the box are insulated by cork of 0.04 W/m. K thermal conductivity[5] The assembly is a box which a centrifugal fan mounted . The centrifugal fan that used for draw, supply of circulation air . This type of fan is used because of it is more suitable for an application which need low to medium air flow rate with low pressure and it is more appropriate for HVAC an application [6]. The box content a filter of air, cooling coil , heater with temperature controller to the adjust temperature of the entering air , circular diffuser with angle 45° since it has high induction (mixing of supply and room air) [7], and two ducts one for return and other for supply air. The chosen of this duct is done taking in to account its roughness that refer to low friction loss ,thermal conductivity and easy for installation due to its flexibility and cost . The hot air drawn from the room by the centrifugal of DV through the return duct with a circular shape having a diameter of 0.31m located at a ceiling level then the return hot air is mixed with different percentages of fresh air that is controlled by using the gates . The mixed air passes through the cooling coil to cool it then passes through the heater to reach the required temperature by the temperature controller . The mixed air is introduced to the room at a floor level by a circular duct with 0.31m diameter which has a diffuser at its outlet with 45° angle as shown in figure (2). The velocity of introduced air is measure by using the hot wire anemometry. The first part of experimental tests the air supply at different velocities (0.75,1,1.25 and1.45)m/s each with constant temperature. The required air velocities were achieved by controlling the percentages of mixing air (return and fresh) through two gates one for return air to vent return air , that is square shape with 20cm (L) x20 cm (W) .The other gate for entering fresh air from ambient that is circle shape with 30cm diameter. The second part of experimental tests is for DV air introduced to the space room with different temperature (18,20,22and 24)°C at constant velocity . As mentioned above the control on the temperature of air is done using by the heater and the temperature controller with different internal loads.

Mathematical Model:-
To design the DV unite the following parameters have been studies
The capacity of DV system is determined by the following equation:
$$Q_{air}=m \cdot Cp (T_r - T_s) \qquad \qquad \text{...(1)}$$
The required mass flow rate calculate from equation :
$$m = \rho \cdot A.V \qquad \qquad \text{... (2)}$$
The area of supplied duct is :
$$A=\frac{\pi}{4}.D^2 \qquad \qquad \text{... (3)}$$

To ensure the efficiency of DV used by validation if DV achieve the thermal condition in occupant zone. In this work PMV-PPD method is used to study the thermal conditions achieved in occupant zone from the following equations[8] :
Heat balance equation for a body :
$$M-W=H+E+ C_{res} +E_{res} \qquad \qquad \text{... (4)}$$

Comfortable equation :
$$M-W=H +E_c+ C_{res} +E_{res} \qquad \qquad \text{... (5)}$$

For determining E ,E_c, C_{res} and E_{res} from following equations:
$$E=3.05 \times 10^{-3} (256 \times t_{sk}^- - 3373 - pa)+E_{sw} \qquad \qquad \text{... (6)}$$

$E_c = 3.05 \times 10^{-3} \times [5733 - 6.99(M-W) - pa] + 0.42 \times (M-W-58.15) \qquad .. \ (7)$

$C_{res} = 0.0014 \times M \times (34 - t_a) \qquad \qquad \qquad \dots \ (8)$

$E_{res} = 1.72 \times 10^{-5} \times M \times (5867 - pa) \qquad \qquad \dots (9)$

E_{sw} and t_{sk} ,in the heat balance equation have to be measured from the external work (W) can in most cases it can be set equal to zero . The met value and clothing level values taken from special tables [9]. The H values is either measured directly using dry heat loss transducer or calculated from the equations .

PMV equation is :

$PMV = (0.303 e^{-0.036M} + 0.028) . [(M-W) - H - E_c - C_{res} - E_{res}] \quad \dots \ (10)$

PPD equation is:

$PPD = 100 - 95 e^{-(0.03353.PMV4 + 0.2179.PMV2)} \qquad \qquad \dots . \ (11)$

Because of complexity of the solution for above equations , a new method is used in this work using the CBE tool produce by Building Energy Center for the calculations of thermal comfortable according to the ASHREA standard 55 -2013 as shown in figure (3) . At the lift hand which represent the input date as illustrated in figure (3) while at the right side give the results of PMV and PPD also tell you if system achieve comfortable conditions or not . This tool presented two methods of comfortable condition which are PMV method and adaptive method. So firstly the selected method and then input the required data[9]. The required input parameters of CBE tool is temperatures , relative humidity and velocity of air in the room .These are obtained from experimental measurement , also metabolic rate , mean radiant temperature and clothing level were needed . Since no device is available to measure the mean radiant temperature and its calculation consumption time, so in previous studies there was a try to avoid its calculation as possible . As a result in this study assume the mean radiant temperature is the same as the air temperature under indoor conditions [10].

Results and Discussion:-

The relationship between the velocity of the supply air and average room air temperature (represented by the average of three temperatures measured at different locations (10,110 and 170) cm for different internal loads is illustrated in figure (4). It can be noticed that the average air room temperature decreases as mass flow rates of the supply air increases for each internal load .This is due to the increase cooling capacity of the supply air with an increase it's velocity. At internal load (1600,1300 and 1000)W, the air room temperature decreasing from (29.13to26)°C, (28.4 to 24.4)°C and (26.96 to 23.45)°C respectively for velocity of supply air increase from (0.75to 1.45) m/s for internal load of (1600,1300 and 1000)W respectively . Also it's clear that for supply air velocity, the average temperature of air room increase as internal load increase due to decreasing cooling capacity of supply air with internal load increase .For internal load increase from (1600 to 1000)W , the average air room temperature decreases from(29.13 to 26) °C for supply air velocity of 0.75m/s.

Figure (5) shows the relationship between the velocity of supply air and vertical temperature difference(VATD) calculated between the levels (10 and 110)cm for different internal loads. It's clear as mass flow rate increases the VATD decreases. This is because of high mixing between air layers as velocity of air increases for each internal load .For internal load (1600,1300 and 1000)W, the VATD decreasing from (3 to 1.12)°C,(2.5 to 0.7)°C and (2 to 0.35)°C respectively for air velocity increase from (0.75 to 1.45)m/s . Also we can notice that as internal load increases ,VATD increases for each air supply velocity due to the low mixing between air layer at high internal load .For internal load increases from (1000 to 1600)W, the VATD increase from (2 to 3)°C for 0.75 m/s supply air velocity.

Figure (6) describes the relationship between the velocity of supply air and vertical temperature difference calculated between levels (10 and 170)cm for different internal load. It's clear that as mass flow rate increases the VATD decreases due to the high mixing between air layers as velocity of air increases. For each internal load ,for internal load (1600,1300 and 1000)W, the VATD decreases from (4 to 1.12)°C,(3.5 to 0.8)°C and (3.1 to 0.4)°C respectively for air velocity increase from (0.75 to 1.45)m/s . Also we can notice that as internal load increases , the VATD increases for each supply velocity due to the low mixing between air layers at high internal load .For internal load increases from (1000 to 1600)W, the VATD increase from (3.1 to 4)°C for 0.75 m/s supply air velocity.

Figure (7) shows the relationship between supply air velocity and air capacity at constant inlet temperature of supply air for different internal load. It can be noticed when the mass flow rate of supply air increases, the cooling capacity of air increases for different internal load. At internal load of (1600 ,1300 and 1000)W, the cooling capacity

increases from (929.42to 1400) , from (790 to 1300) and from (630 to 1000)W respectively for supply air velocity increase from (0.75 to 1.45)m/s . This is attributed to the Re number increase as a result of heat transfer coefficient increase leading to heat transfer by convection increases. Also can notice from figure that as internal load increases , the air capacity increases due to the of increase temperature of return air because at higher internal load the air room temperature is rises. For velocity of 0.75 m/s , the air capacity increase from (600 to 929.42) W when internal load increases from (1000 to 1600)W respectively.

Figure (8) shows the relationship between supply air velocity and PMV for different internal load. It can be noticed that as mass flow rate increases the PMV decreases for different internal load. At load (1600 ,1300 and 1000) W, the PMV decreases from (1.18to - 0.06), from (0.97 to -0.50) and from (0.54 to -0.76)respectively for air supply velocity increase from (0.75 to 1.45)m/s . The causes of decreasing PMV is the meaning approach to neutral sensation. The PMV decreases due to the increase thermal load removed by air .As mass flow rate of air increase this leads to decreasing air room temperature and approach to comfortable conditions . Also notice for different mass flow rates ,the PMV increases as internal load increases, due to the increase air room temperature with an increase internal load. For supply air velocity of 0.75m/s , the PMV increases from 0.54to 1.18 for internal load increase from 1000 to 1600 respectively.

The relationship between air mass flow rates and PPD for different internal load has been shown in figure (9). It can be noticed for different internal load ,as mass flow rates increase the PPD decreases. For internal load of 1600W , the PPD decreases from (34 to 5)% at a velocity of supply air increase from (0.75 to 1.45)m/s .Also at internal load of 1300W, the PPD decreases from (25 to 5) % for supply air velocity increase from(0.75 to 1.25)m/s while at air velocity of 1.45 m/s , the PPD increases to 10% , which means at this velocity the feeling discomfort increases . For internal load of 1000W , PPD decreases from (11 to 6)% at air velocity increase from(0.75 to 1) m/s while at air velocity increase from(1.25 to 1.45)m/s ,the PPD values increases from (8 to 17)%,which means that at velocity more than 1m/s , the number of people feeling discomfort increases .

Figure (10), shows the relationship between supply air temperature   and average   temperature of air room for different internal load and constant supply of air flow rates. It's clear that as supply air temperature increases, the air room temperature increases for different internal load. For internal load of (1600,1300 and 1000) W , the air room temperature increases from (29.13to32)°C , from (28.4 to 31)°C and from (27.7 to 30)°C respectively at supply air temperature increase from (18 to 24)°C ,that is due to the decreasing in air cooling capacity with increased supply air temperature. Also can be noticed that as internal load increase the air room temperature increases too for different supply air temperature due to higher thermal energy with high internal load .For internal load increase from (1000 to 1600)W, the average air temperature increase from (27.7 to 29.13)°C respectively at supply air temperature is 18 °C.

The relationship between temperature of supply air and VATD calculated between level (10 and 110)cm for different internal load is illustrate in figure (11). It's can be noticed that as the temperature of supply air   increases the VATD increases at constant mass flow rates of supply air   due to low mixing between air layers as velocity of air is constant for each internal load .For internal load of (1600,1300 and 1000)W, the VATD increasing from (3 to3.6)°C,(2.5 to 3.4)°C and (2 to 3.1)°C respectively ,for air temperature increase from (18 to 24) °C. Also notice that VATD increases as internal load increases for each air supply temperature because of low mixing between air layers at high internal load .For internal load increase from (1000 to 1600)W, the VATD increases from (2 to 3)°C for 18°C supply air temperature .

The relationship between temperatures of supply air and VATD calculated between levels for (10 and 170)cm for different internal load is seen in figure (12). It's can be noticed as that temperature of supply air   increases , the VATD increases due to low mixing between air layers as the velocity of air is constant for each internal load .For internal load of (1600,1300 and 1000)W, the VATD increasing from (4 to 4.5)°C,(3.5 to 4.2)°C and (3.1 to 3.9)°C respectively for air temperature of supply air increase from (18 to 24)°C. Also we can see that as internal load increases ,VATD increases for each supply air temperature. This is due to low mixing between air layers at high internal load .For internal load increase from (1000 to 1600)W, the VATD increase from (3.1 to 4)°C for 18 °C supply air temperature.

Figure (13) shows the relationship between supply air temperature and air capacity at constant inlet velocity of supply air for different internal load. It's clear as temperature of supply air increases, the cooling capacity of air

decreases for different internal load. At internal load of (1600 ,1300 and 1000)W, the cooling capacity decreases from (929.42to 710.73) , from (790 to 600) and from (630 to 490)W respectively for supply air temperature increase from (18 to 24))°C. This is attributed to the temperature difference between supply and return air decrease. Also notice that as internal load increase , the air capacity increases due to the increase in the temperature of return air because at higher internal load the air room temperature rises. At supply air temperature of 18°C, air capacity increases from (630 to 929.42) W when internal load increases from (1000 to 1600)W respectively.

Figure (14), describes the relationship between PMV and supply air temperature for different internal load with constant air mass flow rate. It can be noticed that as supply air temperature increase, the PMV increases for different internal load due to the increase air room temperature since air capacity decreases with increases temperature of supply air. For internal load of (1600,1300 and 1000)W the PMV increase from (1.18to2.05) , from (0.97 to 1.75) and from (0.54 to 1.45)for supply air temperature increase from (18to 24)°C respectively. Also notice that as internal load increases the PMV increase for different supply air temperature due to increase air room temperature with increases internal load. At internal load increase from(1000 to 1600)W, the PMV increases from (0.54 to 1.18) for supply air temperature is 18°C.

Figure (15), show the relationship between PPD and supply air temperature for different internal load. It's clear as supply air temperature increase the PPD increase for different internal load . At internal load of (1600,1300 and 1000) W the PPD increases from (34to 79) , from (25 to 64) and from (11 to 48) with increased supply air temperature from (18to 24)°C respectively due to decrease in air capacity with increasing supply air temperature. Also it's clear, for different air supply temperature, the PPD increase with increased supply air temperature due to increases air room temperatures .For internal load increase from(1000 to 1600)W, the PPD increase from (11 to 34) for supply air temperature of 18°C.

Conclusion:-
The thermal conditions aren't be achievement for internal load is (1600,1300 and 1000) W at supply air velocity is 0.75 m/s since the value of PMV and PPD lie out range limited by ASHRAE standard 55 while thermal condition is presented for internal load (1600 and 1300) W for supply air velocity is (1.45 and 1.25) m/s respectively due to satisfy limited of ASHRAE standards . But for internal load is 1000 W , the air velocity 1.45 m/s isnt achieve thermal conditions due to sensation of slightly cool so its consider unsuitable velocities for this load. Also the thermal conditions achievement for internal load 1000 W at supply air velocity is (1and 1.25) m/s, due to lie value of PPD and PMV at limitation of ASHRAE 55 standard.

Thermal conditions using CEB tool for supply air temperature is (20, 22 and 24) °C. Illustrate from figures, the thermal conditions aren't achievement due to value of PPD and PMV are outlet range limited by ASHREA 55 standard.

Nomenclature
L:Length , m
W:Width ,m
H:Hieght , m
Q: Heat transfer , W
A:Area m^2
m˙ :Mass flow rater kg/s

Greek Symbols
ρ : Density , kg/m^3
π:Constant ratio$\frac{22}{7}$

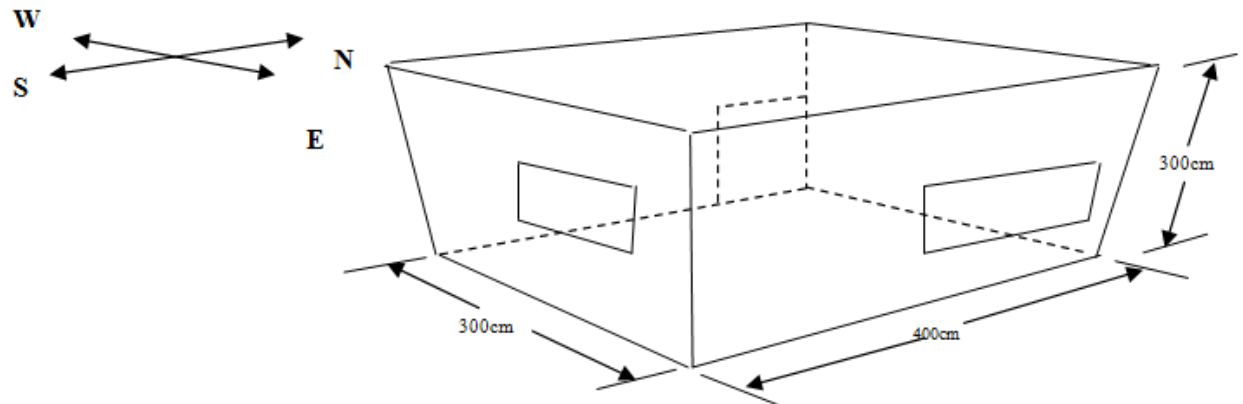**Subscripts**
S:Supply
r:Radtion

Abbreviations
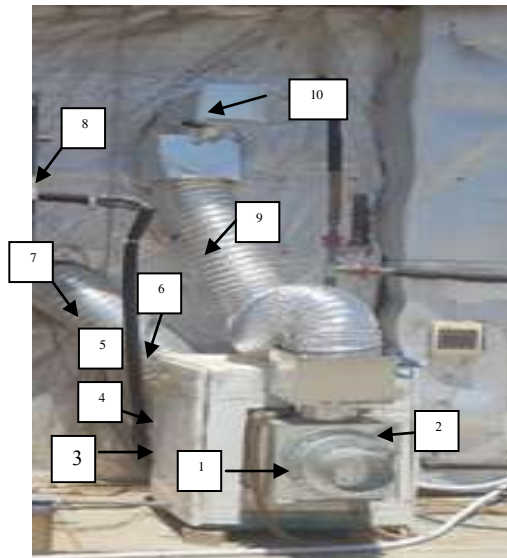DV: Displacement Ventilation

PMV: Predicated Mean Vote
PPD: Percentage People Dissatisfied
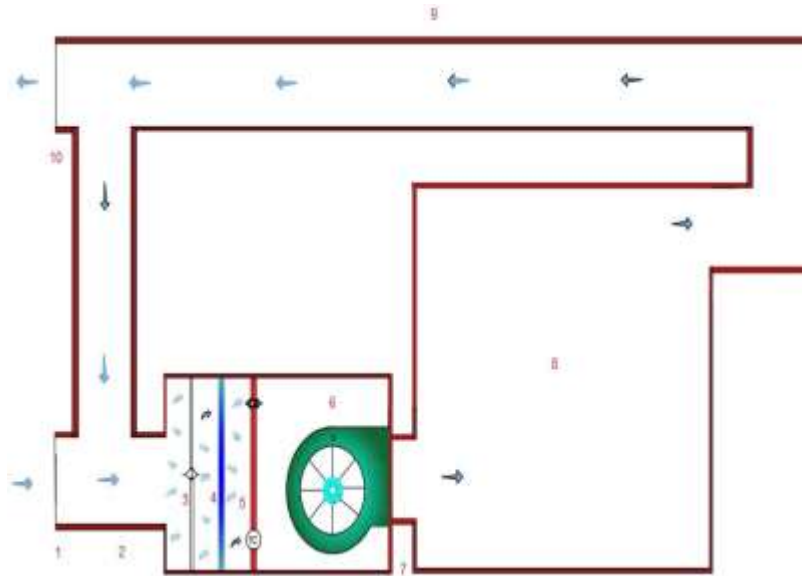CP: Specific Heat at constant Pressure
VATD: Vertical Air Temperature Difference

Figure 1:-The tested room diagram sketch

A:-Photcraphic picture of DV unite with it, s components.

B:-Graphical Picture of DV unite
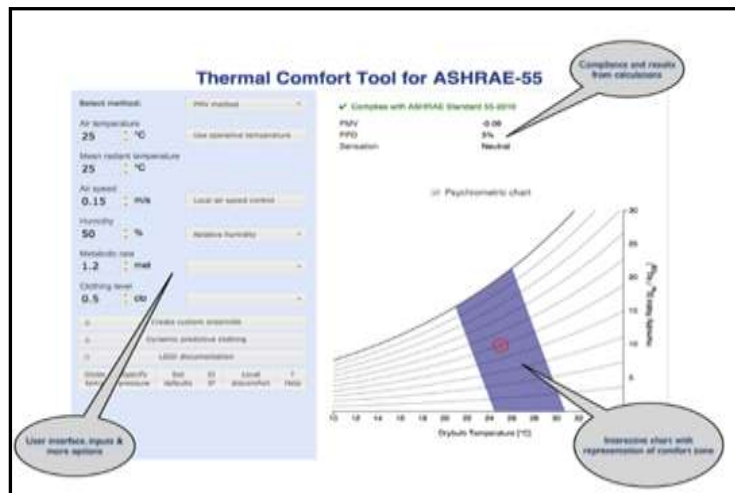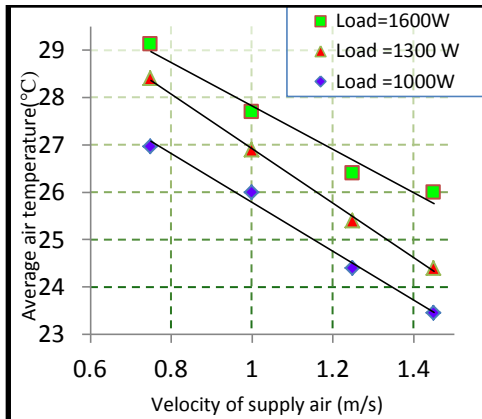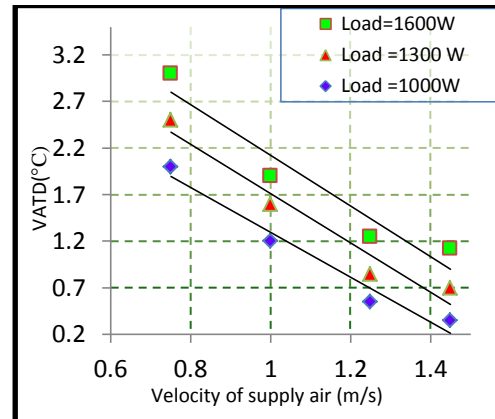**Figure 2:-**The DV components

Figure 3:-The CBC Tool

Number	Description
1	Inlet Gate
2	Inlet Duct
3	Air Filter
4	Cooling Coil
5	Heater
6	Centrifugal Fan
7	Supply Duct
8	Room
9	Return Duct
10	Vent Duct

Figure 4:-The relationship between average air temperature of room and supply air velocity for different internal load.

Figure 5:- The relationship between air temperature difference calculated between level (10and110)cm and supply air velocity for different internal load.
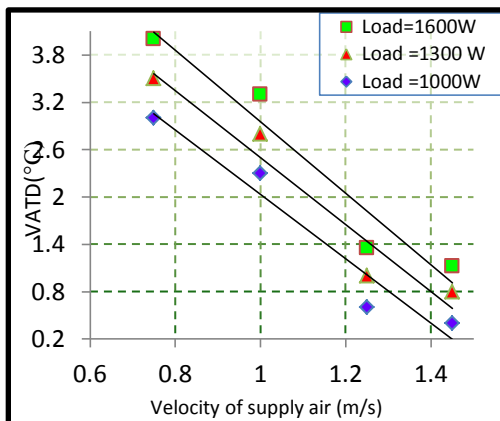
Figure 6:-The relationship between air temperature difference calculated between VATD (170and10)cm and supply air velocity for different internal load.
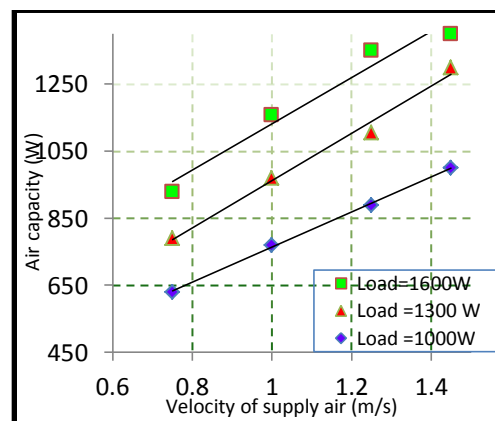
Figure 7:-The relationship between air capacity and supply air velocity for different internal load.
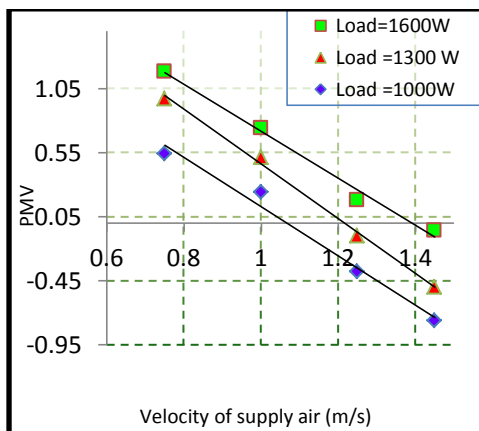
Figure 8:-The relationship between PMV and supply air velocity for different internal load.
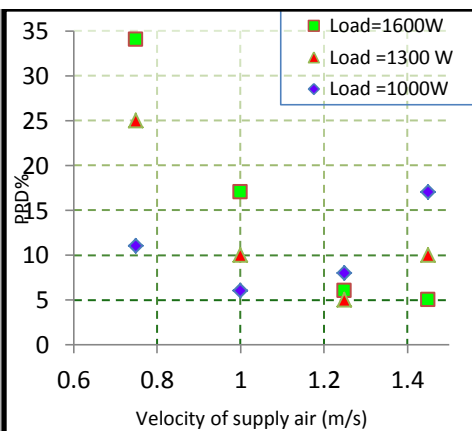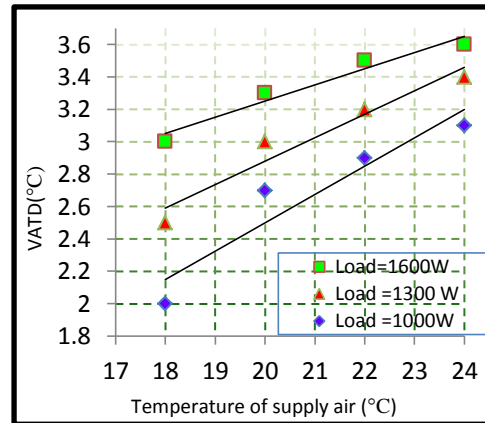

Figure 9:-The relationship between PPD and supply air velocity for different internal load.
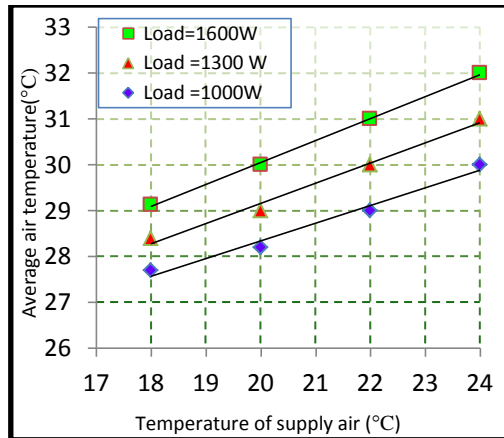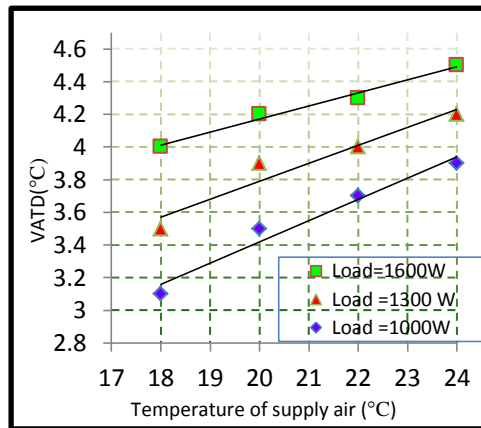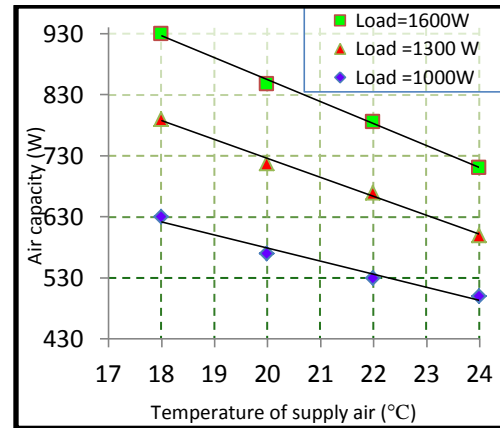
Figure 10:-The relationship between average air temperature of room and supply air temperature for different internal load.
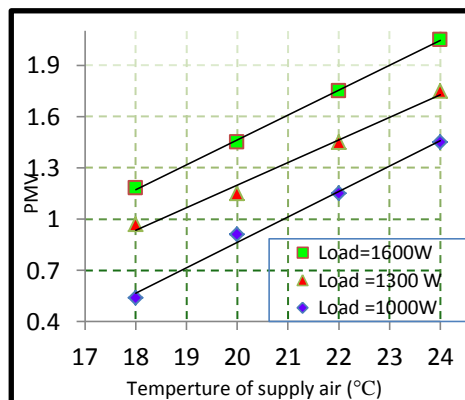

Figure 11:-The relationship between air temperature difference calculated between level (110 and 10)cm and supply air temperature for different internal load.
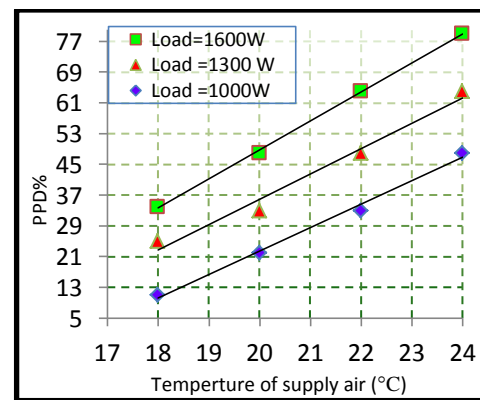

Figure 12:-The relationship between air temperature difference calculated between level (170and10)cm and supply air temperature for different internal load.

Figure 13:-The relationship between air capacity and supply air temperature for different internal load.

Figure 14:-The relationship between PMV and supply air temperature for different internal load.

Figure 15:-The relationship between PPD and supply air temperature for different internal load.

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