ROLES & RESPONSIBILITIES

•Simulated the performance of electrical enclosure under different environment conditions and for different orientations of enclosure using analytical calculations and ANSYS

•Studied the effect of these results on cooling capacity of the fans

•Validated simulation results using theoretical results for different orientations of enclosure and various environmental conditions

RESULTS

• The optimum orientation and environmental conditions required for an electrical enclosure to reduce load on cooling fans was determined

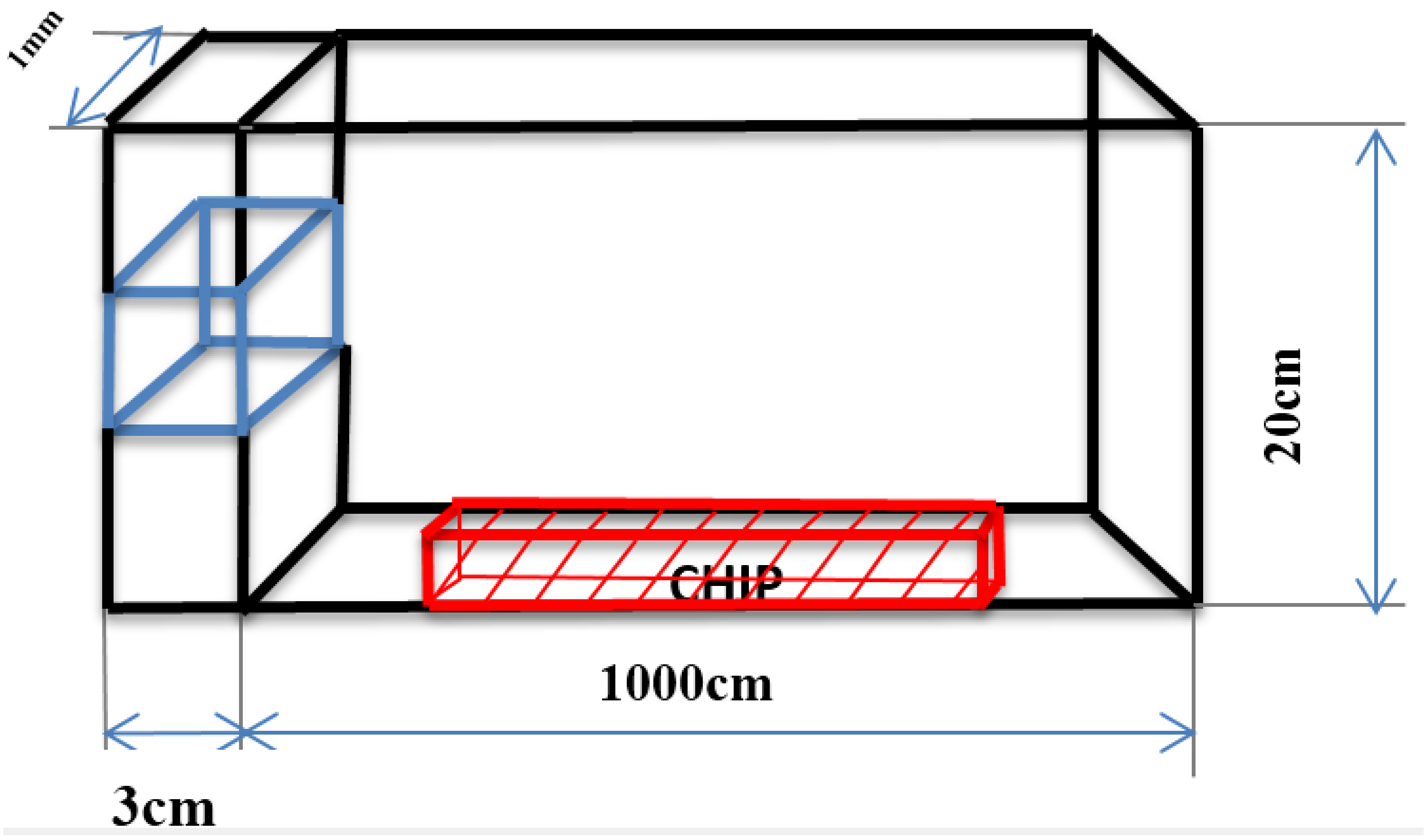
•Acquired the skill of selecting suitable fans by studying the effect of these conditions on cooling capacity of fans

* Simulation results could not be shared due to confidentiality issue

METHODLOGY

- The contribution of free convection and conduction to heat transfer is determined
- The remaining heat load is then transferred by using forced convection
- Based on this load the capacity of fan is decided.

1.A chip is placed inside a plastic enclosure of following dimensions. Will the setup shown in diagram below be sufficient to maintain an temperature of 80°C inside the enclosure with a power loss of 100W ?



Given:

Maximum enclosure temperature = 80℃

Power loss = 100W

To find:

A suitable cooling solution to the setup

Assumptions:

- Since the enclosure is made of plastic the heat transfer in enclosure due to conduction and radiation is neglected.
- The maximum ambient temperature is assumed to be 30℃ and density of air =1.23kg/m^3

Solution:

$$Q = m\, C_p \Delta\, T$$

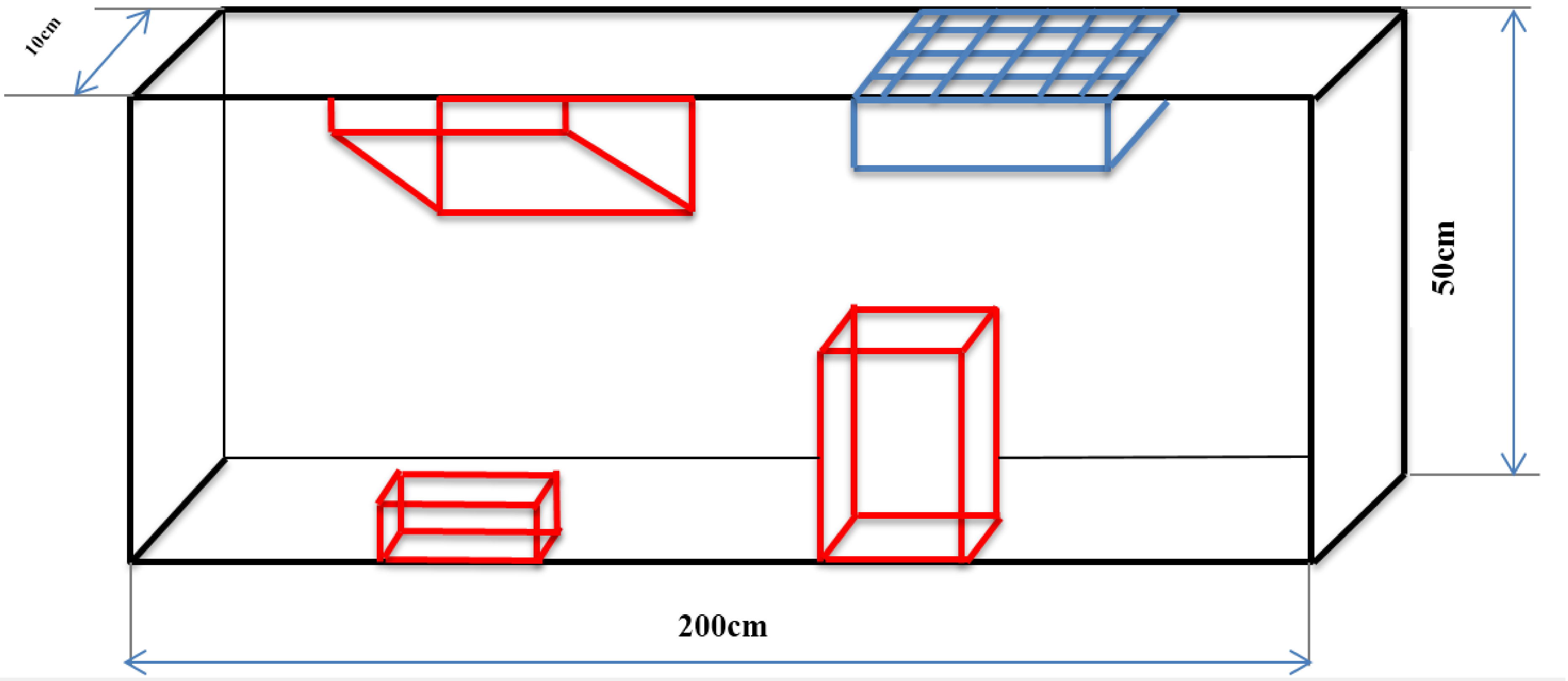
$$m = 0.00124 \text{ kg/s}$$

$$V = \frac{m}{\rho} = 2 \text{ cfm}$$

Hence a fan of capacity 2cfm is required to maintain an enclosure temperature of 80℃ provided ambient temperature is 30℃

2.For the enclosure setup shown below with bottom surface mounted determine the fan capacity with enclosure temperature of 80°C and ambient temperature of 50°C

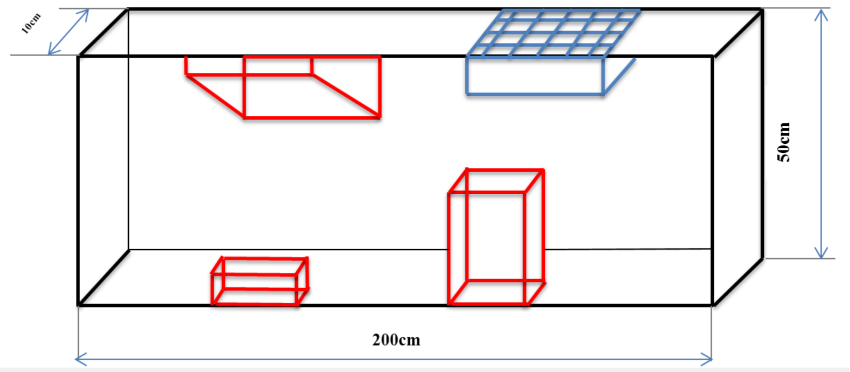
a)If the setup is kept at an height of 1000m,what is the pressure and temperature and fan capacity required?



Given:

Enclosure temperature = 80°C

Ambient temperature = 50°C



To find:

Fan cooling capacity =?

Pressure and temperature at altitude of 1000m and a suitable cooling solution at this altitude

Assumptions:

The enclosure is assumed to be made of carbon steel whose thermal conductivity is 35W/mK

1.**Fan capacity required:**

Heat transfer due to conduction

On front and back surfaces:

$$Q = k\,A\frac{\Delta T}{L} = 0.35 \text{ x } (200 \text{ x } 50 \text{ x } 2) \text{ x } \frac{30}{100} \text{ x } 0.8 = 1680\text{W}$$

(Assuming only 80% of front and back surfaces are exposed to conduction)

On side surfaces:

$$Q = k\,A\frac{\Delta T}{L} = 0.35 \text{ x } (50 \text{ x } 10 \text{ x } 2) \text{ x } \frac{30}{200} = 52.5W.$$

On top surface:

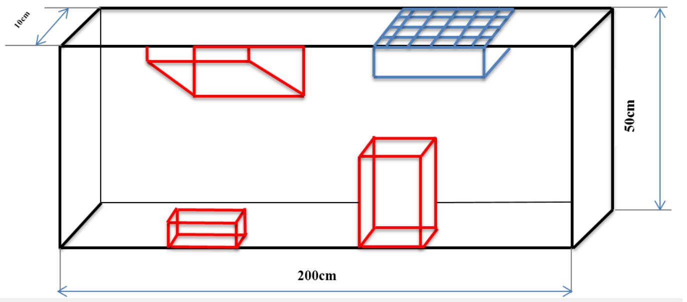
$$Q = k\,A\frac{\Delta T}{L} = 0.35 \text{ x } (200 \text{ x } 10) \text{ x } \frac{30}{50} = 420W$$

Total heat transfer due to conduction is 2153W

Total surface area = 2[(50 x 10) + (200 x 50)] + 200 x 10 =23000 sq cm

Assuming heat transfer due to natural convection per sq cm as 0.05 W/sq cm (Ref:**Hoffmann catalogue**),the total heat transfer due to natural convection is given by 0.05 x 23000 =1150W

Heat load to be transferred by forced convection = Total heat load –[total heat load due to conduction and natural convection]



= 5650 - (2153 + 1150) =2347W.

Heat transfer due to forced convection is given by

$$(Q) = 2347W = m\,C_p \Delta T$$

$$\Rightarrow m = 0.0778 \text{ kg/s}$$

(Since $\Delta T = 30\ ^{\circ}C$ and C_p=1.005KJ/kg mol K)

Assuming ρ as 1.23kg/m^3 at given ambient conditions

$$V = \frac{m}{\rho} = 134 \text{ cfm}$$

Hence a fan of capacity 134 cfm is required to manitain a temperature rise of 30 °C

a) **<u>At an altitude of 1000m:-Pressure</u>**

$$P = P_o e^{-\frac{Mgh}{RT}}$$

On substituting the values P =0.89 bar is obtained for an altitude of 1000m

<u>Temperature:-</u>

$$T = T_o - \frac{x\,m}{1000}$$

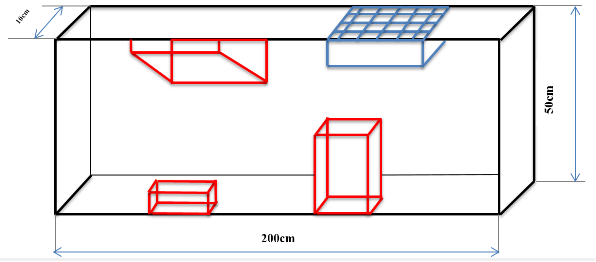
On substituting the values the value of temperature at an altitude of 1000m was found to be 8.8 °C.Since the temperature decreases with increase in altitude natural convection can be used if heat load is less and fans and heat exchangers may be used if heat load is high b)**Iteration 1:**

- Assuming the temperature measurement made inside the enclosure at three separate areas for accurate temperature measurement of enclosure the following values of temperature was obtained

$$T_1 = 53\ °C,\ T_2 = 50°C,\ T_3 = 47°C$$

$$\text{Mean temperature (T)} = T_1 + T_2 + T_3\ /\ 3 = 50.3\ °C \approx 50\ °C$$

- Assuming the enclosure is placed in an cold environment like Ooty,natural convection is sufficient to cool the enclosure. However to prevent condensation an heater is required whose capacity is determined as follows



<u>Heater Capacity:-</u>

Total surface area -23 000 sq cm.

From Hoffmann graph data for Δ T =30 °C and a surface area of 23000 sq cm (≈25sq ft) the minimum heater capacity is determined to be 450W.

Hence only a heater is required to maintain suitable enclosure temperature assuming that heat generated by electronic components in enclosure is very less and this load can be managed by natural convection

Iteration 2:

Assuming Ooty to be at an altitude of 1000m,P= 0.89 bar and T=282K and heat load is decreased by 70% due to natural convection and conduction ,fan capacity is calculated as follows

$$\rho = \frac{P}{RT} = 1.099 kg/m^3$$

Heat load to be removed by forced convection = 2347 x 0.3 =704W

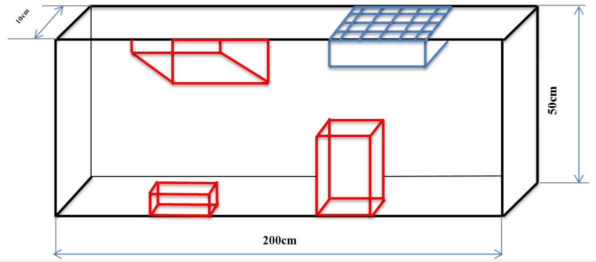
$$(Q) = 704W = m\, C_p \Delta T$$

$$\Rightarrow m = 0.00983 \text{ kg/s}$$

(Since $\Delta T = 71.2°C$ and C_p=1.005KJ/kg mol K)

$$V = \frac{m}{\rho} = 19 \text{ cfm}$$

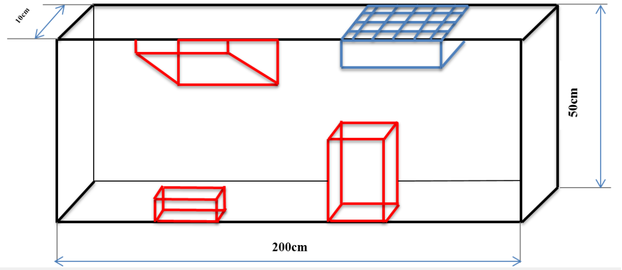
Hence when the enclosure is placed in ooty,a fan of 19cfm along with a heater of capacity 450W is required



- At atmospheric conditions(T=15 °C and P=1bar) for a heat load of 2347W and assuming that same amount of conduction and convection occurs as in first case,the fan capacity required is

$$(Q) = 2347W = m\, C_p \Delta\, T$$

$$\Rightarrow m = 0.0359 \text{ kg/s}$$



At atmospheric conditions , $\rho = 1.23 \text{kg/m}^3$

$$V = \frac{m}{\rho} = 62 \text{ cfm}$$

Hence when the enclosure is placed at sea level at atmospheric conditions the enclosure requires only the use of fan of capacity 62 cfm

Inferences:

- When the enclosure is placed at higher altitude for same conditions the enclosure requires the use of heater along with a fan of suitable capacity.

- The reduction in fan capacity is determined by % of natural convection contributing to heat transfer.

- The heater capacity depends on change in ambient temperature at higher altitude with respect to temperature at sea level.

Iteration 3:-

Assuming the enclosure is placed in very hot environment like Chennai and assuming that the enclosure to be made of white color

Solar load = 0.14 x 24.75sq ft x 94W/sq ft

Where 14% is the amount of solar energy absorbed by the enclosure

Total heat load to be removed = 5650 + 336 =5986W

Assuming that the solar heat causes an increase only in the enclosure temperature and not in the ambient temperature

Total heat load to be removed by forced convection is 5986 -3303 = 2683W

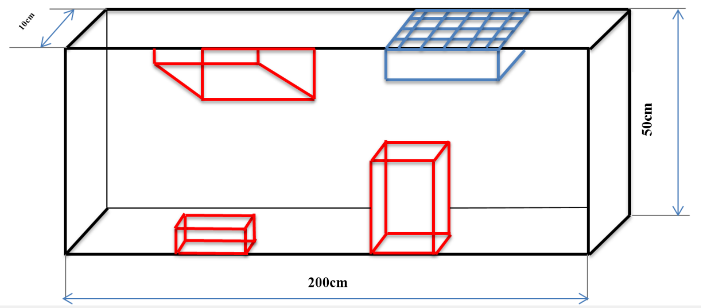
(Assuming that both conduction and natural convection contribute to same amount of heat transfer as in previous case)

Fan capacity required is calculated as

$$(Q) = 2683W = m\, C_p \Delta T$$

$$\Rightarrow \quad m = 0.0889 \text{ kg/s}$$

$$V = \frac{m}{\rho} = 154 \text{ cfm}$$



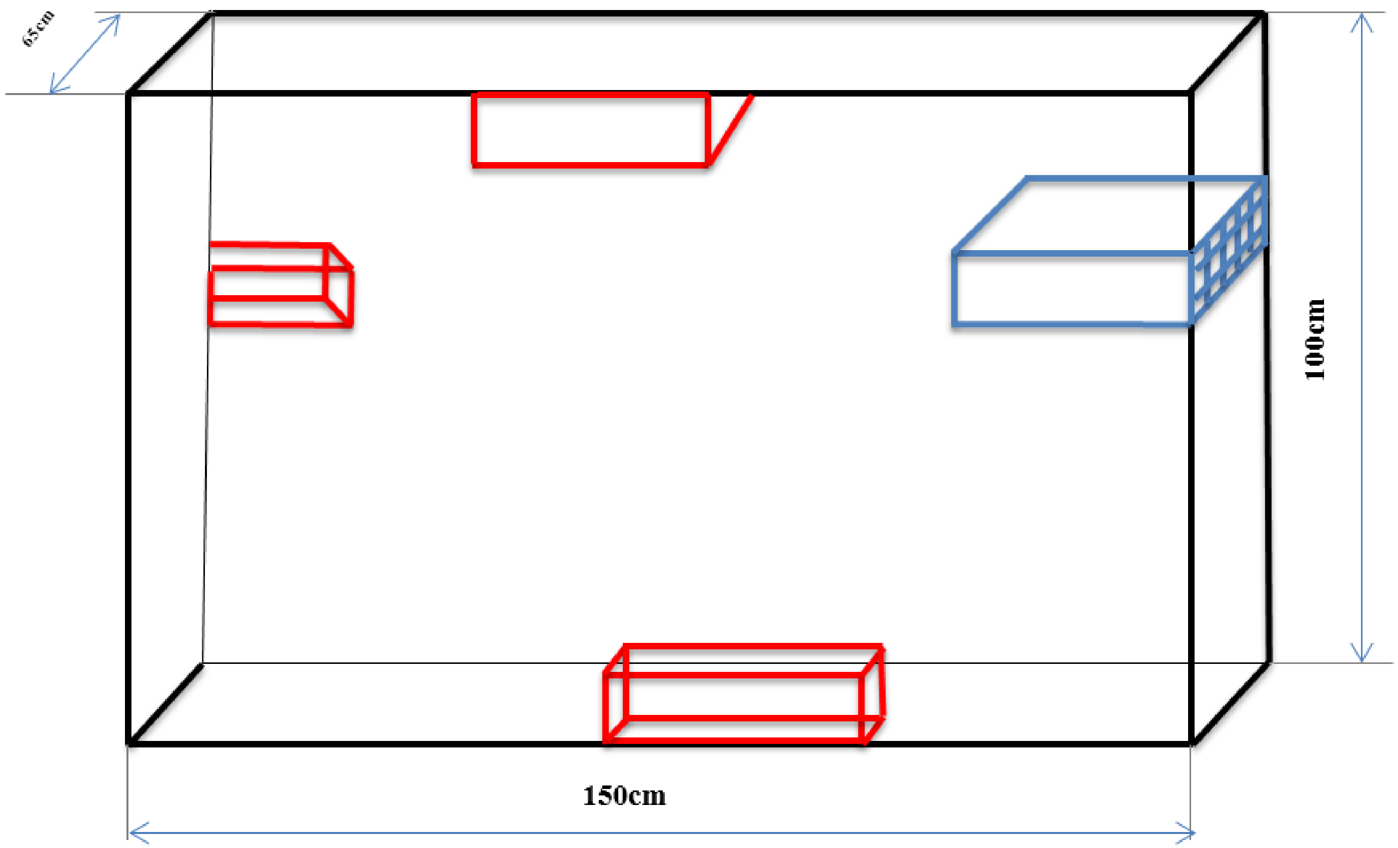
Hence the fan of capacity 154 cfm is required to maintain a operating temperature inside the enclosure with unknown humidity

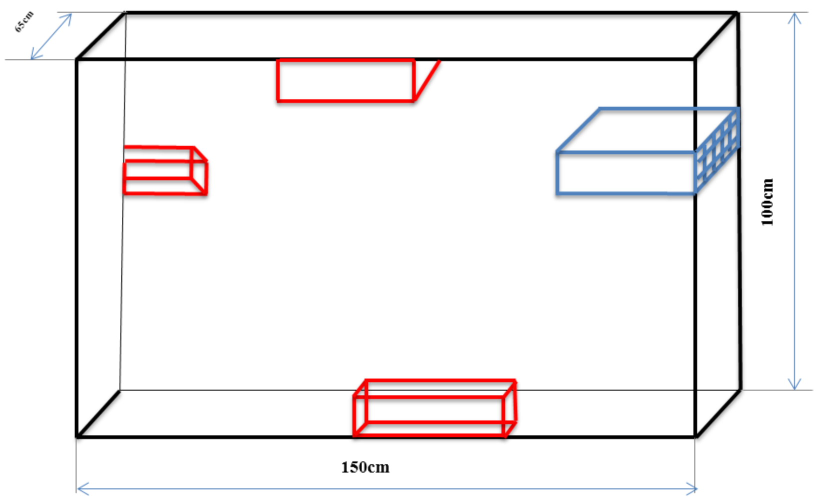
3.Determine the fan capacity for the enclosure setup shown below with bottom surface mounted if enclosure temperature is 50 °C and ambient air temperature is 40 °C for heat load of 4000W.

a)What fan capacity would be required if enclosure is placed at an altitude of 200m?

Determine the corresponding pressure and temperature at this altitude.

b)Where should the fan be positioned in enclosure ? Justify the answer





<u>Given:</u>

Enclosure temperature = 50 °C

Ambient air temperature = 40 °C

<u>To find:</u>

Fan capacity=?

Pressure and temperature at an altitude of 200m=?

<u>Assumptions:</u>

Assume the enclosure material to be stainless steel of k= 20W/mK

<u>Solution:</u>

<u>Fan capacity required:</u>

<u>Heat transfer due to conduction:-</u>

On front and back surfaces:

$$Q = k\,A\frac{\Delta T}{L} = 0.2 \times (100 \times 65 \times 2) \times \frac{10}{100} = 173W$$

On side surfaces:

$$Q = k\,A\frac{\Delta T}{L} = 0.2 \text{ x } (150 \text{ x } 100 \text{ x } 2) \text{ x } \frac{10}{65} = 923W.$$

On top surface:

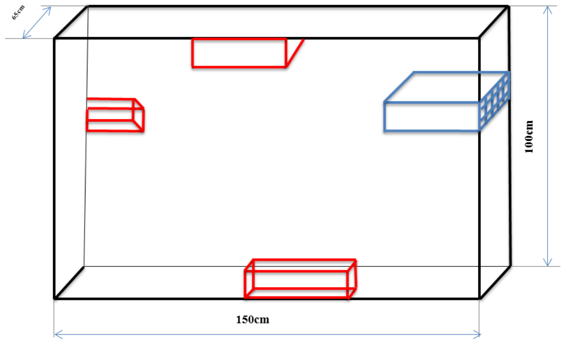
$$Q = k\,A\frac{\Delta T}{L} = 0.2\text{x } (150 \text{ x } 65) \text{ x } \frac{10}{100} = 195W$$

Total heat transfer due to conduction is 1291W

Total surface area = 2[(150 x 65) + (100 x 150)] + 150 x 65 =52750 sq cm

Assuming heat transfer due to natural convection per sq cm as 0.03 W/sq cm (Ref:**Hoffmann catalogue**),the total heat transfer due to natural convection is given by 0.03 x 52750 = 1588W

Heat load to be transferred by forced convection = Total heat load –[total heat load due to conduction and natural convection]



= 4000 - (1291 + 1588) = 1121W.

Heat transfer due to forced convection is given by

$$(Q) = 1121W = m\,C_p \Delta T$$

$$\Rightarrow m = 0.1120 \text{ kg/s}$$

(Since $\Delta T = 10$ °C and C_p=1.005KJ/kg mol K)

Assuming ρ as1.128kg/m^3at given ambient conditions,

$$V = \frac{m}{\rho} = 210\text{cfm}$$

Hence a fan of capacity 210 cfm is required to maintain a temperature rise of 10 ℃

a) <u>At an altitude of 200m:-Pressure</u>

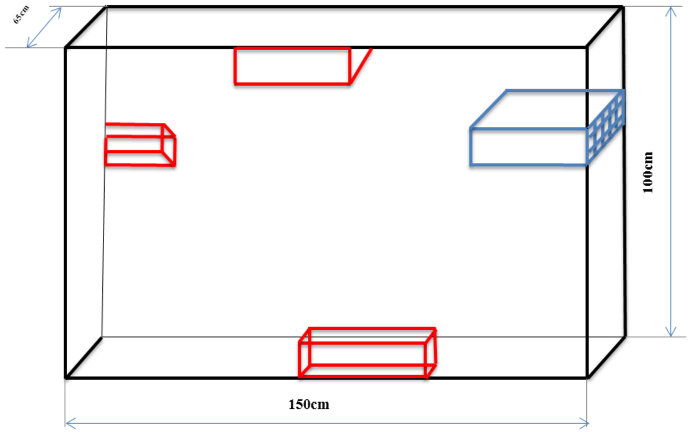
$$P = P_o e^{-\frac{Mgh}{RT}}$$

On substituting the values P =0.989 bar is obtained for an altitude of 200m

<u>Temperature:-</u>

$$T = T_o - \frac{x\,m}{1000}$$

On substituting the values the value of temperature at an altitude of 1000m was found to be 13.76 ℃.



$$\rho = \frac{P}{RT} = 1.19 kg/m^3$$

$$(Q) = 1121W = m\, C_p \Delta T$$

$$m = 0.0311 \text{ kg/s}$$

$$V = \frac{m}{\rho} = 55 \text{ cfm}$$

Hence fan capacity required is less when the enclosure is placed at a higher altitude than when it is placed at enclosure.

Iteration 1:- effect of obstructions:

Assuming a safety margin of 1.25 due to minor heat producing components Q = 1121 x 1.25 = 1408W.

Using Hoffmann chart for Δ T =10 °C and a heat load of 1408W,free airflow without impedance was found to be 320 cfm.

Assuming the devices are moderately packed in enclosure the static pressure is assumed to be 436 Pa **(Ref:Hoffmann data).** Hence actual capacity of fan required is increased by 60cfm.(**Ref:Hoffmann data fan curve).**

Effect of fan grille on design:

Wire fan guards:

This fan guard creates a blockage of air that results in 4-5% reduction in cfm **(Ref:Gardtec fan guards)**

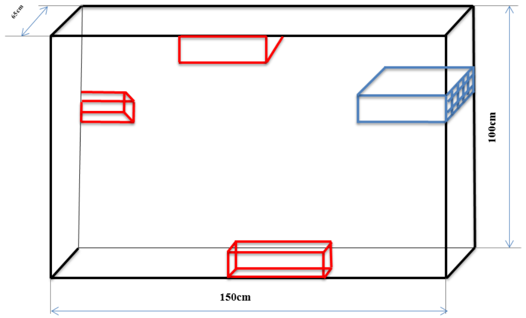
Sheet metal punched fan guards:

This fan guard creates a blockage of air that results in 51% reduction in cfm **(Ref:Gardtec fan guards)**

Assuming the use of wire fan guards the actual fan capacity required is 320 + 60 + (1.05 *320) = 396 cfm

Iteration 2:

Assume enclosure to consist of 4 electronic components that draw 115 VAC at 9.5 AC and each has rated efficiency of 90% the



Total internal load = 4 x 115 x 9.5 x 0.1

= 437 W

Assume the enclosure has a 3HP VFD with efficiency of 95 % then

VFD het load = 3 x 745.6W x 0.05

= 112 W

Assume the enclosure to have an output control line of 75 VAC at 5A

Outgoing power = 375 W

Total heat load = Internal power – External power

= 202W

Since Q is very less, natural convection can be used for cooling the enclosure.

Iteration 3:

Assume the enclosure to be placed in high humid region with high polluted environment like Delhi the heat load is assumed to be 980W.

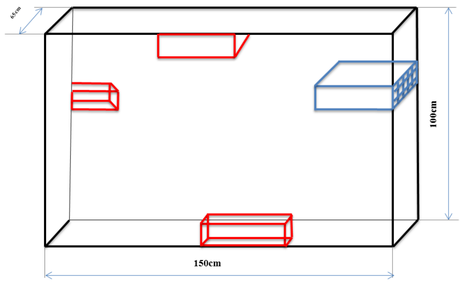
Total surface area = 2[(150 x 65) + (100 x 150)] + 150 x 65 =52750 sq cm

Assuming heat transfer due to natural convection per sq cm as 0.02 W/sq cm (Ref:**Hoffmann catalogue**),and assuming only 85 % of this area is exposed to heat ,the total heat transfer due to natural convection is given by 0.02 x 52750 x 0.85 = 897W

Assuming the heat loss as 195W

Total heat load to be transferred = 897 - 195 = 702W

Hence an air conditioner of minimum capacity of 702
temperature rise of Δ T =10 °C .



Iteration 4:

When horizontal heated from bottom:

Nusselt number for vertical enclosures is given by

$$\overline{Nu}_L = \frac{\bar{h}L}{k} = 0.069\, Ra_L^{1/3}\, Pr^{0.074} \quad \text{for} \quad 3 \times 10^5 \lesssim Ra_L \lesssim 7 \times 10^9$$

Here

$$Ra_L \equiv \frac{g\beta(T_1 - T_2)L^3}{\alpha\nu} \qquad \text{Where } L = \frac{2wh}{2(w+h)}$$

$= 7.065\text{ e}8$

By substituting values in Nusselt number relation the number was found to be 60

Using Nusselt number relation the convection coefficient was found to be $1.60 W/m^2K$

The corresponding heat transfer by free convection is given by

$$Q = hA\Delta T \quad = 10.4W$$

Ref:Heat and mass transfer by Incropera

When vertical:

Nusselt number for vertical enclosures is given by

$$Nu = 0.18\left(\frac{Pr}{0.2 + Pr} Ra_L\right)^{0.29} \quad \text{for} \quad \begin{matrix} 1 < H/L < 2 \\ \text{Any prandtl number} \\ \frac{Pr}{0.2 + Pr} Ra_L > 10^3 \end{matrix}$$

Here

$$Ra_L \equiv \frac{g\beta(T_1 - T_2)L^3}{\alpha\nu} \qquad \text{Where } L = \frac{2wh}{2(w+h)}$$

= 7.065 e8 and H/L =1.5

By substituting values in Nusselt number relation the number was found to be 62

Using Nusselt number relation the convection coefficient was found to be 1.66W/m^2K

The corresponding heat transfer by free convection is given by

$$Q = hA\Delta T \quad = 11W$$

Ref:Heat and mass transfer by Incropera

Ref:Heat and mass transfer by Incropera

At angle of 30 degrees:

$$\overline{Nu}_L = \overline{Nu}_L(\tau = 90^\circ)(\sin \tau)^{1/4} = 52$$

Usning the Nusselt number correlation the conection coefficient h was found to be 1.39 W/m^2K

The corresponding heat transfer by free convection is given by

$$Q = hA\Delta T = 9W$$

At angle of 45 degrees:

$$\overline{Nu}_L = \overline{Nu}_L(\tau = 90^\circ)(\sin \tau)^{1/4} = 57$$

Usning the Nusselt number correlation the conection coefficient h was found to be $1.53 W/m^2K$

The corresponding heat transfer by free convection is given by $Q = hA\Delta T = 9.94W$

At 75 degrees:

$$\overline{Nu}_L = \overline{Nu}_L(\tau = 90°)(\sin \tau)^{1/4} \quad = 61$$

Usning the Nusselt number correlation the conection coefficient h was found to be $1.63 \mathrm{W/m^2K}$

The corresponding heat transfer by free convection is given by

$$\mathrm{Q} = \mathrm{hA\Delta T} \quad = 10.6\mathrm{W}$$

Ref:Heat and mass transfer by Incropera

At altitude of 200m:

When horizontal:

$$Nu=\frac{hL}{k}$$

Usning the Nusselt number correlation the convection coefficient h was found to be $1.593 W/m^2K$

The corresponding heat transfer by free convection is given by $Q = hA\Delta T$ = 37.27W

When vertical:

Usning the Nusselt number correlation the conection coefficient h was found to be $1.62 W/m^2K$

The corresponding heat transfer by free convection is given by $Q = hA\Delta T$ = 38W

Ref:Heat and mass transfer by Incropera

Ref:Heat and mass transfer by Incropera

<u>At angle of 30 degrees:</u>

Usning the Nusselt number correlation the convection coefficient h was found to be $1.344 W/m^2 K$

The corresponding heat transfer by free convection is 31.4W

<u>At angle of 45 degrees:</u>

Usning the Nusselt number correlation the convection coefficient h was found to be $1.47 W/m^2 K$

The corresponding heat transfer by free convection is 34W

<u>At 75 degrees:</u>

Usning the Nusselt number correlation the convection coefficient h was found to be $1.57 W/m^2 K$

The corresponding heat transfer by free convection is 37W

Effect of compressibility:

The assumption that fluid is incompressible is accurate for Mach number <0.3

(Ref:IIT Kharagpur material)

$$\text{Mach number} = \frac{\text{Speed of sound in that medium}}{\text{Speed of sound in air}}$$

Speed of sound in that medium =V =Q/A = 0.14 m/s

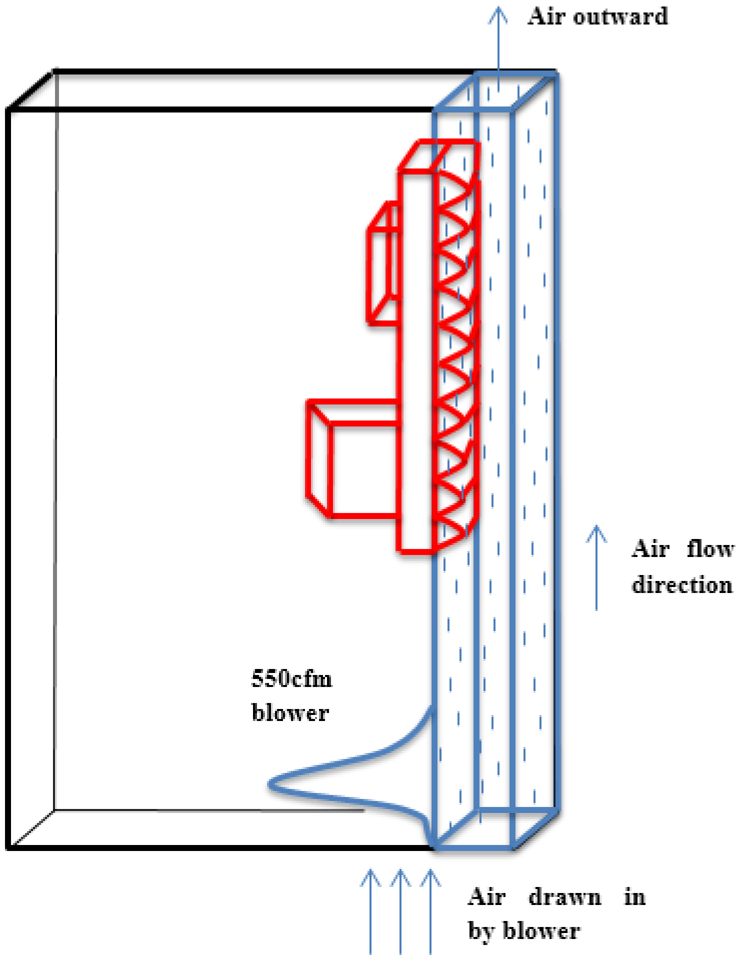
Speed of sound in air = 331 + 0.6(T in ℃) = 355m/s

Here Mach number = 3.94e-4 **<** 0.3

Hence the assumption that fluid is incompressible is accurate for this case b)

FAN PLACED AT INLET		FAN PLACED AT OUTLET	
ADVANTAGES	DISADVANTAGES	ADVANTAGES	DISADVANTAGES
• Better reliability, • Higher mass flow rate for same size of fan • Air can be filtered prior to entry	• Heat load of fan is added to the system • Filters must be changed frequently	Heat load of fan is carried away by virtue of position of fan	Vaccum is created in the enclosure

4.For the back channel cooling enclosure shown below for which the right hand side surface is mounted, justify the advantage of setup compared to other cooling solutions provided in electrical enclosures when the air temperature to the inlet of channel is reduced from 22°C to 10°C with maximum ambient temperature of 50°C if the blower capacity is 550cfm and total power loss is 5000W. Make suitable assumptions.



Length:150cm
Width :600mm
Height:180cm
Width of channel :20cm

Given:

Blower capacity = 700cfm

Air inlet temperatures = 22℃ & 10℃

To find:

To justify the advantage of setup

Assumptions:

Assume that total heat liberated by electronic components is removed by conduction on side surfaces of channel (80 % of area) and convection and conduction on other surfaces and radiation are negligible.

Assume that 1% of total heat liberated by electronic components is removed by conduction through base plate and fins

Assume 35 % of area in back channel is covered by fins.

Assume enclosure material as stainless steel whose k= 20W/mK

Solution:

<u>For existing system:</u>

<u>Rejection temperature of air:</u>

$$\frac{T2}{T1} = \left(\frac{P2}{P1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T2 = 305K$$

(Assuming the maximum specific ratio provided by ASHRAE for blowers AS 1.12)

$$\eta = \frac{T2i - T1}{T2a - T1}$$

$$\Rightarrow T2a = 307K$$

[Assuming a maximum isentropic efficiency of 85% (Ref:Siemens catalogue)]

<u>Heat due to conduction:</u>

<u>On side surfaces:</u>

$$Q = k A\frac{\Delta T}{L} = 0.2 \text{ x } (180 \text{ x } 20 \text{ x } 2) \text{ x } \frac{16}{60} \text{ x } 0.8 = 307W$$

On base plate and fins:

Heat transferred by base plate and fins = 50W

Total surface area exposed for natural convection = [2[(20 x 180)] + (60 x 180) = 12000sq cm

Assuming heat transfer per sq cm to be 0.037W/sq cm total heat transferred by natural convection = 0.03 x 12000 x 0.65 = 234W

Heat load to be removed by forced convection = 5000 – (307 + 50 + 234) =4409W

Now the pressure of air is determined using ideal gas equation

$$\frac{P1V1}{T1} = \frac{P2V2}{T2} \Rightarrow \frac{P1}{T1} = \frac{P2}{T2} \Rightarrow P2 = 1.036\text{bar}$$

(Taking constant volume)

Assuming specific ratio of 1.12,P = 1.12 x P_1 = 1.16bar

(Assuming the maximum specific ratio provided by ASHRAE for blowers)

Mass flow rate of air:

Heat load to be carried by forced convection = Q = 4409 = $mCp\Delta T$

$\Rightarrow$ m = 0.2741kg/s

Density of air:

$$\rho = \frac{P}{RT} = 1.1758 \text{ kg/m}^3$$

Capacity of fan:

$$V = \frac{m}{\rho} = 494\text{cfm}$$

Hence blower of capacity 494 cfm is required for this case

For new system:

Rejection temperature of air:

$$\frac{T2}{T1} = \left(\frac{P2}{P1}\right)^{\frac{\gamma-1}{\gamma}} \quad T2 = 292K$$

(Assuming the maximum specific ratio provided by ASHRAE for blowers AS 1.12)

$$\eta = \frac{T2i - T1}{T2a - T1}$$

$$T2a = 294K$$

[Assuming a maximum isentropic efficiency of 85% (Ref: Siemens catalogue)]

Heat due to conduction

On side surfaces

$$Q = k\,A\frac{\Delta T}{L} = 0.2 \text{ x } (180 \text{ x } 20 \text{ x } 2) \text{ x } \frac{29}{60} \text{ x } 0.8 = 557W$$

On base plate and fins:

Heat transferred by base plate and fins = 50W

Total surface area exposed for natural convection = [2[(20 x 180)] + (60 x 180) = 12000sq cm

Assuming heat transfer per sq cm to be 0.037W/sq cm total heat transferred by natural convection = 0.03 x 12000 x 0.65 = 234W

Heat load to be removed by forced convection = 5000 – (557 + 50 + 234) = 4159W

The temperature and pressure at outlet of blower is not same as conditions as that at inlet of blower due to blower effect: Hence pressure and temperature of air at outlet of blower is determined as follows :

$$\frac{P1V1}{T1} = \frac{P2V2}{T2} \qquad \frac{P1}{T1} = \frac{P2}{T2} \qquad P = 0.9938\text{bar}$$

(Since the process that reduces the temperature is a constant volume process)

Assuming specific ratio of 1.12,P = 1.12 x P_1 =1.1130 bar

(As pressure at 10°C is P_1 = 0.9938bar)

(Assuming the maximum specific ratio provided by ASHRAE for blowers)

Rejection temperature of air:

$$\frac{T2}{T1} = \left(\frac{P2}{P1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T2 = 292K$$

(Assuming the maximum specific ratio provided by ASHRAE for blowers AS 1.12)

$$\eta = \frac{T2i - T1}{T2a - T1}$$

$$\Rightarrow T2a = 294K$$

[Assuming a maximum isentropic efficiency of 85% (Ref:Siemens catalogue)]

Mass flow rate of air:

Heat load to be carried by forced convection = Q = 4294 = $mCp\Delta T$

$\Rightarrow$ m = 0.14274kg/s

Density of air:

$$\rho = \frac{P}{RT} = 1.1777 kg/m^3$$

Capacity of fan:

$$V = \frac{m}{\rho} = 257\text{cfm}$$

Hence a blower of capacity 257cfm is required for this case

Blower performance:

For existing system:

At blower inlet

Pressure at 22°C is P1 = 1.036bar

$$\Rightarrow \rho = \frac{P1}{RT} = 1.2236\ \text{kg/m}^3$$

(Where R = gas constant =287J/kg mol)

For new system:

At blower inlet

Pressure at 10°C is 0.9938bar

$$\rho = \frac{P}{RT} = 1.2235 kg/m^3$$

Since the change in density is very small there is no drastic change in performance characteristics of blower.