8. Forced Convection Heat Transfer

8.1 Introduction

The general definition for convection may be summarized to this definition "energy transfer between the surface and fluid due to temperature difference" and this energy transfer by either forced (external, internal flow) or natural convection.

Heat transfer by forced convection generally makes use of a fan, blower, or pump to provide high-velocity fluid (gas or liquid). The high-velocity fluid results in a decreased thermal resistance across the boundary layer from the fluid to the heated surface. This, in turn, increases the amount of heat that is carried away by the fluid

To understand the convection heat transfer we must know some of the simple relations in fluid dynamics and boundary layer analysis. Firstly we study boundary layer with forced convection flow systems.

8.2 Boundary Layer over Flat Plate

We consider the (x) direction along the wall with (y) direction normal to the wall as in Figure8.1. Where the laminar boundary layer begins at leading edge (x= 0), and followed by transition region and finally to the turbulent region to the trailing edge (x= L). The velocity and temperature of the fluid far away from the surface (out side the boundary layer thickness δ) are the free-stream velocity u_∞ and free-stream temperature T_∞.

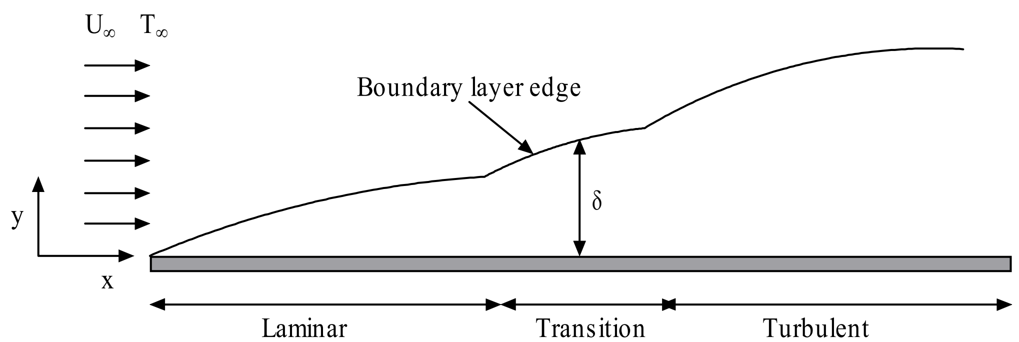

Figure 8.1 Fluid flow over flat plate with laminar, transition, and turbulent boundary layers

8.2.1 Laminar Boundary Layer Equations over Flat Plate ($Re_x \leq 5x10^5$)

The assumptions made to give the simplicity on the analysis are:

1- Steady flow
2- Two-dimensional incompressible viscous flow
3- No pressure variation in the y direction
4- No shear force in the y direction
5- Neglect body force due to gravity

All the basic differential equations can be derived by considering an element control volume inside the laminar region as shown in Figure 8.2.

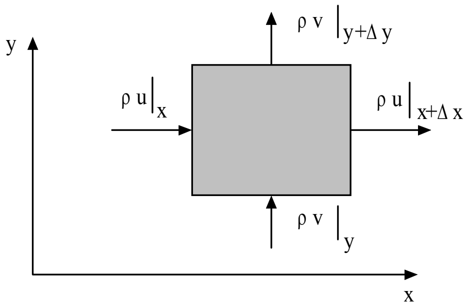

Figure 8.2 Element control volume on laminar region

Continuity equation:

{Rate of mass accumulation within control volume}+
{Net rate of mass flux out of control volume}= 0

Rate of mass accumulation within control volume = $\dfrac{\partial(\rho\Delta x\Delta y)}{\partial t}$ = 0 (steady state assumption)

Net rate of mass flux in x- direction per unit depth = (ρ u]$_{x+\Delta x}$ – ρ u]$_x$)Δy = $\dfrac{\partial(\rho u)}{\partial x}\Delta y\Delta x$

Net rate of mass flux in y- direction per unit depth = (ρ v]$_{y+\Delta y}$ – ρ v]$_y$)Δx = $\dfrac{\partial(\rho \upsilon)}{\partial x}\Delta x\Delta y$

Substitute in continuity equation expression it produce:

$$\frac{\partial(\rho u)}{\partial x}\Delta x\Delta y+\frac{\partial(\rho \upsilon)}{\partial y}\Delta x\Delta y=0$$

$$\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho \upsilon)}{\partial y}=0$$

$$\frac{\partial(u)}{\partial x}+\frac{\partial(\upsilon)}{\partial y}=0 \qquad (8.1)$$

Momentum equation:

By applying Newton's 2nd low on the same element control volume as in Figure 8.2

Time rate of change of linear momentum within the control volume	+	Net rate of linear momentum flux out of the control volume	=	Net rate of linear momentum flux out of the control volume summation of external force acting in the control volume

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Net rate of linear momentum out of the control unit

$$= (\rho uu]_{x+\Delta x} - \rho uu]_x)\Delta y + (\rho vu]_{y+\Delta y} - \rho vu]_y)\Delta x$$

$$= \frac{\partial(\rho uu)}{\partial x}\Delta x\Delta y + \frac{\partial(\rho \upsilon u)}{\partial y}\Delta x\Delta y$$

External forces divided into:

- Pressure force = $-\frac{\partial P}{\partial x}\Delta x\Delta y$
- Viscous force = $\mu\frac{\partial^2 u}{\partial y^2}\Delta x\Delta y$

Substituting in the Newton's 2nd low equation it yeilds

$$\frac{\partial(\rho uu)}{\partial x}\Delta x\Delta y + \frac{\partial(\rho \upsilon u)}{\partial y}\Delta x\Delta y = -\frac{\partial P}{\partial x}\Delta x\Delta y + \mu\frac{\partial^2 u}{\partial y^2}\Delta x\Delta y$$

Or,

$$u\frac{\partial(\rho u)}{\partial x} + \rho u\frac{\partial(u)}{\partial x} + u\frac{\partial(\rho \upsilon)}{\partial y} + \rho \upsilon\frac{\partial(u)}{\partial y} = -\frac{\partial P}{\partial x} + \mu\frac{\partial^2 u}{\partial y^2}$$

From continuity equation we have:

$$\frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho \upsilon)}{\partial y} = 0$$

Then the momentum equation for laminar boundary layer is

$$\rho u\frac{\partial(u)}{\partial x} + \rho \upsilon\frac{\partial(u)}{\partial y} = \mu\frac{\partial^2 u}{\partial y^2} - \frac{\partial P}{\partial x} \qquad (8.2)$$

Energy equation:
For the shown element control volume as in Figure 8.3.with neglected heat conduction in x-direction and applying energy balance, the energy equation may be written as follows:

Energy convected in left face + Energy convected in bottom face + heat conduction in bottom face + net viscous work done on element = energy convected out right face + energy convected out top face + heat conduction out top face

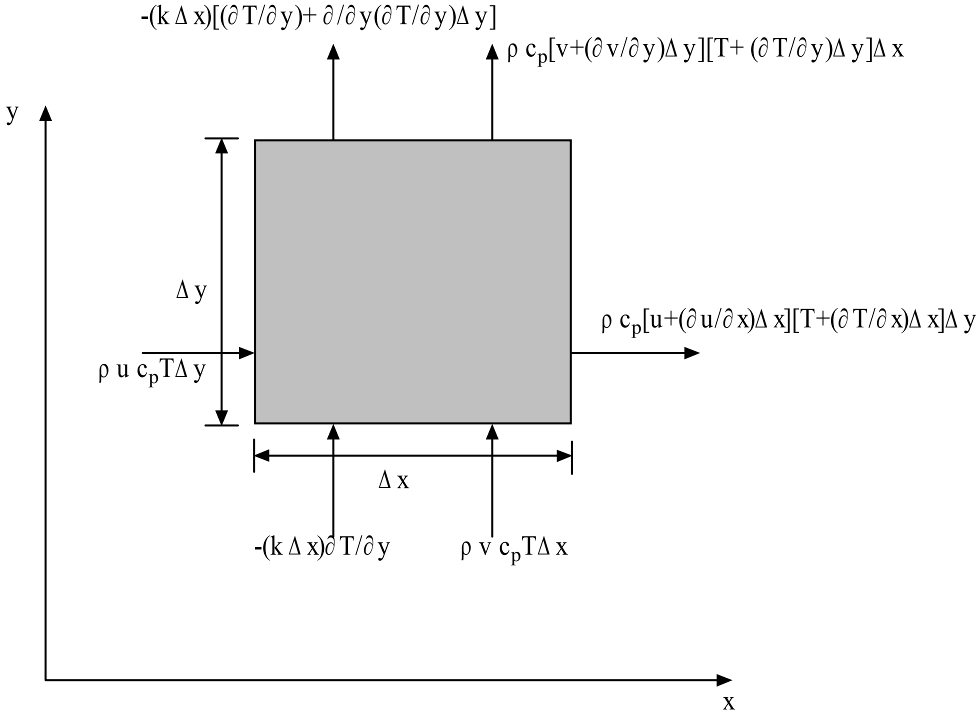


Figure 8.3 Element control volume for energy balance

The viscous shear force is the product of the shear stress and the area per unit depth Δx

$$\mu \frac{\partial u}{\partial y} \Delta x$$

And the distance through which it moves per unit time in respect to the element control volume Δx Δy is

$$\frac{\partial u}{\partial y} \Delta y$$

The net viscous energy delivered to the element control volume

$$\mu \left(\frac{\partial u}{\partial y} \right)^2 \Delta x \Delta y$$

By applying energy balance on the element control volume shown in Figure8.3 neglecting the second order differentials yields to

$$\rho c_p \left[u \frac{\partial T}{\partial x} + \upsilon \frac{\partial T}{\partial y} + T \left(\frac{\partial u}{\partial x} + \frac{\partial \upsilon}{\partial y} \right) \right] \Delta x \Delta y = k \frac{\partial^2 T}{\partial y^2} \Delta x \Delta y + \mu \left(\frac{\partial u}{\partial y} \right)^2 \Delta x \Delta y$$

From continuity Equation 8.1 the energy equation can be written as follow

$$\rho c_p\left[u\frac{\partial T}{\partial x}+\upsilon\frac{\partial T}{\partial y}\right]=k\frac{\partial^2 T}{\partial y^2}+\mu\left(\frac{\partial u}{\partial y}\right)^2$$

Dividing by ρC_p

$$u\frac{\partial T}{\partial x}+\upsilon\frac{\partial T}{\partial y}=\alpha\frac{\partial^2 T}{\partial y^2}+\frac{\mu}{\rho c_p}\left(\frac{\partial u}{\partial y}\right)^2 \tag{8.3}$$

The viscous work term is important only at high velocities since its magnitude will be small compared with other terms when low velocity flow is studied. This may be shown with an order-of-magnitude analysis of the two terms on the right side of energy equation. For this order-of-magnitude analysis we might consider the velocity as having order of the free stream velocity u_∞ and the y dimension of the order of velocity boundary layer thickness δ.

$$u \approx u_\infty \qquad \text{and} \qquad y \approx \delta$$

So that

$$\alpha\frac{\partial^2 T}{\partial y^2} \approx \alpha\frac{T}{\delta^2} \tag{8.4}$$

And

$$\frac{\mu}{\rho c_p}\left(\frac{\partial u}{\partial y}\right)^2 \approx \frac{\mu}{\rho c_p}\frac{u_\infty^2}{\delta^2} \tag{8.5}$$

Now if the ratio between Equations 8.5 and 8.4 is

$$\frac{\mu}{\rho c_p \alpha}\frac{u_\infty^2}{T}=\Pr\frac{u_\infty^2}{c_p T}\langle\langle 1$$

Then we can neglect this term compared to other terms and we can write the energy equation in this simple form.

$$u\frac{\partial T}{\partial x}+\upsilon\frac{\partial T}{\partial y}=\alpha\frac{\partial^2 T}{\partial y^2} \tag{8.6}$$

The solution of these equations (continuity, momentum and energy) is simplified by the fact that, for conditions in the velocity (hydrodynamic) boundary layer fluid properties are independent of temperature.

We may begin by solving the Equations 8.1 and 8.2 (continuity, momentum) to get u and v. Then the energy equation can be solved which depending on calculated results.

The solution of Equations 8.1 and 8.2 can be solved by Blausis exact (analytic) solution for:

- $\frac{\partial P}{\partial x}=0$
- Laminar flow.

In Blausis exact solution, the velocity components are defined in terms of a stream function Ψ(x,y) where

$$u = \frac{\partial \Psi}{\partial y} \tag{8.7}$$

And

$$\upsilon = -\frac{\partial \Psi}{\partial x} \tag{8.8}$$

So that the continuity equation may be intrinsically satisfied

Considering the partial differential equation describing the momentum equation (Equation 8.2), we may use the similarity method in order to convert it into an ordinary differential equation.

Defining the dependent and independent dimensionless variables f and η, will help us in this analytical approach.

$$f(\eta) = \frac{\Psi}{u_\infty \sqrt{vx / u_\infty}} \tag{8.9}$$

$$\eta = y\sqrt{u_\infty / vx} \tag{8.10}$$

The Blausis exact solution is termed a similarity solution, and η is the similarity variable.

This terminology is used because, despite growth of the boundary layer with distance x from the leading edge, the velocity profile u/u_∞, remains geometrically similar as shown in Figure 8.4.

$$\frac{u}{u_\infty} = fun.(\frac{y}{\delta})$$

Where δ is the boundary layer thickness and usually difficult to measure

Assuming this thickness to vary as $(\nu\ x / u_\infty)^{1/2}$, its follows that

$$\frac{u}{u_\infty} = fun.(\eta)$$

Hence the velocity profile is assumed to be uniquely determined by the similarity variable η which depends on x and y directions.

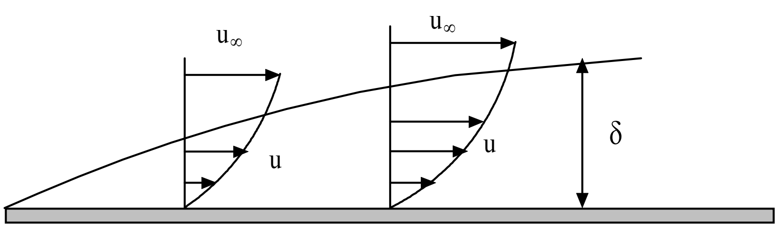


Figure 8.4 the profile u/u_∞ geometrically similar

From Equations 8.7 and 8.8

$$u = \frac{\partial \Psi}{\partial y} = \frac{\partial \Psi}{\partial \eta}\frac{\partial \eta}{\partial y} = u_\infty \sqrt{\frac{vx}{u_\infty}}\frac{df}{d\eta}\sqrt{\frac{u_\infty}{vx}} = u_\infty \frac{df}{d\eta} \tag{8.11}$$

And

$$\upsilon = -\frac{\partial \Psi}{\partial x} = -(u_\infty \sqrt{\frac{\nu x}{u_\infty}} \frac{\partial f}{\partial x} + \frac{u_\infty}{2} \sqrt{\frac{\nu}{u_\infty x}} f)$$

$$\upsilon = \frac{1}{2}\sqrt{\frac{\nu u_\infty}{x}}\left(\eta \frac{df}{d\eta} - f\right) \tag{8.12}$$

By differentiating the velocity components, it may also be shown that

$$\frac{\partial u}{\partial x} = -\frac{u_\infty}{2x}\eta \frac{d^2 f}{d\eta^2} \tag{8.13}$$

$$\frac{\partial u}{\partial y} = u_\infty \sqrt{\frac{u_\infty}{\nu x}} \frac{d^2 f}{d\eta^2} \tag{8.14}$$

$$\frac{\partial^2 u}{\partial y^2} = \frac{u_\infty^2}{\nu x}\frac{d^3 f}{d\eta^3} \tag{8.15}$$

Substituting these equations in the momentum equation, then we obtain

$$2\frac{d^3 f}{d\eta^3} + f\frac{d^2 f}{d\eta^2} = 0 \tag{8.16}$$

This is non linear, third-order differential equation, so that we need three boundary conditions to get a solution, these boundary conditions are:

At η =0

$$f^{/}(\eta) = f(\eta) = 0$$

At η = ∞

$$f^{/}(\eta) = u/u_\infty = 1$$

The solution of Equation 8.16 by numerical integration and the results are given in Table 8.1.

At u/u_∞ =0.99 for η =5, substitute in Equation 8.10

$$\frac{\delta}{x} = \frac{5}{\sqrt{\mathrm{Re}_x}} \tag{8.17}$$

The shear stress can be expressed as

$$\tau_s = \left(\mu \frac{\partial u}{\partial y}\right)_{y=0} = \left(\mu u_\infty \sqrt{\frac{u_\infty}{\nu x}} \frac{d^2 f}{d\eta^2}\right)_{\eta=0}$$

From the Table 8.1

$$\tau_s = 0.332 u_\infty \sqrt{\frac{\rho \mu u_\infty}{x}}$$

And the wall local shear stress coefficient C_f,is given by

$$C_f = \frac{\tau_s}{\frac{1}{2}\rho u_\infty^2} = \frac{0.664}{\sqrt{\mathrm{Re}_x}} \qquad (8.18)$$

Table 8.1: The function *f* (η) for laminar boundary layer over flat plate

$\eta = y\sqrt{\frac{u_\infty}{\nu x}}$	$f(\eta)$	$\frac{df}{d\eta} = \frac{u}{u_\infty}$	$\frac{d^2 f}{d\eta^2}$
0	0	0	0.332
0.4	0.027	0.133	0.331
0.8	0.106	0.265	0.327
1.2	0.238	0.394	0.317
1.6	0.42	0.517	0.297
2	0.65	0.63	0.267
2.4	0.922	0.729	0.228
2.8	1.231	0.812	0.184
3.2	1.569	0.876	0.139
3.6	1.93	0.923	0.098
4	2.306	0.956	0.064
4.4	2.692	0.976	0.039
4.8	3.085	0.988	0.022
5.2	3.482	0.994	0.011
5.6	3.88	0.997	0.005
6	4.28	0.999	0.002
6.4	4.679	1	0.001
6.8	5.079	1	0

Example 8.1: Consider fluid flow at u_∞=0.3 m/s past a flat plate 0.3 m long. Compute the boundary layer thickness at the trailing edge for (a) air and (b) water at 20 °C.

Solution:

Part (a)

From air properties table at 20 °C

ν =15.267x10^{-6} m^2/s.

The trailing edge Reynolds number is

$$\mathrm{Re}_L = \frac{u_\infty L}{\nu} = \frac{(0.3)(0.3)}{15.267\text{x}10^{-6}} = 5895$$

The flow is laminar, from Equation 8.17 the predicted laminar thickness is

$$\frac{\delta}{x} = \frac{5}{\sqrt{5895}} = 0.065$$

At x = 0.3 m

$$\delta = 0.0195 \text{ m}$$

Part (b)

From saturated water properties table at 20°C:

$\nu_{water} = 1.0437x10^{-6}$ m²/s.

The trailing edge Reynolds number is

$$\mathrm{Re}_L = \frac{u_\infty L}{\nu} = \frac{(0.3)(0.3)}{1.0437X10^{-6}} = 86231.67$$

This again satisfies the laminar condition the laminar thickness is

$$\frac{\delta}{x} = \frac{5}{\sqrt{86231.67}} = 0.017$$

At x = 0.3 m

$$\delta = 0.0051 \text{ m}$$

Solving the energy equation (Equation 8.6):

Let the dimensionless temperature $T^* = \dfrac{T - T_s}{T_\infty - T_s}$ and assume similarity solution of the form $T^* = T^*(\eta)$, and substitute in energy equation to give this form.

$$\frac{d^2T^*}{d\eta^2} + \frac{\Pr}{2} f \frac{dT^*}{d\eta} = 0 \qquad (8.19)$$

Where the variable f depend on the Prandtl number values

The appropriate boundary conditions are

$T^*(0) = 0$ And $T^*(\infty) = 1$

The solution may be achieved by numerical integration method for $0.6 \leq \Pr < 50$

It will produce the surface temperature gradient $\left(\dfrac{dT^*}{d\eta}\right)_{\eta=0}$ as the following relation.

$$\left(\frac{dT^*}{d\eta}\right)_{\eta=0} = 0.332 \Pr^{1/3}$$

If $T_s > T_\infty$

Expression for the local heat convection coefficient determined as follow.

$$q_x^{//} = h_x (T_s - T_\infty) = -k \frac{\partial T}{\partial y}$$

$$h_x = \frac{-k}{(T_s - T_\infty)} \frac{\partial T}{\partial y} = -k \frac{T_\infty - T_s}{T_s - T_\infty} \left(\frac{\partial T^*}{\partial y}\right)_{y=0}$$

$$h_x = k \left(\frac{u_\infty}{\nu x}\right)^{1/2} \left(\frac{\partial T^*}{\partial \eta}\right)_{\eta=0}$$

It follows that the local Nusselt number in this form

$$Nu_x = \frac{h_x x}{k} = 0.332\,\mathrm{Re}_x^{1/2}\,\mathrm{Pr}^{1/3} \tag{8.20}$$

And the average heat transfer coefficient can be obtained by integration $\therefore \overline{h_x} = 2h_x$

$$\overline{h_x} = \frac{1}{x}\int_0^x h_x \tag{8.21}$$

$$\overline{Nu_x} = \frac{\overline{h_x}\,x}{k} = 0.664\,\mathrm{Re}_x^{1/2}\,\mathrm{Re}^{1/3}$$

8.3 The Thermal Boundary Layer

Analogous to the velocity boundary layer there is a thermal boundary layer adjacent to a heated (or cooled) plate. The temperature of the fluid changes from the surface temperature at the surface to the free-stream temperature at the edge of the thermal boundary layer as shown in Figure 8.5.

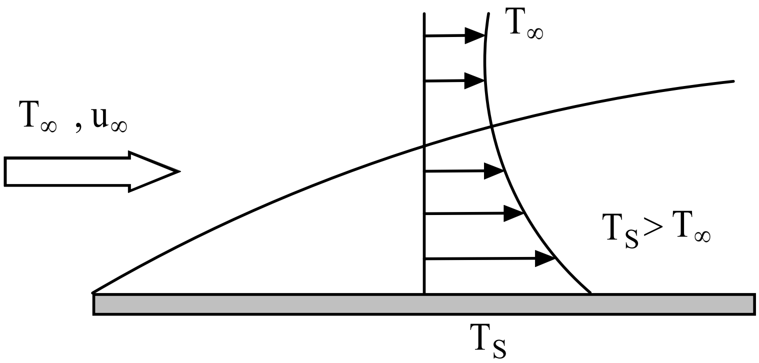


Figure 8.5.Fluid temperature variations inside the thermal boundary layer

The velocity boundary layer thickness depends on the Reynolds number Re_X. But the thermal boundary layer thickness depends both on Re_X and Pr as shown in Figure8.6.

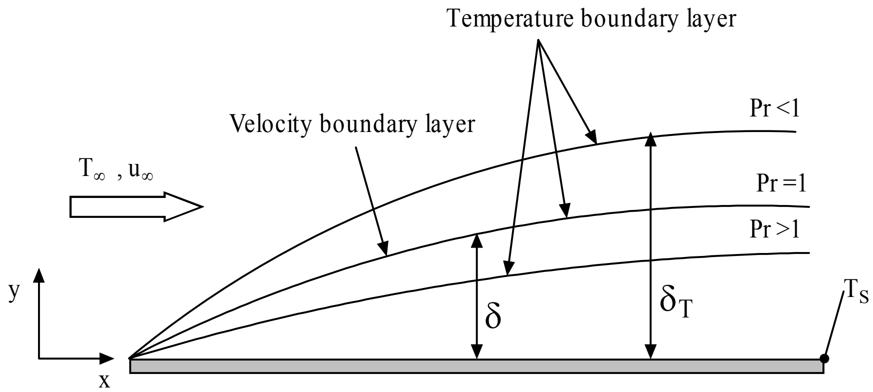


Figure 8.6 thermal boundary layer thicknesses relative to velocity boundary layer thickness at different Prandtle number

For laminar flow ($Re_x < Re_{cr}$):

$$\frac{\delta}{x} = \frac{5}{\sqrt{Re_x}} \qquad \text{At } Pr \geq 0.7 \quad \frac{\delta}{\delta_T} = Pr^{1/3}$$

$$\text{At } Pr \ll 1 \quad \frac{\delta}{\delta_T} = Pr^{1/2} \tag{8.22}$$

For turbulent flow ($Re_x > Re_{cr}$):

$$\frac{\delta}{x} = \frac{0.37}{Re_X^{0.2}} \qquad \delta \approx \delta_T \tag{8.23}$$

8.4 Cooling Air Fans for Electronic Equipment

Air is the most commonly used medium for heat transfer. It is available everywhere on the surface of the planet .Air is usually taken directly from the surrounding atmosphere and returned to it. Many different types of fans are available for cooling electronic equipment. These can generally be divided into two major types: axial and centrifugal fans. These fans can be driven by various types of electric motors, single phase, three phase, 60 cycles, 400 cycles, 800 cycle ac/dc, constant speed, variable speed. The flow rates also vary from 1 cfm to several thousand cfm. When a fan is used for cooling electronic equipment, the airflow direction can be quite important. The fan can be used to draw air through a box or to blow air through a box. A blowing fan system will raise the internal air pressure within the box, which will help to keep dust and dirt out of a box that is not well sealed. A blowing system will also produce slightly more turbulence, which will improve the heat transfer characteristics within the box. However, when an axial flow fan is used in a blowing system, the air may be forced to pass over the hot fan motor, which will tend to heat the air as it enters the electronic box, as shown in Figure 8.7.

An exhaust fan system, which draws air through an electronic box, will reduce the internal air pressure within the box. If the box is located in a dusty or dirty area, the dust and dirt will be pulled into the box through all of the small air gaps if the box is not sealed. In an exhaust system, the cooling air passes through an axial flow fan as the air exits from the box, as shown in Figure 8.8. The cooling air entering the electronic box is therefore cooler.

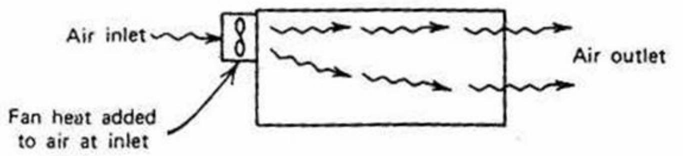


Figure 8.7 Axial flow fan blowing cooling air through a box

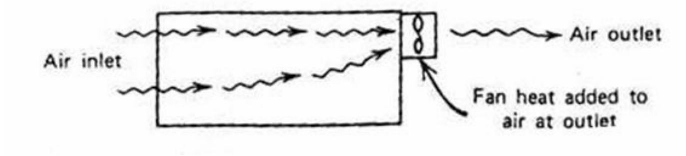


Figure 8.8 Axial flow fan drawing cooling air through a box

8.5 Static Pressure and Velocity Pressure

Airflow through an electronic box is due to a pressure difference between two points in the box, with the air flowing from the high-pressure side to the low-pressure side. The flow of air will result in a static pressure and a velocity pressure. Static pressure is the pressure that is exerted on the walls of the container or electronic box, even when there is no flow of air; it is independent of the air velocity. Static pressure can be positive or negative, depending upon whether it is greater or less than the outside ambient pressure.

Velocity pressure is the pressure that forces the air to move through the electronic box at a certain velocity. The velocity pressure depends upon the velocity of the air and always acts in the direction of the airflow. The amount of cooling air flowing through an electronic box will usually determine the amount of heat removed from the box. The higher the air flow rate through the box the higher heat will be removed. As the airflow through the box increases, however, it requires an even greater pressure to force the air through the box.

Static and velocity pressures can be expressed in lb/in^2 and g/cm^2. However, these values are usually very small, so that it is often more convenient to express these pressures in terms of the height of a column of water. The velocity head (H_v) is a convenient reference that is often used to determine pressure drops through electronic boxes. The velocity head can be related to the air flow velocity as follow

$$V = \sqrt{2gH_v} \tag{8.24}$$

Where:

V = Velocity of the air
g = gravitational acceleration
H_v = velocity head in centimeter of water

The Equation 8.24 can be modified using standard air with a density of 0.0012 g/cm^3 at 20.5 ^{o}C and 1 bar, this is shown in Equation 8.25.

$$V = \sqrt{\frac{2(979.6\, cm/\sec^2)(1\, g/cm^3 water)(H_V\ cm\, water)}{0.0012\ \ g/cm^3 air}}$$

$$V = 1277\sqrt{H_v(cm\, water)} = cm/\sec. \tag{8.25}$$

The total head will be the sum of the velocity head and the static head as follow

$$H_t = H_v + H_s \tag{8.26}$$

We have many cases for measurement of the total heads inside the electronic box as shown in the Figures 8.9, 8.10, and 8.11. In case of a pressurized electronic box with no air flows the total pressure equal to the static pressure as shown in Figure 8.9. While in case of a fan blows air through the electronic box, the pressure within the box will be slightly higher than the outside air pressure. A velocity head will now be developed, as shown in Figure 8.10. But, in case of a fan draws the air through the box, the pressure within the box will be slightly lower than the outside air pressure, and the pressure head characteristics will appear as shown in Figure 8.11. And the total head is still constant as shown by Equation. 8.26.

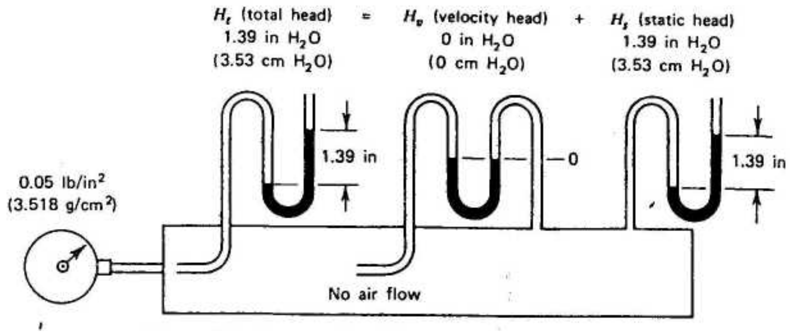


Figure 8.9 a pressurized electronic box with no air flow

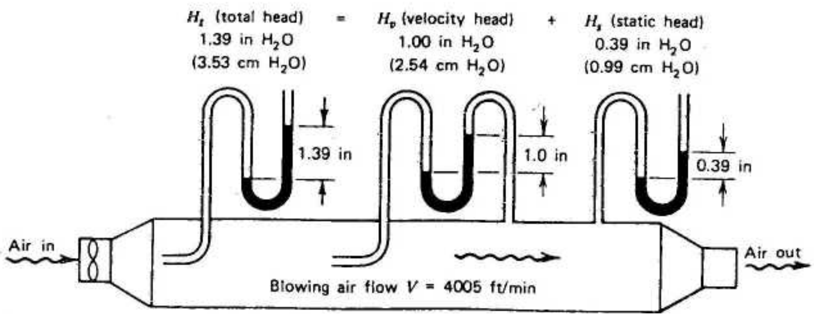


Figure 8.10 pressure head characteristics when the fan blows air through an electronic box

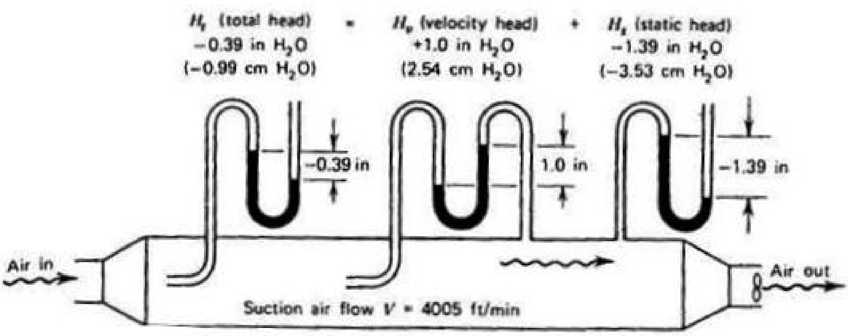


Figure 8.11 pressure head characteristics when the fan draws air through an electronic box

8.6 Fan Performance Curve

Electronic boxes that are cooled with the use of fans must be carefully evaluated to make sure the fan will provide the proper cooling. If the fan is too small for the box, the electronic system may over heat and fail. If the fan is too big for the box, the cooling will be adequate, but the larger fan will be more expensive, heavier, and will draw more power.

Air flowing through the electronic box will encounter resistance as it enters the different chambers and is forced to make many turns. The flow resistance is approximately proportional to the square of the velocity, so that it is approximately proportional to the square of the flow rate in cubic feet per minute (cfm). When the static pressure of the air flow through a box is plotted against the air flow rate, the result will be a parabolic curve. This curve can be generated by considering the various flow resistances the airflow will encounter as it flows through the box. The method of analysis is to assume several different flow rates through the box and then to calculate a static pressure drop though the box for each flow rate. This curve is called impedance curve for the electronic box as shown in Figure 8.12.

Once the box flow impedance curve has been developed, it is necessary to examine different fan performance curves to see how well the fans will match the box. A typical fan performance curve is shown in Figure 8.13.

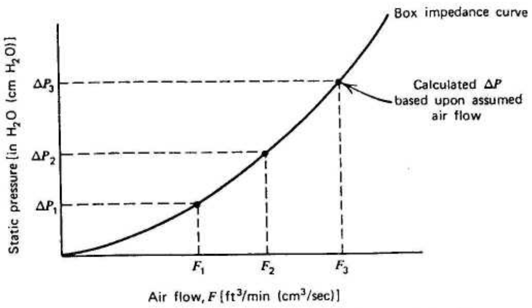


Figure 8.12 air flow impedance curve for the electronic box

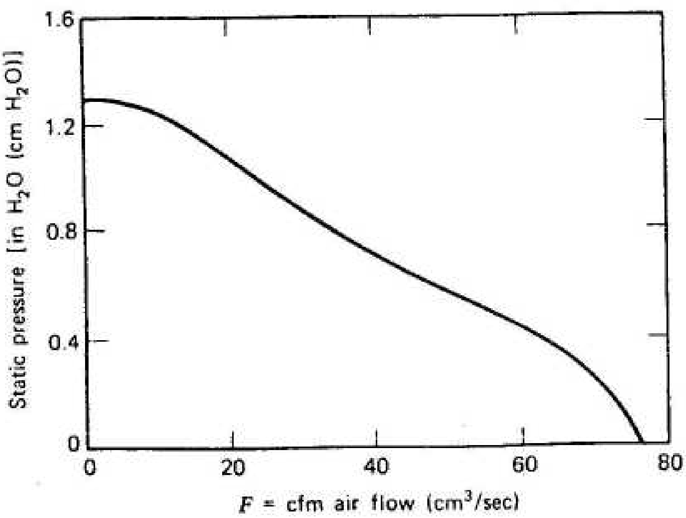


Figure 8.13 fan impedance curve

If the impedance curve for the box is superimposed on the impedance curve for the fan, they will intersect. The point of intersection represents the actual operating point for the system, as shown in Figure 8.14.

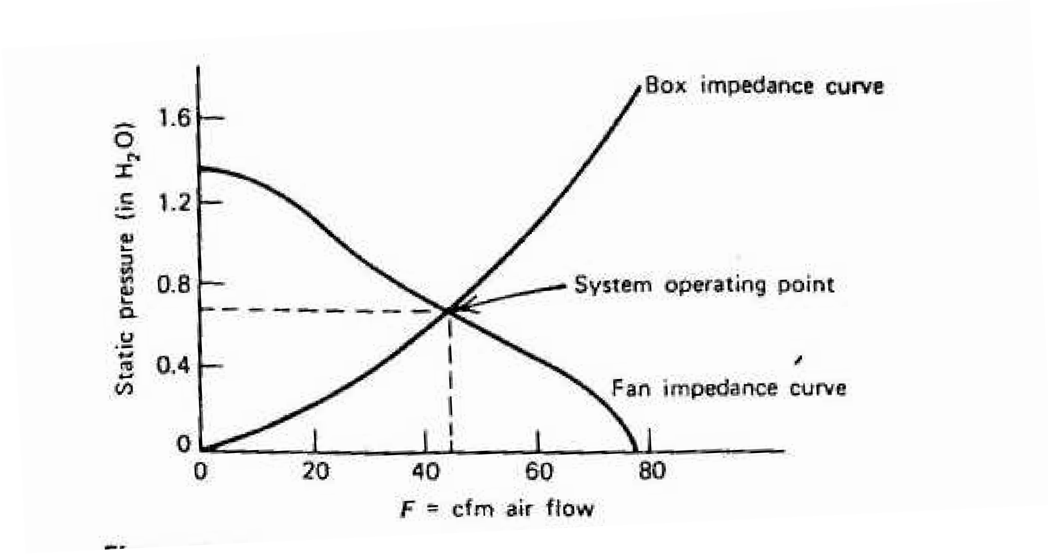


Figure 8.14 intersection of fan and box impedance curves

8.7 Cabinet Cooling Hints

In addition to selecting a fan, there may be some choice in the location of the fan or fans, and in this regard, the illustration in Figure 8.15 may prove useful. The following comments should also be kept in mind with regard to fan location:

1) Locate components with highest heat dissipation near the enclosure air exits
2) Size the enclosure air inlet and exit vents at least as large as the venturi opening of the fan used
3) Allow enough free area for air to pass with velocity less than 7 meters/sec
4) Avoid hot spots by spot cooling with a small fan
5) Locate components with the most critical temperature sensitivity nearest to inlet air to provide the coolest air flow
6) Blow air into cabinet to keep dust out, i.e. pressurize the cabinet
7) Use the largest filter possible, in order to reduce pressure drop and keep the system from the dust

8.8 Design Steps for Fan-Cooled Electronic Box Systems

The system as shown in Figure 8.16 must be capable of continuous operation in a 55 °C (131 °F) ambient at sea level condition. The maximum allowable hot spot component surface temperature is limited to 100 °C (212 °F). The system contains seven PCBs, each dissipating 20 watts, for a total power dissipation of 140 watts. This does not include fan power dissipation.

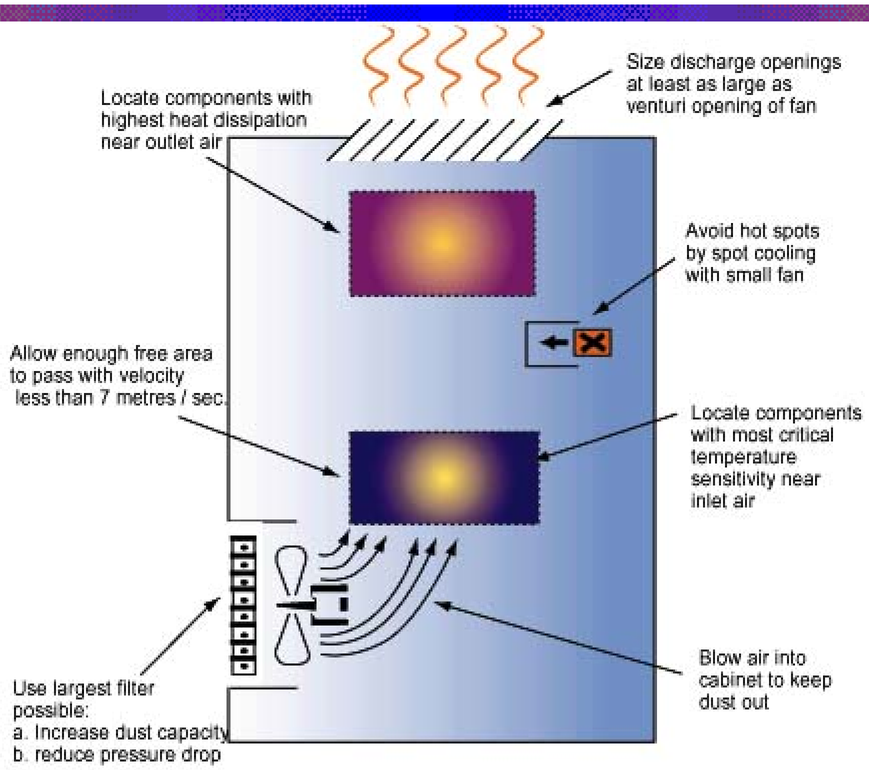


Figure 8.15 Cabinet cooling hints

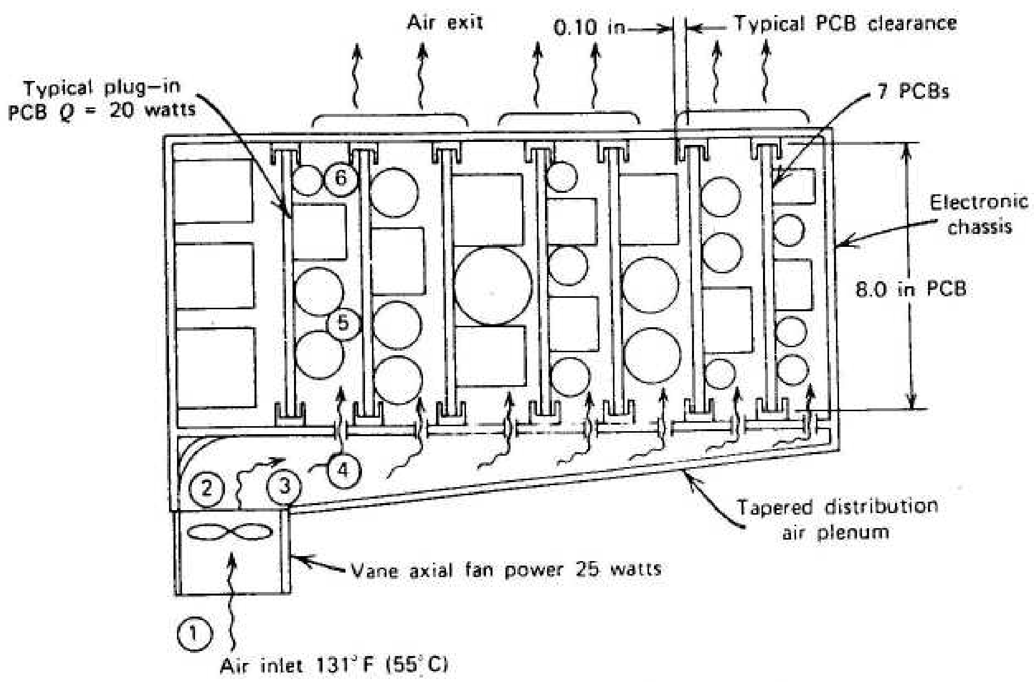
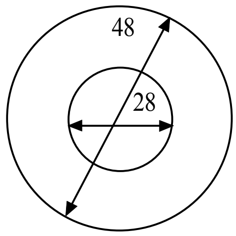


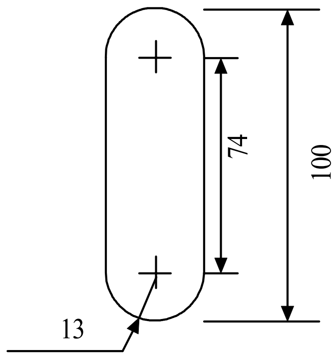
Figure 8.16 plan view of fan-cooled electronic box

The flow area at the partitions designed on the drawing are as follows

Inlet to fan is an annuals (dimensions are in mm)



90^o turn and transition from a round section at the fan to an oval section

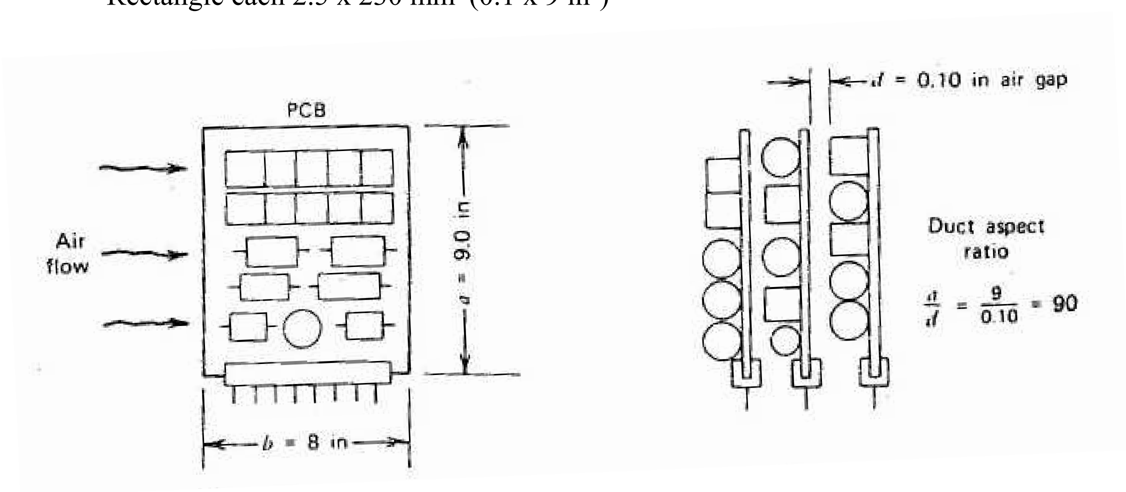


Contraction and transition to rectangular section a rectangle 8 x 125 mm2

Plenum entrance to a PCB duct rectangle each 1.5 x 155 mm2

PCB channel duct as shown in the following drawing

Rectangle each 2.5 x 230 mm^2 (0.1 x 9 in^2)



Design procedures:

Two fans are available for cooling the box: Fan A is three phase 15500 rpm that has a 25 Watts motor. Fan B is single phase, with an 18 Watts motor that operates at 11000 rpm at sea level. The box must be examined in two phases to ensure the integrity of the complete design. In phase 1, the thermal design of the box is examined, with the proposed fan, to make sure the component hot spot temperature of 100°C (212°F) is not exceeded. In phase 2, the electronic chassis airflow impedance curve is developed and matched with several fans, to make sure there is sufficient cooling air available for the system.

Phase 1: Electronic box thermal design:
To be on the safe side, base the calculation on the use of larger, 25 W motor, Fan A
The total energy to be dissipated would then be
$q = 140 + 25 = 165$ W
Air required for cooling is

$$m^{\bullet} = \frac{q}{c_p (t_{a,o} - t_{a,i})}$$

Past experience with air-cooled electronic systems has shown that satisfactory thermal performance can be obtained if the cooling air exit temperature from the electronic chassis does not exceed 70 °C (160 °F), so that assume exit air temperature $t_{a,o}$= 70 °C

From the air property table at mean temperature t_m= (70+55)/2 = 62.5 °C

$\rho = 1.052$ kg/m^3
$\nu = 19.23 \times 10^{-6}$ m^2/s
Pr = 0.7
$C_p = 1008$ J/kg.K
k = 0.0289 W/m..°C

Air required for cooling is

$$m^{\bullet} = \frac{165}{1008(70-55)} = 0.01091 \text{ kg/s}$$

By calculating the heat transfer coefficient between the air and PCB's and, hence, the temperature rise of the PCB's above the ambient air. For this purpose we calculate the Reynolds' number

$$\text{Re} = \frac{\text{V}D_H}{\nu}$$

$$\because D_H = \frac{4A}{P} = \frac{4 \ x \ 9 \ x 0.1}{2(9+0.1)} = 0.197 \ in = 0.005 \ m$$

$$and \quad \text{V} = \frac{m^{\bullet}}{\rho \ A} = \frac{0.01091}{1.052 \ (7x9x0.1)x(0.0254)^2} = 2.55 \ m/s$$

$$\therefore \text{Re} = \frac{2.55 \ x \ 0.005}{19.23 \ \text{x}10^{-6}} = 663$$

The heat transfer coefficient for laminar flow through ducts is shown in the following relation

$$\frac{\bar{h} \ D_H}{k} = \overline{Nu}_D = 1.86 \left(\frac{\text{Re} \ \ \text{Pr}}{L / D_H} \right)^{1/3}$$

$$= 1.86 \left(\frac{663 \ x 0.7 \ x 0.005}{8 \ x \ 0.0254} \right)^{1/3} = 4.19$$

$$\bar{h} = 24.2 \ \text{W/m}^2.\text{K}$$

The total heat transfer is

$$q = h \ S_{eff} \ \Delta t_h$$

Actually, the back surface of a PCB is not available for heat transfer; the practice is to assume 30 percent only available for this purpose

$$S_{eff} = 7\ x1.3\ x8\ x\ 9\ (0.0254)^2 = 0.423\ m^2$$

$$\Delta t_h = 165/24.2\ x0.423 = 16.1\ ^oC$$

Therefore, maximum component surface temperature

$$t_{max} = t_{a,o} + \Delta t_h = 70 + 16.1 = 86.1\ ^oC$$

This is acceptable surface temperature since it is less than 100 °C

Phase 2: Electronic chassis air flow impedance curve:
The air flow conditions are examined at six different points in the chaises, where the maximum static pressure losses are expected to occur, as shown in Figure 8.16. These static pressure losses are itemized as follow:

1- Air inlet to fan
2- 90^o turn and transition to an oval section
3- Concentration and transition to a rectangular section
4- Plenum entrance to PCB duct
5- Flow through PCB channel duct
6- Exhaust from PCB duct and chaises

The following table gives the ratio between static head loss to velocity head at the different positions

Position number	H_s / H_v
1	1
2	0.9
3	0.4
4	2
5	1
6	1

The flow areas at each position are:

Position 1: $A_1 = \frac{\pi}{4}\left(d_o^2 - d_i^2\right) = \frac{\pi}{4}\left(0.048^2 - 0.028^2\right) = 1.194\ x10^{-3}\ m^2$

Position 2: $A_2 = 0.026\ x0.074 + \pi\ x0.013^2 = 2.455\ m^2$

Position 3: $A_3 = 0.008\ x0.125 = 1\ x10^{-3}\ m^2$

Position 4: $A_4 = 0.0015\ x0.155\ x7slots = 1.628\ x10^{-3}\ m^2$

Position 5: $A_5 = 0.0025\ x0.23\ x7 = 4.025\ x10^{-3}\ m^2$

Position 6: $A_5 = A_6 = 4.025\ x10^{-3}\ m^2$

The following table gives velocity, velocity heads at each position at 10 cfm (0.283 m^3/min)

Position	V (cm/s.)	H_v(cm H_2O)
1	400	0.098
2	376	0.087
3	488	0.146
4	290	0.0516
5	116	0.0082
6	116	0.0082

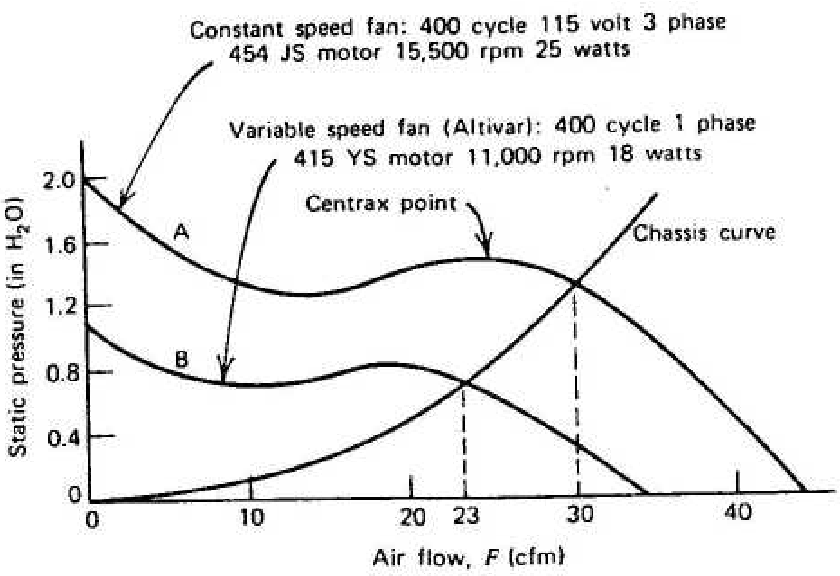
Performing the test under different cfm air flow: let the flow rate also at 20 cfm, and 30 cfm
The losses for each another flow calculated from

$$H = (H_s / H_v)\, H_{v,10} \left(\frac{V^\bullet}{10}\right)$$

The following table gives the static pressure loss in (cm H_2O) at 10 cfm, 20 cfm, and 30 cfm

Position	10 cfm	20 cfm	30 cfm
1	0.098	0.392	0.882
2	0.086	0.345	0.777
3	0.0729	0.292	0.657
4	0.103	0.411	0.927
5	0.0082	0.0325	0.0731
6	0.0082	0.0325	0.0731
Total	0.3761	1.5	3.3892

Then drawing the chassis air flow impedance curve at different fan curves as shown below



The minimum flow rate required for this system is

$$V^\bullet = \frac{m^\bullet}{\rho} = \frac{0.01091}{1.052} = 0.01\, m^3/s = 10370.7\, cm^3/s$$

From the impedance curve it shows:

The flow rate supplied by fans A is

$$V^{\bullet} = 30\text{cfm} = 14157\ cm^3 / s$$

The flow rate supplied by fans B is

$$V^{\bullet} = 23\text{cfm} = 10854\ cm^3 / s$$

So that both fans A and B can supply more than the minimum required flow rate, either fan will be acceptable.