Analysis And Preliminary Design Of An Axial Flow Compressor

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Abstract- *Analysis of an axial flow compressor involves, analyzing air flow angles at various stages and calculating the pressure ratio between last stage and the first stage with the help of ambient parameters. Temperature, Mach no. and axial flow velocity are treated in a ratio and other parameters are calculated using Euler's turbo machinery equations and velocity triangles. An attempt has been made to conduct an analysis on a real time 1-stage compressor and accumulating the flow parameters, readings and result and furthermore, design a 6-Stage compressor from the parameters obtained.*

Keywords- Analysis, design, Euler, turbo, machinery, flow, compressor, parameters, ratio, equations, temperature, pressure, stage.

I. INTRODUCTION

Axial flow Compressors

This paper revolves around the concept of an axial flow compressor. An axial flow compressor is termed so because the flow that enters this turbo machine is parallel to the axis of this compressor or in simpler terms parallel to the axis of the rotating shaft of the compressor and exits in the same direction too.

These blades are of two types:

1) Rotating Blades or Rotor.
2) Stationary Blades or Stators

A pair of rotor and stator blade combines to form a stage. So, an axial flow compressor is also sometimes said to be a 3-stage or a 4-stage compressor. It compresses the air or gas through it by first accelerating the flow through rotor and then diffusing it through stator which causes an increase in pressure (important parameter is pressure) of the air flow. In addition to the rotor blades, another row of variable blades known as the inlet guide vanes are placed at the entry of the compressor. These inlet guide vanes ensure that the incoming air enters at the needed or desired flow angles. These guide vanes may also be pitch variable and can be used in order to vary the flow as per requirements. Every stage provides a little increase in the pressure and when several such stages are placed together, they can create a huge pressure rise which is exactly what is needed by any compressor. (Integrated Publishing, n.d.)

II. LITERATURE REVIEW

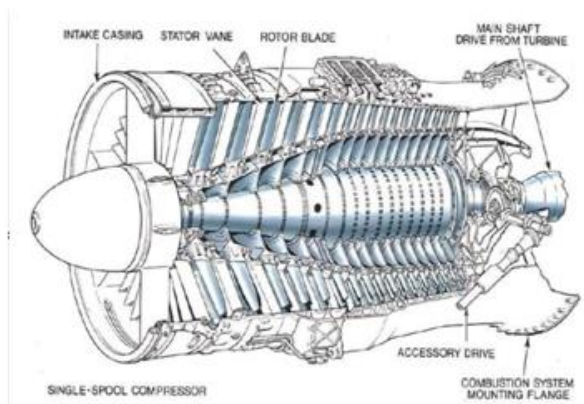


Figure (1) Single spool Axial Flow Compressor

Euler's Turbo machinery Equations

For any compressor with a steady flow the applied torque is equal to the change in the

Angular momentum of the fluid, mathematically:

$$\tau_A = \dot{m}\,(r_e v_e - r_i v_i) \qquad (1)$$

where, τ_A is the Torque applied, $\dot{m}$ is the Mass flow rate, r_e is the radius at exit, r_i is the radius at inlet, v_e is the velocity at exit, v_i velocity at inlet

The input power or the work done by the shaft can be written as:

$$Wc = \omega\tau A = \dot{m}\omega(reve - rivi) \qquad (2)$$

Where, W_c is work done.

Under ideal conditions, for turbo machinery of a compressible gas the specific heat can be assumed to be constant hence we can write:

$$c_p(T_t e - T_t i) = \omega(r_i v_i - r_e v_e) \qquad (3)$$

Where, c_p is the coefficient of heat at constant pressure.

Velocity Triangle

The flow through the different stages of a compressor is analyzed by the help of velocity triangles. The velocity triangle is a vector triangle that consists of the different velocities that happen to occur during the flow through a stage.

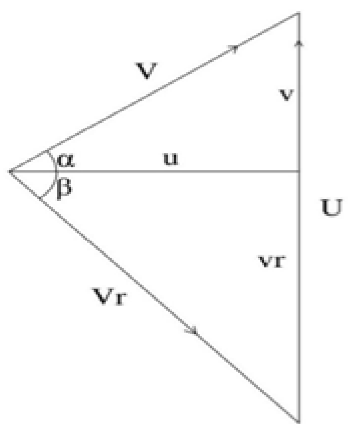


Figure (2) Velocity Triangle

V = Entry Velocity of flow
U = Tangential Velocity of Rotor Blades
V_r = Relative Velocity
α = Absolute Angle
β = Relative Angle
u = Axial Velocity of flow
v = tangential velocity of flow
v_r = relative velocity

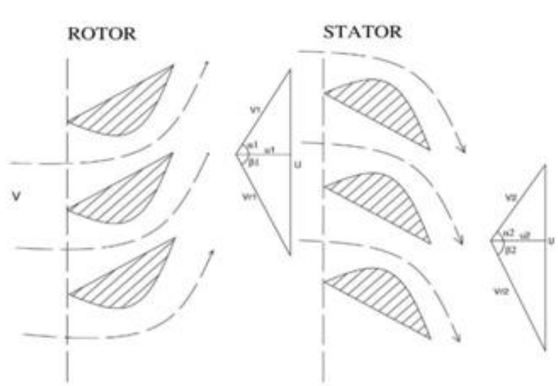


Figure (3) Stage Flow velocity triangles

III. METHODOLOGY AND CALCULATIONS

Analysis of an axial flow compressor consists of many steps; we start with the known parameters. Then we use the velocity triangle to conduce all the unknown values from all the known ones.

Analysis of the first stage is shown with all the important equations.

NOTE: analysis of the other stages is similar to the analysis of the first stage.

Parameters:

- Inlet Velocity = V_1 (m/s)
- Mach No. = M_1
- Temperature (T_1) = T_1 (K°)
- Absolute Temperature (T_{t_1}) = T_{t_1}(K°)
- Absolute Pressure (P_{t_1}) = $P_{t_1}(Bar)$
- Tangential Velocity = *U (m/s)*
- Axial Velocity Ratio = η
- Incidence angle = α°

STAGE 1- ROTOR

$a_1 = \sqrt{\gamma R T_1}$ (Speed of sound)
$V_1 = M_1 a_1$
$u_1 = V_1 \cos(\alpha_1)$ (From Velocity Triangle)
$v_1 = V_1 \sin(\alpha_1)$ (From Velocity Triangle)
v_{1R} = U - v_1
$\beta_1 = \tan^{-1}(\frac{v_{1R}}{u_1})$ (From Velocity Triangle)
$V_{1R} = u_1/\cos(\beta_1)$

$$P_1 = \frac{P_{t_1}}{\left(1 + \left(\frac{\gamma - 1}{2}\right) M_1{}^2\right)^{\frac{\gamma}{\gamma - 1}}}$$

STAGE 1 – STATOR

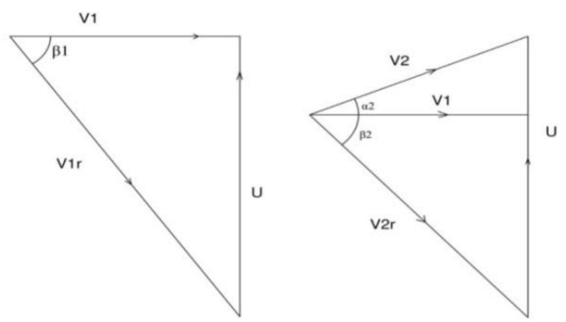
$\frac{u_1}{u_2}$ = η and $c_p\left(T_{t_2} - T_{t_1}\right) = U(v_2 - v_1)$

From Velocity Triangle

$\rightarrow c_p\left(T_{t_2} - T_{t_1}\right) = U(u_2 \tan(\alpha_2) - u_1 \tan(\alpha_1))$ because $\alpha_2 = \beta_1$.

$$\alpha_2 = \tan^{-1}\{\frac{1}{u_2}\left(\frac{c_p}{U}\left(T_{t_2} - T_{t_1}\right) + u_1 \tan(\alpha_1)\right)\} V_2$$

$$= u_2/\cos{(\alpha_2)}T_2 = T_{t_2} - \frac{{V_2}^2}{2c_p}$$
$$a_2 = \sqrt{\gamma R T_2}$$
$$M_2 = \frac{V_2}{a_2}$$
$$v_2 = V_2 \sin{(\alpha_2)}$$
$$v_{2R} = U - v_2$$
$$\beta_2 = tan^{-1}\left(\frac{v_{2R}}{u_2}\right)$$
$$\Delta Ts = \frac{\lambda U u_1(\tan{(\beta_1)} - \tan{(\beta_2)})}{cp}$$
$$V_{2R} = \frac{u_2}{\cos{(\beta_2)}}$$
$$P_{t_2} = \left(\frac{T_{t_2}}{T_{t_1}}\right)^{\frac{\gamma}{\gamma-1}} P_{t_1}$$
$$P_2 = \frac{P_{t_2}}{\left(1+\frac{\gamma-1}{2}{M_2}^2\right)^{\frac{\gamma}{\gamma-1}}}$$



Fig(v)

Figure (4) Rotor and Stator Velocity triangles

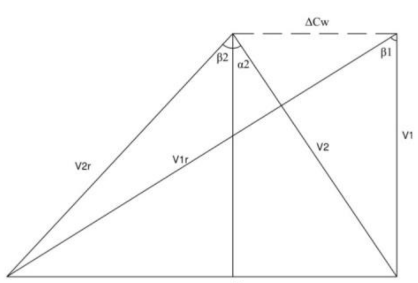


Fig (vi)

Figure (5) Combined velocity triangle for stage 1

The above figures portray the velocity triangles for the 1-stage compressor. Here we observe that the axial flow velocity is equal to V_1, this is due to the absence of the I.G.V as discussed above. Calculating the unknown parameters in the above figure:

$$V_{1r} = \sqrt{U^2 + V_1^2} = 386.94\ m/s$$
$$\beta_1 = cos^{-1}\left({V_1}/{V_{1r}}\right) = 64.79^\circ$$
$$T_1 = T_{01} - \frac{V_1^2}{2c_p} = 289.45\ k$$
$$P_1 = P_{01}\left(\frac{T_1}{T_{01}}\right)^{\gamma/\gamma-1} = 1.01\ bar$$
$$M_1 = \frac{V_1}{\sqrt{\gamma R T_1}} = 0.483$$

An important factor that is present for every compressor that has been designed or is to be designed is the De Haller criterion. Due to high speed fluid flow we see some deflections between the rotor and stator. The amount of deflection required is shown by the relative velocities that enter each stage V_1 and V_2, and also the change in whirl velocity ΔC_w. Considering a fixed value of β_1 , it is obvious that, increasing the deflection by reducing β_2 entails a reduction in V_2. In other terms, high fluid deflection means high rate of diffusion. Hence the allowable diffusion is assessed by the de Haller number, defined as $\frac{V_2}{V_1}$; a limit of $\frac{V_2}{V_1} \nless 0.72$ was set, lower values leading to excessive losses. Because of its simplicity the de Haller no. is stilled used for preliminary analysis and design, but for final design calculations the diffusion factor is preferred.

$$\therefore\ V_2 \nless 0.72 \times V_1 V_2 = 118.8\ m/s$$

It was seen that a rise of 15° was effective across the stage and to calculate the flow angles and whirl velocity at stator, it is continued as:

$$\Delta C_w = \frac{\Delta T_s C_p}{U} \ \Rightarrow\ C_{w2} - C_{w1} = \frac{\Delta T_s C_p}{U}\ ,$$

But C_{w1} is 0 because of incident angles of the inlet velocity due to absence of IGV.

$$\therefore\ C_{w2} = \frac{\Delta T_s C_p}{U} = 43.07\ \text{m/s}$$
$$\beta_2 = tan^{-1}\left(\frac{U - C_{w2}}{V_1}\right) = 61.67^\circ$$
$$\alpha_2 = sin^{-1}\left(\frac{C_{w2}}{V_2}\right) = 21.26^\circ$$
$$M_2 = \frac{V_2}{\sqrt{\gamma R T_2}} = 0.33$$

It can be seen that M_2 which is the Mach number at the stator is less than M_1 which is the mach number at the rotor. This alone shows that the flow is decelerated at the stator and all the kinetic energy from the rotor is converted to a rise in pressure.

$$T_{02} = T_{01} + \Delta T_s = 318^{\circ}K$$

$$P_{02} = P_{01}\left(\frac{T_{02}}{T_{01}}\right)^{\gamma/\gamma-1} = 1.19\ bar$$

Pressure ratio across the compressor =

$$\frac{P_{02}}{P_{01}} = 1.184$$

Hence the above analysis gives the estimate of the pressure ratio across one stage which increases further stage by stage. Apart from that the flow angles are also calculated for a 1-stage compressor.

Table (1) 1st Stage analysis parameters

	α(Deg)	β (Deg)	T_0(K)	P_0(bar)	M
Rotor	0	64.79	303	1.01	0.48
Stator	21.26	61.67	318	1.19	0.33

Designing a 6-stage compressor

The conditions that we are taking for the design are similar to the 1-stage compressor; hence a suitable design point may emerge as follows:

Compressor pressure ratio = 5.15

Air mass flow = 20 kg/s
Pressure = 1.01 bar
Temperature = 303 K

With these data specified, it is now necessary to investigate the aerodynamic design of the compressor. It is assumed that this compressor also does not have any inlet guide vanes.

Determination of rotational speed and annulus dimensions

The previous analysis was done for a compressor that was running on a blade speed of 350 m/s so, it would be wise to keep that as a constant parameter for the design. Besides that, previous experience also suggests that a blade speed of 350 m/s will lead to an axial velocity between 150 to 200 m/s. The axial velocity we saw earlier was 165 m/s and this design proceeds with it. The hub-tip ratio varies between 0.4 – 0.6 and the tip radius of the blade is a function of the hub-tip ratio. To satisfy continuity:

$$m = \rho A v$$

$$\frac{m}{\rho v} = \pi r_t^2 - \pi r_r^2$$

$$\Rightarrow \pi r_t^2\left(1 - \frac{r_r^2}{r_t^2}\right)$$

$$r_t = \sqrt{\frac{m}{\rho v \pi\left(1 - \frac{r_r^2}{r_t^2}\right)}}$$

Here r_t is the tip radius, r_r is root radius or hub radius, ρ is the density of air, v is the axial velocity of air and m is the mass flow rate.

$$T_1 = T_{01} - \frac{v^2}{2c_p}$$

Substituting the values, we get $T_1 =$ 289.45 K,

$$P_1 = P_{01}\left(\frac{T_1}{T_{01}}\right)^{\frac{\gamma}{\gamma-1}} = 0.86 \text{ bars}$$

$$\rho_1 = \frac{P_1}{RT_1} = 1.035\ kg/m^3$$

We have now calculated all the necessary values to calculate the variance of the tip radius with the hub-tip ratio.

$$r_t = \sqrt{\frac{m}{\rho v \pi\left(1 - \frac{r_r^2}{r_t^2}\right)}} \Rightarrow \sqrt{\frac{20}{1.035 \times 165 \times \pi \times \left(1 - \frac{r_r^2}{r_t^2}\right)}}$$

Another important factor to keep in mind is the revolutions per second of the blade, that will help decide the appropriate rotational speed accordingly as per design.

$$U_t = 2\pi r_t N \Rightarrow N = \frac{U_t}{2\pi r_t}$$

As discussed above, the hub-tip ratio is a subject to variance between 0.4 – 0.6 and due to this we will find variance in the tip ratio and also the revolutions per second. Nevertheless, by this method a wide number of design options will come upfront out of which one suitable is to be chosen.

Varying the hub-tip ratio (H_{tr}) and tabulating the resulting data, we have:

Table (2) Hub Tip ratio – Revolutions

H_{tr}	r_t (m)	N (rev/s)
0.41	0.211	262.017
0.42	0.212	260.661
0.43	0.213	259.267
0.44	0.214	257.832
0.45	0.216	256.357

From the above tabulated data 255 revs/s is suitable and will prove to be a good factor for designing. Hub-tip ratio corresponding to 255 revs/s is 0.468 and the tip radius is 0.2185 meters.

$$r_r = r_t \times 0.468 = 0.1024$$

Inlet Annulus dimensions:

$r_t = 0.2185\ m$
$r_r = 0.1024\ m$
N = 255 revs/s

Pressure ratio to attain at the exit of the compressor $P_f = 4.15$
Static pressure at the exit = $P_{02} = P_{01} \times P_f = 4.19\ Bar$
$T_{02} = P_f^{\frac{1}{\eta}\times\frac{\gamma-1}{\gamma}} \times T_{01} = 476.04\ \text{K}$
$\eta = 0.9$ = Polytrophic efficiency
$T_2 = T_{02} - \dfrac{v^2}{2c_p} = 462.51\ \text{K}$
$P_2 = P_{02}\left(\dfrac{T_2}{T_{02}}\right)^{\frac{\gamma}{\gamma-1}} = 3.7877\ \text{bar}$
$\rho_2 = \dfrac{P_2}{RT_2} = 2.85\ kg/m^3$

P_2, T_2, ρ_2 are all parameters at the exit or outlet of the compressor, to calculate the exit annulus dimensions we use the same concept of continuity but this time around a mean radius that is common for the entire compressor.

Mean radius can be calculated as follows:

$$r_m = \frac{r_t + r_r}{2} = 0.1604$$

As per continuity, $\dot{m} = \rho A v$ the compressor is more or less in a cylindrical shape hence to estimate its blade height we can calculate its area as,

$$A = 2\pi r_m h \implies h = \frac{A}{2\pi r_m}$$
$$A = \frac{m}{\rho_2 v} \implies 0.0425\ m^2$$
$$h = \frac{A}{2\pi r_m} = 0.0421\ m$$

Exit Annulus Dimensions:

$$r_t = r_t - \frac{h}{2} = 0.1974$$
$$r_r = r_r + \frac{h}{2} = 0.1234$$

Table (3) Annulus Dimensions

Annulus Dimensions	r_t (m)	r_r (m)
Inlet	0.2185	0.1024
Exit	0.1974	0.1234

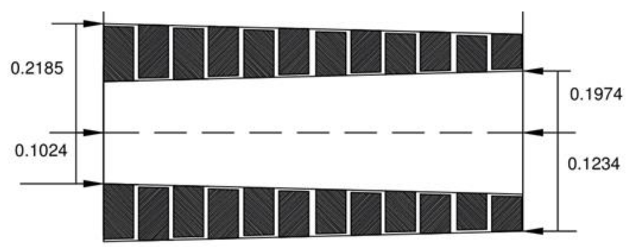


Figure (6) 2D Compressor design

Calculation of air flow angles

We have designed the basic structure of a compressor including its annulus dimensions and number of stages in it. Now we calculate the air flow angles at every stage for a better understanding of the blades and how the pressure rises in between stages.

We first assume the work done factor:

λ_2 = 0.93 for stage 2
λ_3 = 0.88 for stage 3
λ_4 = 0.83 for stage 4
λ_5 = 0.78 for stage 5
λ_6 = 0.73 for stage 6

STAGE 1

We need to calculate the change in whirl velocity:

$$\Delta T_{0s} = \frac{\lambda U_m \Delta C_w}{c_p} \Rightarrow \Delta C_w = \frac{\Delta T_{0s}\ c_p}{\lambda U_m} = 113.92\ m/s$$

$$\Delta C_w = C_{w2} - C_{w1}$$

For the first stage $C_{w1} = 0$, Because of no IGV.

$\therefore\ \Delta C_w = C_{w2} = 113.92\ m/s$

While calculating β_2 earlier, we did not account for the change in whirl velocity. To calculate suitable value of β_2 we must calculate it in terms of C_{w2}, and from fig(5) we can see that,

$\beta_2 = tan^{-1}\left(\frac{U_m - C_{w2}}{\nu}\right)$
$\beta_2 = 40.96^{\circ}$ (Corrected value)
Similarly, $\alpha_2 = tan^{-1}\left(\frac{C_{w2}}{\nu}\right)$ from fig (vi).
$\alpha_2 = 34.64^{\circ}$

Deflection of Rotor blade: It is due to the change in the relative angles between the axial velocity and the relative velocity, Deflection in rotor blades is the key reason for diffusion.

$\Delta\beta = \beta_1 - \beta_2$
$\Delta\beta$ for first stage = 16.37°

Calculating the pressure ratio of the stage:

$\frac{P_{03}}{(P_{01}}) = (1 + \frac{\eta \Delta T_{0s}}{T_{01}})^{\frac{\gamma}{\gamma-1}}$

$P_{03} = P_{01}\left(1 + \frac{\eta \Delta T_{0s}}{T_{01}}\right)^{\frac{\gamma}{\gamma-1}} = 1.32$ bar
$\eta = 0.9$
$\Delta T_{0s} = 27.05$ K
$T_{01} = 303\ K$, temperature at inlet.
$T_{03} = T_{01} + \Delta T_{0s} = 330.05\ K$
Temperature at Stage 1 exit = 330.05 K
Pressure at stage 1 exit = 1.32 bar

We have completed the air flow analysis of stage 1. Now we have to assume or set a value for α_3 the direction or angle at which the air will exit from the stator, because this α_3 will now be α_1 for the next rotor of stage 2. Hence for that we have to focus on the degree of freedom (Λ). The inlet velocity is not same for stage 2 rotor as it was for the stage 1 rotor. (HIH Saravanamutto)

STAGE 2

$\Lambda \approx 1 - \frac{C_{w2} + C_{w1}}{2U_m} \& \Lambda = \frac{\nu}{2U_m}(\tan\beta_1 + \tan\beta_2)$

Substituting values of $C_{w2}\ and\ C_{w1}$, $\Lambda = 0.78$

Λ is very high, so we assume a lower value for the degree of freedom.

$\Lambda = 0.68$ (suitable value)
Parameters for stage 2: $\Delta T_{0s} = 28.55\ K, \lambda = 0.93.$
Calculating $\beta_1 \& \beta_2 \Rightarrow$
$\Delta T_{0s} = \frac{\lambda U_m \nu(\tan\beta_1 - \tan\beta_2)}{c_p} \Rightarrow \frac{c_p \Delta T_{0s}}{\lambda U_m \nu} = \tan\beta_1 - \tan\beta_2$
$\Lambda = \frac{\nu}{2U_m}(\tan\beta_1 + \tan\beta_2)$
$\Rightarrow \frac{\Lambda 2 U_m}{\nu} = \tan\beta_1 + \tan\beta_2$

Solving the two equations for $\beta_1 \& \beta_2$ we obtain values:

$\beta_1 = 54.88^{\circ}$ and $\beta_2 = 34.74^{\circ}$
$\frac{U_m}{\nu} = \tan\alpha_1 + \tan\beta_1$
$\frac{U_m}{\nu} = \tan\alpha_2 + \tan\beta_2$

Using the above equations to calculate α_1 and α_2 we get:

$\alpha_1 = 7.81^{\circ}$ And $\alpha_2 = 40.87^{\circ}$ and $\alpha_3 = 7.81^{\circ}$
$C_{w1} = \nu \tan\alpha_1 = 22.63\ m/s$
$C_{w2} = \nu \tan\alpha_2 = 142.77\ m/s$
$\Delta C_w = C_{w2} - C_{w1} = 120.05\ m/s$, this is greater than whirl velocity for first stage because of high stage temperature rise and lower work done factor.

Fluid deflection = $\beta_1 - \beta_2 = 20.14^{\circ}$

$\frac{P_{03}}{(P_{01}}) = \left(1 + \frac{\eta \Delta T_{0s}}{T_{01}}\right)^{\frac{\gamma}{\gamma-1}}$

$P_{03} = P_{01}\left(1 + \frac{\eta \Delta T_{0s}}{T_{01}}\right)^{\frac{\gamma}{\gamma-1}}$
= 1.72bar
$T_{03} = T_{01} + \Delta T_{0s} = 358.61\ K$

STAGE 3

For a better design we take the degree of freedom 50% and let it be so for all the remaining stages.

$\Lambda = 0.50$ (Suitable value)

Parameters for stage 3: $\Delta T_{0s} = 28.55\ K, \lambda = 0.88$.

Calculating $\beta_1 \& \beta_2 \Longrightarrow$

$$\Delta T_{0s} = \frac{\lambda U_m v(\tan \beta_1 - \tan \beta_2)}{c_p} \quad \Longrightarrow \frac{c_p \Delta T_{0s}}{\lambda U_m v} = \tan \beta_1 - \tan \beta_2$$

$$\Lambda = \frac{v}{2U_m}(\tan \beta_1 + \tan \beta_2) \Longrightarrow \qquad \frac{\Lambda 2 U_m}{v} = \tan \beta_1 + \tan \beta_2$$

Solving the two equations for $\beta_1 \& \beta_2$ we obtain values:

$\beta_1 = 49.34^\circ$ and $\beta_2 = 21.54^\circ$

$$\frac{U_m}{v} = \tan \alpha_1 + \tan \beta_1$$

$$\frac{U_m}{v} = \tan \alpha_2 + \tan \beta_2$$

Using the above equations to calculate α_1 and α_2 we get:

$\alpha_1 = 21.54^\circ$ sAnd $\alpha_2 = 49.34^\circ$ and $\alpha_3 = 21.54^\circ$

$C_{w1} = v \tan \alpha_1 = 65.09\ m/s$

$C_{w2} = v \tan \alpha_2 = 191.97\ m/s$

$\Delta C_w = C_{w2} - C_{w1} = 126.87\ m/s$, this is greater than whirl velocity for first stage because of high stage temperature rise and lower work done factor.

Fluid deflection = $\beta_1 - \beta_2 = 27.804^\circ$

$$\left(\frac{P_{03}}{P_{01}}\right) = \left(1 + \frac{\eta \Delta T_{0s}}{T_{01}}\right)^{\frac{\gamma}{\gamma - 1}}$$

$$P_{03} = P_{01}\left(1 + \frac{\eta \Delta T_{0s}}{T_{01}}\right)^{\frac{\gamma}{\gamma - 1}} = 2.19 \text{ bar}$$

$$T_{03} = T_{01} + \Delta T_{0s} = 387.17\ K$$

The remaining design process proceeds ahead as above. The next 3 stages are designed using the same process above.

III. RESULTS

As inferred from above, the analysis and design procedure has been conducted using Euler's Turbo machinery equations and velocity triangles. These have been tabulated below along with suitable graphical representations.

Table (4) Air flow angles

STG	α_1	α_2	β_1	β_2	$\Delta\beta$
1	0	34.64	57.33	40.9	16.37
2	7.8	40.87	54.88	34.7	20.14
3	21.5	49.34	49.34	21.5	27.80
4	20.3	49.9	49.9	20.3	29.52
5	19.0	50.51	50.51	19.0	31.46
6	17.2	51.32	51.32	17.2	34.08

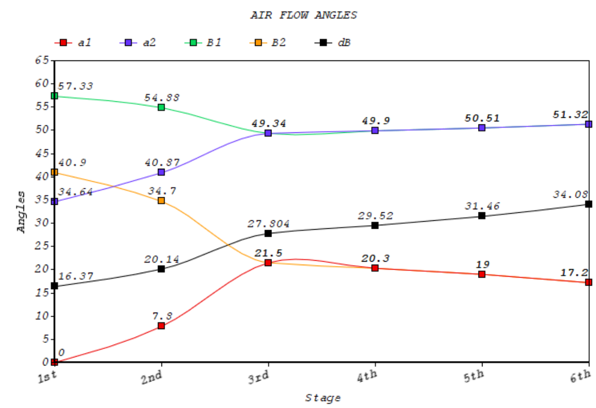


Figure (4) Air flow angles

Table (5) Other parameters

Stages	C_{w_1} (m/s)	C_{w_2} (m/s)	ΔC_w	P_{0_3} (bar)	T_{0_3} (k)
1	0	113.9	113.9	1.32	330.0
2	22.6	142.7	120.0	1.72	358.6
3	65.0	191.9	126.8	2.19	387.1
4	61.2	195.7	134.5	2.74	415.7
5	56.9	200.1	143.1	3.38	444.2
6	51.1	205.9	154.7	4.189	475.1

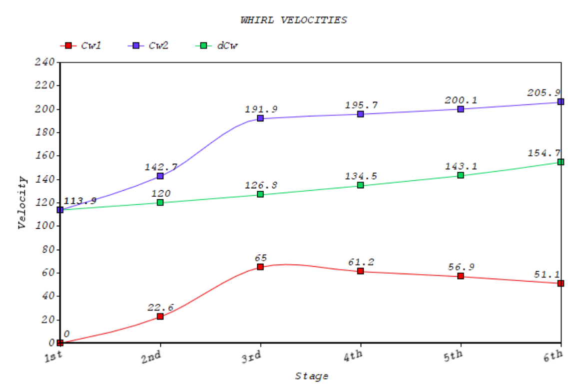
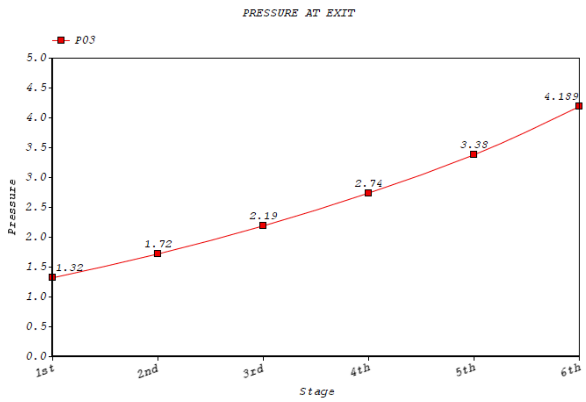


Figure (5) Whirl Velocity



Figure(6) Pressure Variance

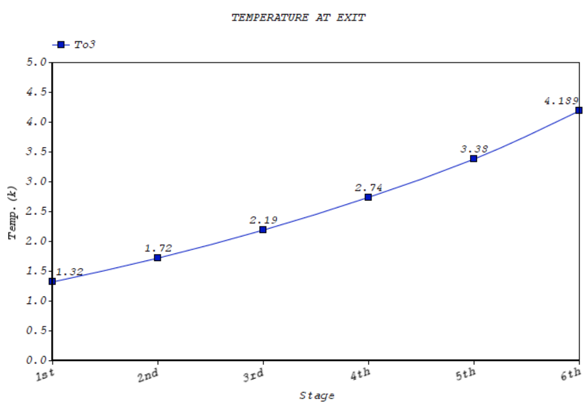


Figure (7) Temperature Variance

Above are the results of the design for a 6-stage compressor, but these are just the theoretical parameters that are needed to practical build one. A practical design can be seen as:

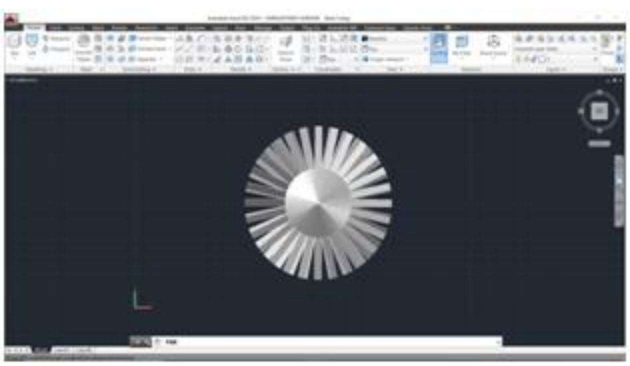


Figure (8) Top View

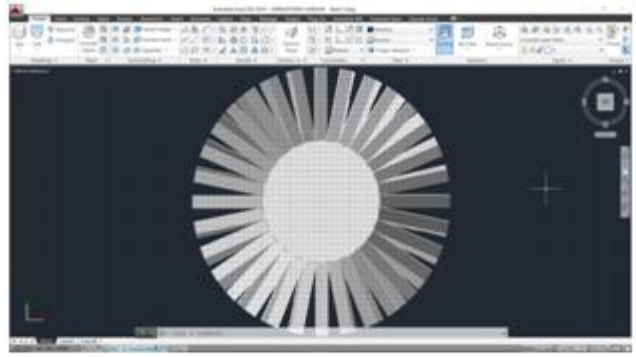


Figure (9) Bottom View

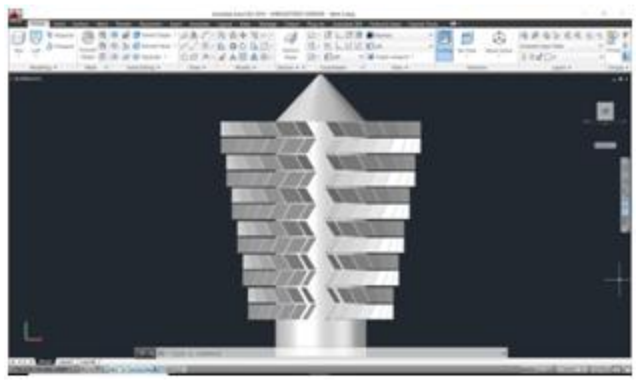


Figure (10) Side View

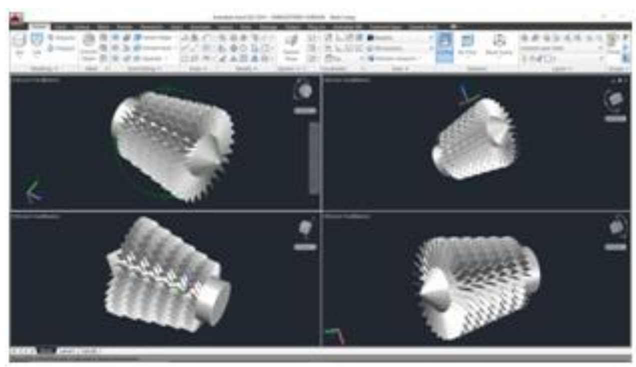


Figure (11) Multiple View port

IV. CONCLUSION

All the preliminary calculations have been carried out on the basis of a constant mean diameter. Another solution is to design the compressor for a constant outer diameter because, this allows the mean blade speed to increase along with the stage number as a result of which for a given rise in temperature the difference in whirl velocity (ΔC_w) reduces. The de Haller number also increases and a reduction in the fluid deflection is seen. Since the blade speed is high, a higher temperature rise can be seen in the later stages and also consequently a higher pressure ratio is obtained. The analysis of the 1 stage compressor made it easier to design a 6-stage compressor as it provided the necessary parameters essential to a preliminary design. Frontal area of the compressor is of critical importance as it defines the amount of air to be pulled inside and for this purpose the mean radius design was selected.

VI. FUTURE SCOPE WORK

Designing and modifications are the two fields that show a potential growth in the near future, and the basics of design require the initial requirements in the first place to begin with. These calculative methods can be applied to conduct a cumulative analysis or even to design a complete jet engine. Further analysis and research can be conducted after experimentations with new technology. For example, completely electrifying the jet engine, for this purpose the design would now also focus on the electrical calculations apart from the mechanical ones. This seems unreliable now, but would require a lot of experimentations and work if it is to happen. With technological advancement, the parameters are always changing. So the design is to kept updated to face and tackle every designing challenge. So, it is seen to be an absolute tautology that the future of this work has to grow in order to keep up with the fast progressing industry.

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VIII. APPENDIX

List of Symbols

τ_A = Torque applied
$\dot{m}$ = Mass flow rate
r_e = Radius at exit
r_i = Radius at inlet
v_e = Velocity at exit
v_i = Velocity at inlet
W_c = Work done
c_p = Coefficient of heat at constant pressure
T_{t_e} =Temperature at exit
T_{t_i} =Temperature at inlet
V = Entry Velocity of flow
U = Tangential Velocity of Rotor Blades
V_r = Relative Velocity
α = Absolute Angle
β = Relative Angle
u = Axial Velocity of flow
v = tangential velocity of flow
v_r = relative velocity
a = Speed of sound
P_{t_1}= Absolute pressure
λ = Work done factor