Performance Evaluation Of Axial Flow Compressor Using Stages Characteristics

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Abstract

The performance characteristics of axial flow compressors due to variation in size of blade, pressure, temperature and shaft rotational speed determined the output variables such as work output and efficiency. The performance depends upon the blade diameter, mass flow rate, density of the flowing fluid, stage pressure ratio, stage delivering pressure and temperature.

This paper focus on how parameters obtained from each stage, influence the Performance of the axial compressor. These various parameters that has influence on the performance of the compressor were analyzed with tables and graphs, to show clearly their inputs to the compressor performance. For the axial compressor to deliver compressed air to the combustor for onward delivery to the turbine , each stage should be able to attained a better performance that is better than the preceding stage. This work also focus on how energy is exchange between the blades and the moving fluid (compressed air) and variation of flow over the blades.

Keywords: *Axial Velocity, Blade angle, Compressor, Mach Number, Pressure ratio, Transonic.*

I. INTRODUCTION

Axial flow compressor has a rotating blade called the rotor and a stationary blade called the stator. The combination of a stator and a rotor form a stage. Each compressor design, consist of a series of Stages. A Stage comprises of a rotor and a stator as shown in figure 1.1.

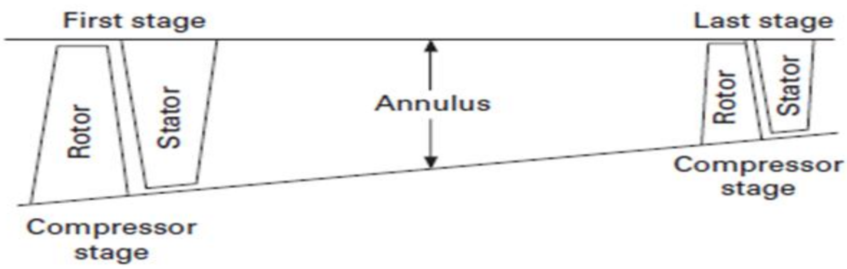

Figure 1.1: Axial compressor showing first and last stage

An inlet guide vane are provided to guide the air at the correct angle at entry to the first row of the moving blade. The rotor impart kinetic energy to the air to increase its pressure and the stator decelerate and redirect the at a Suitable angle for entry to the next row of moving blades in the proper direction(Diffusion) and convert part of the kinetic energy to pressure energy. Modern axial flow compressors may give efficiencies of 86– 90% [1].
The flow of air through the compressor is in the direction of the axis of the compressor and, therefore, it is called an axial flow compressor. The height of the blades decreases as the fluid moves through the compressor.
As the pressure increases in the direction in the direction of flow, the volume of the air decreases. To keep the air velocity the same for each stage, the blade height is reduced along the axis of the compressor.The amount of diffusion in the rotor and stator is controlled by the design of the compressor and is often called the reaction of the stage. When all the diffusion takes place in the rotor, the reaction is said to be 100%, and when all the diffusion takes place in the stator, the reaction is to be 0% (also known as impulse stage). High diffusion in the rotor or stator reduces the efficiency of the compressor and it is normal practice to design for 50% reaction in which case the diffusion is equal in the stator and rotor.

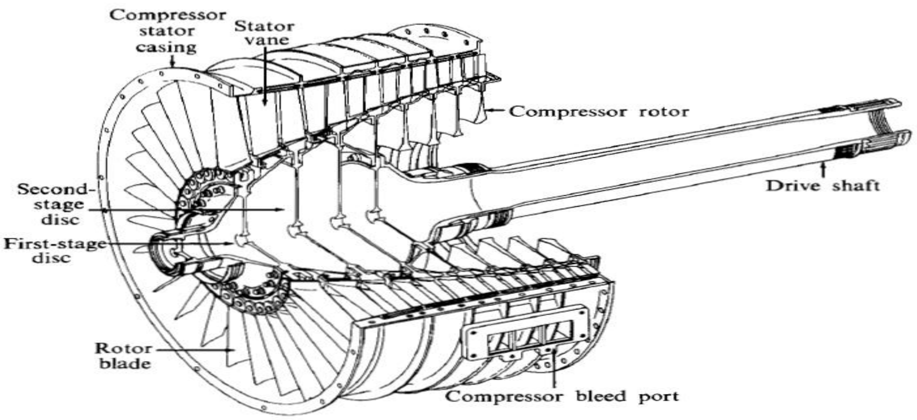

Fig 1.2 : Cutaway sketch of a typical axial compressor [2]

Axial flow compressor has the inherent ability to provide higher pressure ratio at reduce specific fuel consumption compare to a centrifugal compressor. Efficiency is higher with high overall pressure ratio. It has much larger mass flow rate that is beneficial for a given thrust per unit frontal area for jet Engines.

In aircrafts the advantage of the smaller diameter axial- flow compressor can offset the disadvantage of the increased length and weight]compared with an equivalent centrifugal compressor[3].

A. DESIGN SPECIFICATIONS

This paper is to study and verify various parameters obtain from design calculations, that affect the performance of axial flow compressor when the compressor pressure ratio, mass flow rate, turbine inlet temperature, hub-tip ratio, blade axial velocity and polytropic efficiency and thrust, are known.

Suitable design point under sea-level static condition of pressure and temperature. P_a =1.01 bar, T_a= 288K

- Compressor pressure ratio 18:1
- Air-mass flow 50kg/s
- Reaction 50%
- η_c 0.88
- Hub-tip ratio 0.5
- Axial flow velocity 150m/s
- Tip speed 360m/s

II. DESIGN PROCESS

A. ROTATIONAL SPEED

$\dot{m}= \rho_1 A V_f$ ……………….2.1

$A=\pi\ [r_t^2 - r_h^2]$ ……………….2.2

$\dot{m} = \rho_1 \pi r_t^2 V_f\ [1-(\frac{r_r}{r_t})^2]$ ……………..2.3

$r_t^2 = \frac{\dot{m}}{V_f \rho_1 \pi [1-(\frac{r_r}{r_t})^2]}$ …………….2.4

at sea-level static condition

T_{01} = 288K Assuming no loss at the intake

P_{o1} = 1.01 bar

Assuming there is no (IGV) at the intake

V_{w1} = 0 V_1=V_f= 150m/s

From flow energy equation

$T_1 = T_{o1} - \frac{V_1^2}{2\,C_p}$…………………………2.5

C_p = 1.005kJ/kgK

$T_1 = 288 - \frac{150^2}{2 \times 1.005 \times 1000}$

T_1 = 276.8K

From continuity flow equation

$P_1 = P_{01} [\frac{T_1}{T_{01}}]^{\frac{\gamma}{\gamma-1}}$ ………………………2.6

γ = 1.4

$P_1 = 1.01[\frac{276.8}{288}]^{\frac{1.4}{1.4-1}}$

P_1 = 0.879 bar

From equation of state

$\rho_1 = \frac{P_1}{R \times T_1}$ ………………….2.7

$\rho_1 = \frac{0.879 \times E5}{287 \times 276.8}$

ρ_1 = 1.106kg/m³

From equation 2.4

$r_t^2 = \frac{50}{150 \times 1.106 \times 3.142\,[1-(0.5)^2\,]}$

r_t^2 = 0.128

$r_t = (0.128)^{0.5}$

r_t = 0.36m

$\frac{r_r}{r_t} = 0.5$
$r_r = 0.36 \times 0.5$
$r_r = 0.18\text{m}$
$U_t = 360\text{m/s}$
$\boldsymbol{U_t = 2 \times \pi \times r_t \times N}$ **……………………………2.8**
$N = \frac{360}{2 \times 3.142 \times 0.36}$
$N = 159.13\text{rps}$
Rotational speed of the axial compressor is 159.13rps

B. MACH NUMBER

Assuming axial velocity to be constant across the annulus and no inlet guide vane

$\boldsymbol{V_{1t}^{\,2} = U_{1t}^{\,2} + V_f^{\,2}}$ **…………………….2.9**
$V_{1t}^{\,2} = 360^2 + 150^2$
$V_{1t} = 390\text{m/s}$
$\boldsymbol{a = (\gamma \times R \times T_1)^{0.5}}$ **………………2.10**
$a = (1.4 \times 287 \times 276.8)^{0.5}$
$a = 333.5\text{m/s}$
$\boldsymbol{M_{1t} = \frac{V_{1t}}{a}}$ **………………………………..2.11**
$M_{1t} = \frac{390}{333.5}$
$M_{1t} = 1.169$
The first stage is transonic.

C. ANNULUS DIMENSION AT EXIT FROM THE COMPRESSOR

Compressor delivery pressure

$\boldsymbol{T_{02} = T_{01}\left(\frac{P_{02}}{P_{01}}\right)^{\frac{n-1}{n}}}$ **…………………………..2.12**
$\boldsymbol{\frac{n-1}{n} = \frac{1}{\eta_c}\frac{\gamma-1}{\gamma}}$ **……………………………..2.13**
$\frac{n-1}{n} = \frac{1}{0.88}\,\frac{1.4-1}{1.4}$
$\frac{n-1}{n} = 0.325$
$T_{02} = 288(18)^{0.325}$
$T_{02} = 736.8\text{K}$
It is assume that at exit from the stator of the last stage, there is no swirl
$V_2 = V_f = 150\text{m/s}$
The static temperature, and density at exit were calculated below
$T_2 = 736.8 - \frac{150^2}{2 \times 1.005 \times 1000}$
$T_2 = 726.61\text{K}$
$P_{02} = 1.01 \times 18$
$P_{02} = 18.18\text{ bar}$
$\boldsymbol{P_2 = P_{02}\left(\frac{T_2}{T_{02}}\right)^{\frac{\gamma}{\gamma-1}}}$ **……………………2.14**
$P_2 = 18.18\left(\frac{726.61}{736.8}\right)^{\frac{1.4}{1.4-1}}$
$P_2 = 17.32\text{ bar}$
$\rho_2 = \frac{17.32 \times E5}{287 \times 726.61}$
$\rho_2 = 8.305\text{kg/m}^3$
$\boldsymbol{A_2 = \frac{\dot{m}}{\rho_2 \times V_f}}$ **………………………………2.15**
$A_2 = \frac{50}{8.305 \times 150}$
$A_2 = 0.040\text{m}^2$
Exit annulus area is 0.040m^2
$\boldsymbol{r_m = \frac{r_r + r_t}{2}}$ **………………………2.16**
$r_m = \frac{0.36 + 0.18}{2}$
$r_m = 0.27\text{m}$
$\boldsymbol{h = \frac{A_2}{2\pi r_m}}$ **………………2.17**
$h = \frac{0.040}{2 \times 3.142 \times 0.27}$
$h = 0.024\text{m}$
The blade height at exit = 0.024m
Radii of the last stator at exit are
$r_t = 0.27 + \frac{0.024}{2}$ $r_r = 0.27 - 0.012$
$r_t = 0.27 + 0.012$
$r_t = 0.282\text{m}$
$r_r = 0.27 - 0.012$
$r_r = 0.258\text{m}$

Radius	**Inlet**	**Outlet**
r_r	0.18m	0.258m
r_t	0.36m	0.282m

Table 2.1 shows the summary of the annulus dimension at inlet and outlet

III. COMPRESSION STAGES

The overall stagnation temperature rise through the axial compressor is T_{02} - T_{01} (448.8K). The stage temperature rise (ΔT_{0s}) can vary widely in different compressor designs, depending on the

$$\boldsymbol{U_m = 2\pi r_m N} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots 3.1$$

U_m = 2× 3.142×0.27× 159.13

U_m = 270 m/s

Temperature rise is calculated with equation 3.2

A. Estimation of number of stages

application and weight. Rather than choosing a value at random, it is instructive to estimate a suitable ΔT_{0s} based on the mean blade speed.[4]

$$\boldsymbol{\Delta T_{0s} = \frac{\lambda U_m V_f(\tan \text{б}_{1} - \tan \text{б}_2)}{C_p}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots 3.2$$

With an axial velocity at entry to the first stage and no inlet guide vane (IGV) From the velocity diagram in figure3.1

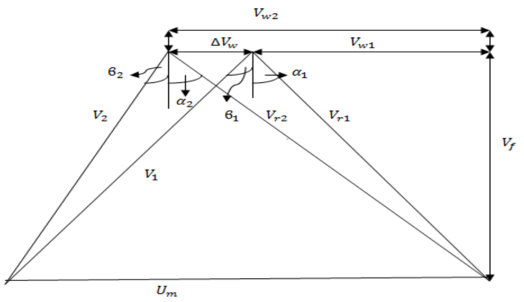

Fig 3.1: showing the velocity diagram

$$\boldsymbol{\tan \text{б}_1 = \frac{U_m}{V_f}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 3.3$$

$\tan \text{б}_1 = \frac{270}{150} = 1.8$

$\text{б}_1 = tan^{-1} 1.8$

б_1 =60.95°

$$\boldsymbol{V_1 = \frac{V_f}{\cos \text{б}_1}} \quad \ldots\ldots\ldots\ldots\ldots\ldots 3.4$$

$V_1 = \frac{150}{\cos 60.95°}$

V_1 = 308.9 m/s

To estimate possible deflection in the rotor, de Haller number criterion is applied

$$\boldsymbol{\frac{V_2}{V_1} \geq 0.72} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots 3.5$$

V_2 = 308.9 ×0.72

V_2 = 222.41m/s

$$\boldsymbol{\cos \text{б}_2 = \frac{v_f}{V_2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots 3.6$$

$\cos \text{б}_2 = \frac{150}{222.41}$

$\text{б}_2 = cos^{-1}$ 0.6744

б_2 = 47.59°

From equation 3.2 and neglecting work factor λ

$\Delta T_{0s} = \frac{150 \times 270 (\tan 60.95° - \tan 47.59°)}{1.005E3}$

ΔT_{0s} = 28.43K

Temperature rise per stage is 28.43K

$$\text{Number of stages} = \boldsymbol{\frac{\textit{Temperature rise through the compressor}}{\textit{Temperature rise per stage}}} \ldots \mathbf{3.7}$$

Number of stages = $\frac{448.8}{28.43}$

Number of stages = 16 stages

IV. STAGE DESIGN

A. STAGE ONE

From the velocity diagram in fig 3.1

$$\boldsymbol{V_{w1} = V_f \tan \alpha_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots 4.1$$

$$\boldsymbol{V_{w2} = V_{w1} + \Delta V_w} \quad \ldots\ldots\ldots\ldots\ldots\ldots 4.2$$

For the first stage α_1= 0 because no inlet guide vanes (IGV) at entry. Calculations of stage

For the purpose of this research, a value of 0.98 work-done factor (λ) is applied to stage one

$$\boldsymbol{\Delta V_w = \frac{C_p \Delta T_0}{\lambda U_m}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 4.4$$

$\Delta V_w = \frac{1.005E3 \times 28.43}{0.98 \times 270}$

ΔV_w = 107.98m/s

Since V_{w1}= 0 V_{w2} = 107.98m/s

temperature rise are based on rotor considerations only, but care must be taken to ensure that the diffusion in the stator is kept to a reasonable level.[4].

$\tan \text{б}_1 = \frac{270}{150} = 1.8$

$\text{б}_1 = tan^{-1} 1.8$

б_1 =60.95°

$$\boldsymbol{\tan \text{б}_2 = \frac{U_m - V_{w2}}{V_f}} \quad \ldots\ldots\ldots\ldots\ldots\ldots 4.5$$

$\tan \text{б}_2 = \frac{270 - 107.98}{150}$

$\tan \text{б}_2$ = 1.080

$\beta_2 = tan^{-1}1.080$
$\beta_2 = 47.21°$
$\tan\alpha_2 = \frac{V_{w2}}{V_f}$ 4.6
$\alpha_2 = tan^{-1}0.7199$
$\alpha_2 = 35.75°$
The velocity diagram of the first stage is shown in fig 1.2
The deflection of the rotor blade
$\delta_{rotor} = \beta_1 - \beta_2$ 4.7
Deflection of the rotor blade for stage one
$\delta_{r1} = 60.95° - 47.21°$
de Haller number = $\frac{cos\ 60.95°}{cos\ 47.21°}$
de Haller number = 0.714
Pressure ratio for stage one
$(\frac{P_{02}}{P_{01}})_n = (1 + \frac{\eta_c \Delta T_0}{T_{01}})^{\frac{\gamma}{\gamma-1}}$..............4.10

$(\frac{P_{02}}{1.01})_1 = (1 + \frac{0.89 \times 28.43}{288})^{3.5}$
$P_{r1} = 1.34$
$P_{02} = 1.356$ bar
$T_{02} = T_{01} + \Delta T_0$ 4.11
$T_{02} = 288 + 28.43$
$T_{02} = 316.43K$
Pressure difference in stage one
$\Delta_p = P_{02} - P_{01}$ 4.12
$\Delta_{p1} = 0.346$ bar
Degree of Reaction

$\tan\alpha_2 = \frac{107.98}{150}$
$\tan\alpha_2 = 0.7199$

$\delta_{r1} = 13.74°$
The deflection of stator blade
$\delta_{stator} = \alpha_2 - \alpha_1$ 4.8
$\delta_{s1} = 35.75°$
The diffusion can be check using de Haller number
de Haller number = $\frac{cos_{\beta 1}}{cos_{\beta 2}}$ 4.9

$\Lambda = 1 - \frac{V_{w2}+V_{w1}}{2U_m}$ 4.13

$\Lambda = 1 - \frac{107.98}{2 \times 270}$
$\Lambda = 0.80$
STAGE TWO
λ=0.96 ,Λ = 0.70
$\beta_1 = 58.43°$
$\beta_2 = 41.75°$
$\delta_{r2} = 16.68°$
$\tan\alpha_1 + \tan\beta_1 = \frac{U_m}{V_f}$ 4.14
$\tan\alpha_2 + \tan\beta_2 = \frac{U_m}{V_f}$ 4.15

4.2 WORK DONE PER STAGE

$W_{cs} = U_m(V_{w2} - V_{w1})$...............4.16

The summary calculation of the stages are shown in table 4.

Stages	P_r	P_{02} (bar)	T_{02} (K)	ΔV_w (m/s)	λ	W_{cs} (kJ)
1	1.143	1.36	316.43	107.98	0.98	29.2
2	1.150	1.78	344.86	110.07	0.96	29.7
3	1.156	2.27	373.29	112.56	0.94	30.4
4	1.163	2.86	401.72	118.05	0.92	31.9
5	1.164	3.55	430.15	116.70	0.90	31.5
6	1.172	4.33	458.58	120.28	0.88	32.5
7	1.173	5.24	487.01	123.00	0.86	33.2
8	1.18	6.45	515.44	126.00	0.84	34.0
9	1.21	7.61	543.87	129.50	0.82	35.0
10	1.22	8.92	572.30	132.30	0.80	35.7
11	1.23	10.38	600.73	135.60	0.78	36.6
12	1.24	12.16	629.16	139.20	0.76	37.6
13	1.26	14.16	657.59	142.95	0.74	38.6
14	1.28	16.37	686.02	146.97	0.72	39.7
15	1.34	18.83	714.45	151.20	0.70	40.8
16	1.34	21.52	742.88	155.55	0.68	42.0

Table 4.1 The summary calculation of other stages are shown in

V. CONCLUSION

The performance of an axial flow compressors depends on it applications. For industrial application, their flow is subsonic with inlet relative velocity mach number of 0.4 – 0.8 with stage pressure ratio of 1.05 – 1.2. For Aerospace application, its flow is transonic with inlet relative mach number of 0.7 - 1.2with stage pressure ratio of 1.15 – 1.6 [5]. This research work fall into the category of aerospace application because the inlet mach number is 1.169. From the results obtain from the calculations, figure 5.1 indicate that as the stage pressure ratio increases, the delivery temperature for each stage also increase.

This temperature increase across stages, help to increase the work input from the compressor per stage to the fluid (air).

The graph shown in figure 5.2 shows how delivery pressure varies with pressure ratio across each stage.

From basic thermodynamics, temperature is directly proportional to pressure, figure 5.3 is a verification of the thermodynamics law (Charles law).

The higher the difference between the rotor speed at exit and inlet to a stage, the higher the work available for compression. This also increase the delivery pressure per stage.

Pressure delivered across the stage increases as the pressure ratio increase, to attain the desire or design pressure output of the axial compressor. This deliver pressure from the compressor is to sustain the pressure of the combustor.

A departure from design specifications such as the flow rate will change the axial velocity component. The speed of rotation (N) of the shaft calculated, any deviation from the value will change the blade mean speed U_m, which will alter the angle β_1 at which the flow approaches the rotor.

The result obtained from this research can be functionally improve with simulation.

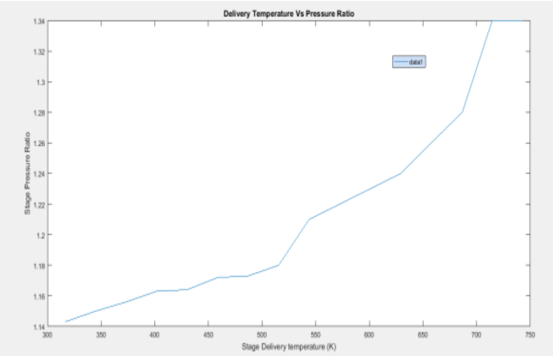

Fig 5.1 : stage pressure ratio Vs Stage deliver temperature.

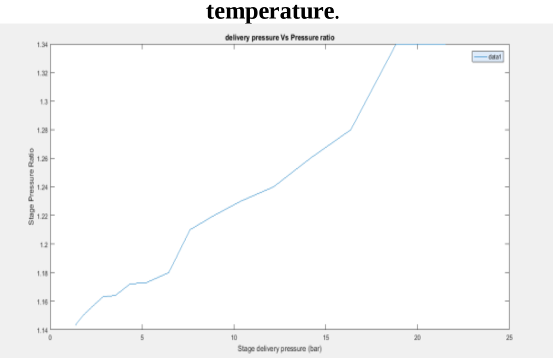

Fig: 5.2 stage pressure ratio Vs Stage deliver pressure.

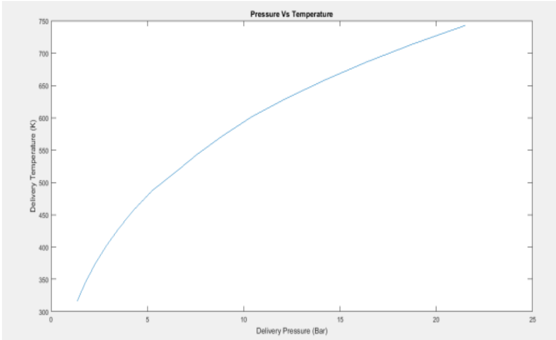

Fig 5.3: Stage temperature Vs Stage Pressure

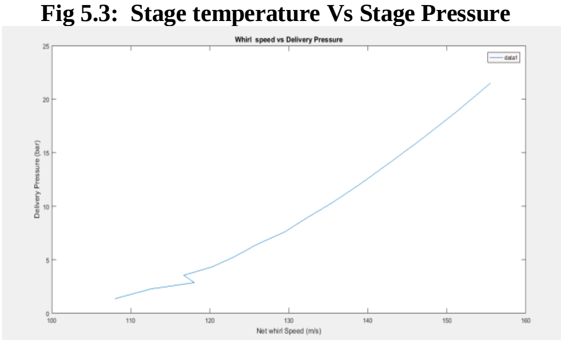

Fig5.4: Delivery Pressure Vs Net Whirl speed

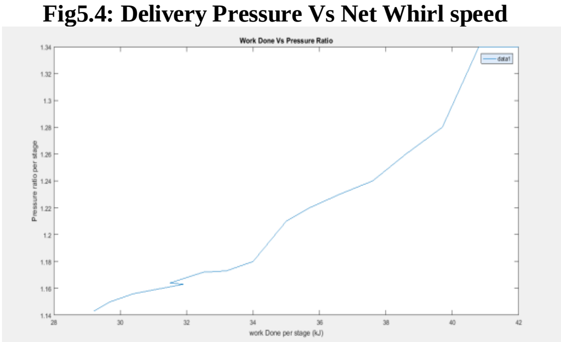

Fig 5.5: showing Work Done Vs Pressure ratio

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