Performance Evaluation Of Axial Flow Compressor Using Stages Characteristics

Avwunuketa, Ayedun Alex¹, Adamu, Mohamed Lawal²,Ajao,Tofunmi Ayodele³
Aircraft Engineering Department, Faculty of air Engineering, School of Postgraduate studies, Air Force Institute of Technology, Kaduna Nigeria.

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.

1. 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.

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.

Figure 1.1: Axial compressor showing first and last stage
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.

**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, \( T_a = 288 \)K

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

**II. DESIGN PROCESS**

**A. ROTATIONAL SPEED**

\[
\dot{m} = \rho_1 A \dot{V}_f \quad \text{.........2.1}
\]

\[
A = \pi \left( r_t^2 - r_h^2 \right) \quad \text{...........2.2}
\]

\[
\dot{m} = \rho_1 \pi r_t^3 \dot{V}_f \left[ 1 - \left( \frac{r_h}{r_t} \right)^2 \right] \quad \text{.........2.3}
\]

\[
r_t^2 = \frac{\dot{m}}{\pi \rho_1 \pi \left[ 1 - \left( \frac{r_h}{r_t} \right)^2 \right]} \quad \text{.........2.4}
\]

at sea-level static condition \( T_{01} = 288 \)K  Assuming no loss at the intake \( P_{01} = 1.01 \) bar

Assuming there is no (IGV) at the intake \( V_{w1} = 0 \quad V_1 = 150 \)m/s

From flow energy equation

\[
T_1 = T_{01} - \frac{V_1^2}{2C_p} \quad \text{...............2.5}
\]

\[C_p = 1.005 \text{kJ/kgK} \]

\[
T_1 = 288 - \frac{150^2}{2 \times 1.005 \times 1000} = 276.8 \text{K}
\]

From continuity flow equation

\[
P_1 = P_{01} \left( \frac{T_1}{T_{01}} \right)^{\frac{\gamma}{\gamma-1}} \quad \text{...............2.6}
\]

\[\gamma = 1.4\]

\[
P_1 = 1.01 \left[ \frac{276.8}{288} \right]^{\frac{1.4}{1.4-1}} = 0.879 \text{ bar}
\]

From equation of state

\[
\rho_1 = \frac{P_1}{R \times T_1} \quad \text{...............2.7}
\]

\[\rho_1 = \frac{0.879 \times 85}{287 \times 276.8} = 0.106 \text{kg/m}^3\]

From equation 2.4

\[
r_t^2 = \frac{50}{150 \times 1.106 \times 3.142 \left[ 1 - \left( 0.5 \right)^2 \right]} = 0.128
\]

\[
r_t = \left( 0.128 \right)^{0.5} = 0.36 \text{m}
\]
\[
\begin{align*}
N &= \frac{360 \times \pi \times r_t \times N}{2 \times 3.142 \times 0.36} \\
N &= 159.13 \text{ rps}
\end{align*}
\]
Rotational speed of the axial compressor is 159.13 rps

**B. MACH NUMBER**

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

\[
\begin{align*}
V_{1t}^2 &= U_{1t}^2 + V_f^2 \quad \text{....................2.9} \\
V_{1t}^2 &= 360^2 + 150^2 \\
V_{1t} &= 390 \text{ m/s}
\end{align*}
\]

\[
\begin{align*}
a &= \left( r \times R \times T \right)^{0.5} \quad \text{...............2.10} \\
&= \left( 1.4 \times 287 \times 276.8 \right)^{0.5}
\end{align*}
\]

\[
\begin{align*}
a &= 333.5 \text{ m/s}
\end{align*}
\]

\[
\begin{align*}
M_{1t} &= \frac{V_{1t}}{\sqrt{T_0}} \quad \text{....................2.11} \\
M_{1t} &= \frac{390}{333.5} \\
M_{1t} &= 1.169
\end{align*}
\]
The first stage is transonic.

\[
\begin{align*}
T_{02} &= \frac{T_{01} \left( \rho_{02} \right)^{n-1}}{\pi} \quad \text{....................2.12} \\
T_{02} &= \frac{288 \left( \frac{18}{150} \right)^{0.325}}{\pi} \\
T_{02} &= 736.8 \text{ K}
\end{align*}
\]

It is assume that at exit from the stator of the last stage, there is no swirl

\[
\begin{align*}
V_2 &= V_f = 150 \text{ m/s}
\end{align*}
\]

The static temperature, and density at exit were calculated below

\[
\begin{align*}
T_2 &= 736.8 - \frac{150^2}{2 \times 1.005 \times 1000} \\
T_2 &= 726.61 \text{ K} \\
P_{02} &= 1.01 \times 18 \\
P_{02} &= 18.18 \text{ bar} \\
h &= 0.024 \text{ m}
\end{align*}
\]
The blade height at exit = 0.024 m

Radial of the last stator at exit are

\[
\begin{align*}
r_t &= 0.27 + \frac{0.024}{2} - r_c = 0.27 - 0.012
\end{align*}
\]

\[
\begin{align*}
r_t &= 0.27 + 0.012 \\
r_c &= 0.282 \text{ m}
\end{align*}
\]

\[
\begin{align*}
r_c &= 0.27 - 0.012 \\
r_c &= 0.258 \text{ m}
\end{align*}
\]

\[
\begin{align*}
P_2 &= P_{02} \left( \frac{T_2}{T_{02}} \right)^{\frac{1}{n-1}} \quad \text{....................2.14} \\
P_2 &= 18.18 \left( \frac{726.61}{736.8} \right)^{1.4} \\
P_2 &= 17.32 \text{ bar} \\
\rho_2 &= \frac{287 \times 726.61}{8305 \times 150} \\
\rho_2 &= 0.040 \text{ m}^3 \\
A_2 &= \frac{m}{\rho_2 \times V_f} \quad \text{....................2.15} \\
A_2 &= \frac{50}{8.305 \times 150} \\
A_2 &= 0.040 \text{ m}^2 \\
r_m &= \frac{r_c + r_t}{2} \quad \text{....................2.16} \\
r_m &= \frac{0.282 + 0.27}{2} \\
r_m &= 0.27 \text{ m} \\
h &= \frac{A_2}{2 \pi r_m} \quad \text{....................2.17} \\
h &= \frac{0.040}{2 \times 3.142 \times 0.27} \\
r_t &= 0.27 + 0.012 \\
r_c &= 0.282 \text{ m}
\end{align*}
\]

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

<table>
<thead>
<tr>
<th>Radius</th>
<th>Inlet</th>
<th>Outlet</th>
</tr>
</thead>
<tbody>
<tr>
<td>( r_c )</td>
<td>0.18m</td>
<td>0.258m</td>
</tr>
<tr>
<td>( r_t )</td>
<td>0.36m</td>
<td>0.282m</td>
</tr>
</tbody>
</table>
III. COMPRESSION STAGES

The overall stagnation temperature rise through the axial compressor is $T_{02} - T_{01}$ (448.8K). The stage temperature rise ($\Delta T_{0s}$) can vary widely in different compressor designs, depending on the application and weight. Rather than choosing a value at random, it is instructive to estimate a suitable $\Delta T_{0s}$ based on the mean blade speed.[4]

The stage temperature rise ($\Delta T_{0s}$) can be calculated using equation 3.2:

$$\Delta T_{0s} = \frac{\Delta \dot{m}_w V_f (\tan \delta_1 - \tan \delta_2)}{C_p}$$

With an axial velocity at entry to the first stage and no inlet guide vane (IGV) from the velocity diagram in figure 3.1:

$$\cos \delta_1 = \frac{150}{227.41}$$

$$\delta_1 = \cos^{-1} 0.6744$$

$$\delta_2 = 47.59^\circ$$

From equation 3.2 and neglecting work factor $\lambda$:

$$\Delta T_{0s} = 28.43K$$

Temperature rise per stage is 28.43K

Number of stages = Temperature rise through the compressor / Temperature rise per stage

Number of stages = 448.8K / 28.43 = 16 stages

A. ESTIMATION OF NUMBER OF STAGES

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

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IV. STAGE DESIGN

A. STAGE ONE

From the velocity diagram in fig 3.1:

$$V_{w1} = V_f \tan \alpha_1$$  

$$V_{w2} = V_{w1} + \Delta V_w$$

For the first stage $\alpha_1 = 0$ 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 ($\lambda$) is applied to stage one:

$$\Delta V_w = \frac{C_p \Delta T_g}{\Delta \dot{m}_m}$$

$$\Delta \dot{m}_m = 1.005E3 \times 28.43 \times 0.98 \times 270$$

$$\Delta V_w = 107.98m/s$$

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

$$\cos \delta_1 = \frac{150}{227.41}$$

$$\delta_1 = \cos^{-1} 0.6744$$

$$\delta_1 = 60.95^\circ$$

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Since $V_{w1} = 0$, $V_{w2} = 107.98m/s$
\[ \delta_1 \text{= tan}^{-1} 1.080 \]
\[ \delta_2 = 47.21^\circ \]
\[ \tan \alpha_2 = \frac{v_{w2}}{V_f} \]
\[ \alpha_2 = 35.75^\circ \]
\[ \alpha_2 = 0.7199 \]
\[ \delta_2 = 35.75^\circ \]

The velocity diagram of the first stage is shown in fig. 1.2.

Deflection of the rotor blade
\[ \delta_{\text{rotor}} = \delta_1 - \delta_2 \]

Deflection of the rotor blade for stage one
\[ \delta_{r1} = 60.95^\circ - 47.21^\circ \]

de Haller number = \[ \frac{\cos 60.95^\circ}{\cos 47.21^\circ} \]
de Haller number = 0.714

Pressure ratio for stage one
\[ \left( \frac{P_{02}}{P_{01}} \right)_{n} = \left( 1 + \frac{\eta \cdot \Delta T_{0}}{T_{01}} \right)^{0.5} \]

Pressure difference in stage one
\[ \Delta P = P_{02} - P_{01} \]
\[ \Delta P_{1} = 0.346 \text{ bar} \]
Degree of Reaction
\[ \lambda = 0.80 \]

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

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

Table 4.1 The summary calculation of other stages are shown in

V. CONCLUSION

The performance of an axial flow compressors depends on its 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 - 1.2 with 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 \( \delta_1 \) at which the flow approaches the rotor.

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


