Five-Stage Axial Flow Compressor for Gas Turbine

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Abstract - The goal of this paper is to calculate the blade design of axial flow compressor. Main objective of the compressor usage is to compress the fluid and to deliver higher pressure than its original pressure. So, the stagnation pressure ratio and overall pressure ratio are calculated in this paper. Blade mean diameter is 0.295 m for all five stages. The number of blades for each stage calculated in this paper is 82 blades for first stage, 90 blades for second stage, 117 blades for third stage, 175 blades for fourth stage and 307 blades for fifth stage. In this study, inlet flow rate is 20451 CFM, outlet flow is 2811.427 CFM, compressor horsepower is 1060 hp, polytropic head is 255.649 kNm/kg and efficiency of compressor is 91.83%. And then, the general performance curves of axial flow compressor are described based on the inlet flow rate, pressure ratio and head. When these performance curves are compared with the original performance curve of axial flow compressor, the curves are nearly similar. So, the design calculation results of axial flow compressor are satisfied.

Index Terms - gas turbine, rotor, stator, blade design, axial compressor

I. INTRODUCTION

Axial compressors are compressors in which the fluid flows mainly parallel to the axis of rotation. An axial compressor comprises a number of stages as determined by the required overall pressure rise. Each stage includes a rotating row of blades (rotor) followed by a stationary row of blades or vanes (stator). The rotor blades impart momentum to the fluid thus increasing the total energy of the flow and propelling the fluid along the axis of the machine. The stator vanes convert much of the fluid momentum into pressure energy so that a rise in the static pressure across the stage occurs whilst the mean axial velocity of the flow through the stage is approximately constant. The angles of the blades and vanes relative to the direction of flow are critical to the pressure rise and operating efficiency of a stage.

II. DESIGN CALCULATION OF FIVE-STAGE AXIAL FLOW COMPRESSOR FOR GAS TURBINE

A. Design Calculation Of Blade Design For Axial Flow Compressor

An axial flow compressor used in gas turbine engine has constant axial velocity throughout the compressor of 152 m/s, blade mean velocity of 162 m/s and it is delivered 10.5 kg of air per second at a rotational speed of 10500 rpm. Reaction ratio is 50 % for each stage and the work done factor is 0.92. Stage efficiency and polytropic efficiency is 84 % and 87 %. Stagnation temperature rise for each stage is 15°K. Inlet stagnation temperature and pressure is 288°K and 1 bar. Blade aspect ratio and pitch-chord ratio are 3 and 0.5. In this research, the design data of axial flow compressor used in gas turbine engine is picked up from the Ta Dar Oo airway in Myanmar.

Axial flow compressor has the following known data;

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Constant axial flow velocity, $C_a$</td>
<td>152 m/s</td>
</tr>
<tr>
<td>Blade mean velocity, $U_m$</td>
<td>162 m/s</td>
</tr>
<tr>
<td>Mass flow rate, $m$</td>
<td>10.5 kg/s</td>
</tr>
<tr>
<td>Rotational speed, $N$</td>
<td>10500 rpm</td>
</tr>
<tr>
<td>Reaction ratio, $R$</td>
<td>50 %</td>
</tr>
<tr>
<td>Work done factor, $\lambda$</td>
<td>0.92</td>
</tr>
<tr>
<td>Inlet stagnation temperature, $T_{01}$</td>
<td>288°K</td>
</tr>
<tr>
<td>Inlet stagnation pressure, $P_{01}$</td>
<td>1 bar</td>
</tr>
<tr>
<td>Stagnation temperature rise, $\Delta T_{0s}$</td>
<td>15°K</td>
</tr>
<tr>
<td>Stage efficiency, $\eta_s$</td>
<td>84 %</td>
</tr>
<tr>
<td>Polytropic efficiency, $\eta_p$</td>
<td>87 %</td>
</tr>
<tr>
<td>Blade aspect ratio, AR</td>
<td>3</td>
</tr>
<tr>
<td>Pitch-chord ratio, s/c</td>
<td>0.5</td>
</tr>
</tbody>
</table>

Assume properties of air, $\gamma = 1$ , $C_p = 1005$ kJ/kg°K , $MW = 28.96$ , $R = 287$ J/kg°K
(i) Determination of the Blade Mean Diameter, \( U_m = \pi D_m \times N \times \frac{60}{T} \)

(ii) Calculation of Stagnation Pressure Ratio, \( R_s = \left[ 1 + \frac{n \times \Delta T_{bs}}{T_{bs}} \right]^{\gamma - 1} \)

(iii) Calculation of the Air and Blade Angles for Rotor, \( \lambda = \frac{C_s}{C_t} (\tan \beta_1 + \tan \beta_2) \)

(iv) Calculation of the Rotor Inlet Properties,

Absolute Velocity, \( \cos \alpha = \frac{C_a}{C_1} \)

Whirl Velocity, \( \sin \alpha = \frac{C_w}{C_1} \)

Relative Velocity, \( \cos \beta = \frac{C_v}{C_1} \)

(v) Calculation for First Stage

Number of blades, blade hub and blade tip diameters for first stage can be calculated by \( T_{bs} = T_{bs} + \Delta T_{bs} \).

Outlet stagnation pressure can be calculated from stagnation pressure ratio equation, \( R_s = \frac{P_{o2}}{P_{o1}} \).

Outlet static temperature, \( T_2 = T_{o2} - \frac{C_s^2}{2C_p} \)

Outlet static pressure, \( \frac{P_2}{P_{o2}} = \left[ \frac{T_2}{T_{o2}} \right]^{\gamma - 1} \)

Inlet static temperature, \( T_1 = T_{o1} - \frac{C_s^2}{2C_p} \)

Inlet static pressure, \( \frac{p_1}{p_{o1}} = \left[ \frac{T_1}{T_{o1}} \right]^{\gamma - 1} \)

Air Density, \( \rho_1 = \frac{T_1}{p_1} \)

mass, \( m = \rho_1 \times A_1 \times C_s \)

Blade Height, \( A_1 = \pi \times D_m \times h_1 \)

Number of Blades, \( z = \pi \times \frac{D_m}{s} \)

Tip and Hub Diameter, \( D_t = D_m + h_1, \ D_h = D_m - h_1 \)

(vi) Calculation for Second, Third, Fourth and Fifth Stage

Outlet stagnation temperature and pressure at the previous stage is inlet stagnation temperature and pressure for the next stage. So, stagnation temperature, stagnation pressure, static temperature, static pressure, air density, flow area, blade height, blade chord, blade thickness, blade pitch, number of blades, blade hub and blade tip diameters for the remaining stages of compressor rotor can be described by the following result table.

<table>
<thead>
<tr>
<th>Stage</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>( T_{o1} ) (°K)</td>
<td>288</td>
<td>303</td>
<td>318</td>
<td>333</td>
<td>348</td>
</tr>
<tr>
<td>( P_{o1} ) (bar)</td>
<td>1</td>
<td>1.162</td>
<td>1.569</td>
<td>2.462</td>
<td>4.489</td>
</tr>
<tr>
<td>( R_s )</td>
<td>1.162</td>
<td>1.35</td>
<td>1.569</td>
<td>1.823</td>
<td>2.118</td>
</tr>
<tr>
<td>( T_{o2} ) (°K)</td>
<td>303</td>
<td>318</td>
<td>333</td>
<td>348</td>
<td>363</td>
</tr>
<tr>
<td>( P_{o2} ) (bar)</td>
<td>1.162</td>
<td>1.569</td>
<td>2.462</td>
<td>4.489</td>
<td>9.507</td>
</tr>
<tr>
<td>( T_1 ) (°K)</td>
<td>276.05</td>
<td>291.05</td>
<td>306.05</td>
<td>321.05</td>
<td>336.05</td>
</tr>
<tr>
<td>( P_1 ) (bar)</td>
<td>0.862</td>
<td>1.009</td>
<td>1.372</td>
<td>2.166</td>
<td>3.972</td>
</tr>
<tr>
<td>( T_2 ) (°K)</td>
<td>282.91</td>
<td>297.91</td>
<td>312.91</td>
<td>327.91</td>
<td>342.91</td>
</tr>
<tr>
<td>( P_2 ) (bar)</td>
<td>0.914</td>
<td>1.248</td>
<td>1.98</td>
<td>3.645</td>
<td>7.789</td>
</tr>
<tr>
<td>( \rho_1 ) (kg/m³)</td>
<td>1.088</td>
<td>1.208</td>
<td>1.562</td>
<td>2.351</td>
<td>4.118</td>
</tr>
<tr>
<td>( A_1 ) (m²)</td>
<td>0.063</td>
<td>0.0571</td>
<td>0.044</td>
<td>0.0294</td>
<td>0.0167</td>
</tr>
<tr>
<td>( h_1 ) (m)</td>
<td>0.068</td>
<td>0.062</td>
<td>0.048</td>
<td>0.032</td>
<td>0.018</td>
</tr>
<tr>
<td>( c_1 ) (m)</td>
<td>0.020</td>
<td>0.021</td>
<td>0.016</td>
<td>0.0105</td>
<td>0.006</td>
</tr>
<tr>
<td>( t_1 ) (m)</td>
<td>0.00022</td>
<td>0.00021</td>
<td>0.00016</td>
<td>0.00011</td>
<td>0.00006</td>
</tr>
<tr>
<td>( s_1 ) (m)</td>
<td>0.011</td>
<td>0.0102</td>
<td>0.008</td>
<td>0.0052</td>
<td>0.003</td>
</tr>
<tr>
<td>( n ) (blades)</td>
<td>82</td>
<td>90</td>
<td>117</td>
<td>175</td>
<td>307</td>
</tr>
<tr>
<td>( D_t ) (m)</td>
<td>0.363</td>
<td>0.357</td>
<td>0.343</td>
<td>0.327</td>
<td>0.313</td>
</tr>
<tr>
<td>( D_h ) (m)</td>
<td>0.227</td>
<td>0.233</td>
<td>0.247</td>
<td>0.263</td>
<td>0.277</td>
</tr>
</tbody>
</table>
B. Calculation of Power of Axial Flow Compressor

(i) To calculate the compressor power,

\[
W = C_p \times \Delta T_{\text{overall}} = C_p \times (T_{02} - T_{01}) = 75.375 \text{ kJ/kg}
\]

Compressor power, 

\[
\text{Power} = m \times W = 10.5 \text{ kg/s} \times 75.375 \text{ kJ/kg} = 1060 \text{ hp}
\]

(ii) Calculation of Polytropic Head

Assume \( Z = 1 \), \( M_W = 28.96 \)

Specific gas constant is 

\[
R = \frac{8314}{M_W} = 287 \text{ J/kg°C}
\]

(iii) Static temperature at compressor outlet can be calculated by

\[
T_{2} = T_{02} \left( \frac{P_2}{P_{02}} \right)^{\gamma / (\gamma - 1)} = 363 - 200.952 \times 2 \times 1005 = 342.9 \text{°K}
\]

(iv) Outlet static pressure can be calculated from

\[
\frac{P_2}{P_{02}} = \left( \frac{T_2}{T_{02}} \right)^{\gamma / (\gamma - 1)} = 7.789 \text{ bar}
\]

(v) Pressure ratio across the compressor is

\[
r_p = \frac{P_2}{P_1} = 9.036
\]

(vi) Polytropic head can be calculated by

\[
H_p = Z \times R \times T_1 \times \left( \frac{n}{n-1} \right) \left( \frac{r_p}{n-1} \right) = 255.649 \text{ kNm/kg}
\]

C. Calculation of Inlet and Outlet Mach number

(i) Inlet Mach number is calculated by

\[
M_1 = \frac{V_1}{\sqrt{\gamma \times R \times T_1}} = 0.6
\]

(ii) Outlet Mach number is calculated by

\[
M_2 = \frac{C_2}{\sqrt{\gamma \times R \times T_2}} = 0.54
\]

(iii) Calculation of Inlet and Outlet Flow

Inlet flow , \( Q_1 = \frac{m \times Z \times R \times T_1}{P_1} = 9.65 \text{ m}^3 \text{ s}^{-1} = 9.65 \text{ m}^3 \text{ (1 m)}^3 \times \frac{1 \text{ m}}{\text{ min}} \times 60 \text{ s} = 20451 \text{ CFM}
\]

Outlet flow , \( Q_2 = \frac{m \times Z \times R \times T_2}{P_2} = 1.326 \text{ m}^3 \text{ s}^{-1} = 1.326 \text{ m}^3 \text{ (1 m)}^3 \times \frac{1 \text{ m}}{\text{ min}} \times 60 \text{ s} = 2811.427 \text{ CFM}
\]

D. Efficiency of Axial Flow Compressor

To calculate the efficiency of compressor, ideal stagnation temperature at compressor outlet can be determined by using following equation,

\[
T_{02}' = \left[ \frac{P_{02}}{P_{01}} \right]^{\gamma / (\gamma - 1)} \times T_{01} = 356.873 \text{°K}
\]

Efficiency of compressor, \( \eta_c = \frac{T_{02}' - T_{01}}{T_{02} - T_{01}} \times 100\% = 91.83\% \)

E. Classification of Compressor Types

Compressors can be classified based on the inlet flow and pressure ratio across the compressor.

Type of compressor is multi-stage axial flow compressor from the value of pressure ratio and inlet flow. Thus, the design of axial flow compressor is satisfied.

F. Performance Curves for Axial Flow Compressor

Performance curve is important for the design of compressor and this performance curve can be checked whether this design is satisfied or not satisfied. Figure 1 shows the performance curve of axial flow compressor based on the inlet flow rate, \( Q \) and pressure ratio and Figure 2 provides the performance curve using the value of inlet flow rate, \( Q \) and head. The design performance curves are nearly similar to the original performance curve of axial flow compressor. So, the design calculations of axial flow compressor are satisfied.

![Figure 1. Performance Curve for Axial Flow Compressor](http://dx.doi.org/10.29322/IJSRP.8.9.2018.p8129)

![Figure 2. Performance Curve for Axial Flow Compressor](http://dx.doi.org/10.29322/IJSRP.8.9.2018.p8129)
III. CONCLUSION

This paper is attempted to design multi-stage axial flow compressor for gas turbine engine. The design of compressor calculated in this paper is blade mean diameter of 0.295 m for all five stages. Number of blades for the first stage rotor is 82 blades, the second stage rotor is 90 blades, the third stage rotor is 117 blades, the fourth stage rotor is 175 blades and 307 blades for the fifth stage rotor. Material selection for the rotor blade is steel. Rotor air inlet angle $\alpha_1$ is 11.3° and rotor air outlet $\alpha_2$ is 40.85°. Rotor blade inlet angle $\beta_1$ is equal to rotor air outlet angle $\alpha_2$ and rotor blade outlet angle $\beta_2$ is equal to rotor air inlet angle $\alpha_1$ since the reaction ratio for this compressor is 50%. Inlet absolute velocity, $C_1$ is 155 m/s, inlet whirl velocity, $C_{w1}$ is 30.37 m/s and inlet relative velocity, $V_1$ is 200.95 m/s. Inlet relative Mach number, $M_1$ is 0.6. Outlet absolute velocity, $C_2$ is 200.95 m/s, outlet whirl velocity, $C_{w2}$ is 131.43 m/s and outlet relative velocity, $V_2$ is 155 m/s. Outlet Mach number, $M_2$ is 0.54. Inlet and outlet flow is 20451 CFM and 2811.427 CFM. Polytropic head is 255.649 kNm/kg. Compressor horsepower is 1060 hp and the efficiency of compressor is 91.83%.

APPENDIX

![Figure A.1. Section View of Axial Flow Compressor Rotor](image1)

![Figure A.2. Section View of First Stage Rotor](image2)

![Figure A.3. Section View of Second Stage Rotor](image3)

![Figure A.4. Section View of Third Stage Rotor](image4)

![Figure A.5. View of Fourth Stage Rotor](image5)

![Figure A.6. View of Fifth Stage Rotor](image6)
ACKNOWLEDGMENT

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