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Research Article

Design Methodology of a Two Stage Axial Compressor

Gaddam Srikanth [Ȧ*], S.Srinivas Prasad [Ȧ], V.Mahesh Kumar [Ȧ] and B.Mounica Reddy [Ȧ]

[Ȧ] Department of Aeronautical Engineering, MLR Institute of Technology, Hyderabad, India

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Abstract

The main objective of the paper is creating a method on how one can model an axial flow compressor. A systematic design approach has been used in the present study in order to design an axial flow compressor. The design calculations were based on thermodynamics, gas dynamic, fluid mechanics, aerodynamic and empirical relations. A two- dimensional compressible flow is assumed with constant axial and rotor blade velocities. A free-vortex swirl distribution was used in the design. These calculations include thermodynamic properties of the working fluid, number of rotor and stator blades, tip and hub diameters, blade dimensions (chord, length and space) for both rotor and stator, velocity triangles before and after the rotor and stator, mach number, solidity, flow and blade angles (blade twist). A repeated stage calculation is made to calculate the above parameters along compressor stages. The twist of the blades can be calculated along the blade length at any required number of sections selected by the designers to obtain smooth blade twist profile. NACA 65010 has been selected as a blade profile for the compressor at root, mean and tip. Here it is calculated at three main sections, the hub, the mean and the tip. The blade selected for this study is NACA 65010. The data obtained from the systematic approach has been utilised to generate three dimensional compressors.

Keywords: *Design of compressor, Gas Turbines*

1. Introduction

Axial compressor is one of the most common types of compressor. It finds its major application in large gas turbine engine like those of power today's jet aircraft, it derive its name from the fact that the air being compressed has very little motion in the radial direction, in contrast, the radial motion of the air in centrifugal compressor is much longer than the axial motion[1]. In general, the axial machines have much greater mass flow but much less pressure ratio per stage because of the boundary layer behavior, fundamentally, the axial compressor is limited by boundary layer behavior in adverse (positive) pressures gradients, each blade passage of compressor may be thought of as a diffuser, so that the boundary layer on all its wall are subject to a pressure increase unless this pressure gradient is kept under control, separation or stall will occur[2]. The compressor is made up of two major assemblies. The rotor with its blades and the casing with its stationary blades (stator) and this make one stage, the rotor increase the angular velocity of the fluid resulting increase in total temperature, total pressure and static pressure. The following stator decreases the angular velocity of the fluid, resulting in an increase in the static pressure and sets the flow up for the following rotor[3]. The axial compressor may be designed with constant tip diameter or with constant mean diameter or with constant hub diameter or with all varying; however the mean blade radius does not usually change very much. The blade length varies in order to accommodate the variation in air density so that the axial velocity component will be approximately uniform[4].

2. Design Specifications

The compressor has been designed for the operating conditions given in Table 1.

Table 1 Design Specifications

Specifications	
Type of compressor	Constant mean diameter
Mass flow rate	10 kg/s
Pressure ratio	1.75
Ambient conditions	Atmospheric conditions
Working fluid	Air
Constant axial velocity	150 m/s

3. Design Calculations

3.1 Annulus Geometry

For a constant mass flow rate, from the continuity equation,

$m = \rho A C_a$

The area of cross section for the flow can be expressed as

$$A = \pi r_t^2 \left(1 - \left(\frac{r_h}{r_t}\right)^2\right)$$

*Corresponding author: **Gaddam Srikanth**

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At sea level, $T_{01} = T_a = 288$ K
$P_{01} = P_a = 1.01325\ bar$

For the given axial velocity, the static temperature and pressure from the below formulae,

$$\Delta T = \frac{specific\ work}{C_p}$$

$$\frac{P_1}{P_{01}} = (\frac{T_1}{T_{01}})^{\frac{\gamma}{\gamma-1}}$$

Here, it is obtained T_1=276.8 K and P_1=0.881 bar
From the equation of state, $\rho = 1.109\ kg/m^3$
From the continuity equation, so at inlet
A_1=0.060 m^2.
Taking ratio of hub radius to tip radius as 0.5, as is suggestible,
$h = \frac{A_1}{2\pi r_m}$
h=0.07997 m
and $r_t = r_m + \frac{h}{2}$
$r_h = r_m - \frac{h}{2}$
r_t=0.159 m
r_h=0.0798 m
r_m=0.1194 m
At the delivery,
$P_{02} = \pi_c P_{01}$ = 1.773 bar
For a compressor with a polytrophic efficiency of 0.9;
T_{02}=344K
So, T_2=332.80K
P_2=1.579bar
$\rho_2 = 1.653\ kg/m^3$
A_2=0.0403 m^2
h=0.0537 m
r_t=0.1462 m
r_h=0.0925 m

3.2. Estimation of Number of Stages

The temperature rise through the compressor = T_{02}-T_{01}=344-288=56K
The mean blade speed U_m=2πr_m = 262.5m/s
With a pure axial velocity at the entry with no inlet guide vanes,
$tan\beta_1 = \frac{U}{C_a}$ = 1.75

$\beta_1 = 60.25^0$

And $V_1 = \frac{C_a}{\cos\beta}$ =302.28 m/s.

In most compressor stages both the rotors and the stators are designed to diffuse the fluid, and hence transform its kinetic energy into an increase in static enthalpy and static pressure. The more the fluid is decelerated, the bigger pressure rise, but boundary layer growth and wall stall is limiting the process. To avoid this, de Haller proposed that the overall deceleration ratio, i.e. V_2/V_1 and C_2/C_3 in a rotor and stator respectively, should not be less than 0.72 (historic limit) in any row.
Taking the de Haller number as 0.72, as is appropriate,
$\frac{V_2}{V_1} = 0.72$

V_2=217.64 m/s

$\beta_2 = 46.43^0$

$\Delta T_{0s} = \frac{\lambda U C_a(tan\beta_1 - tan\beta_2)}{c_p}$ = 27.9K
Therefore, the number of stages = $\frac{\Delta T_0}{\Delta T_{0s}}$=2.004 ≅ 2 stages.

If the axial velocity is set to be constant throughout the compressor, the blades at the end of the compressor will be very short and thus have higher losses and more susceptible to mechanical stresses. Setting the axial velocity ratio will take this in to consideration. As the fluid is working itself towards the end of the compressor, boundary layer growth starts to appear on the compressor housing. This will result in a narrower path for the fluid to flow through.

3.3 .Calculation of Velocity Triangles

The velocity triangles at inlet and outlet are shown in Fig 1

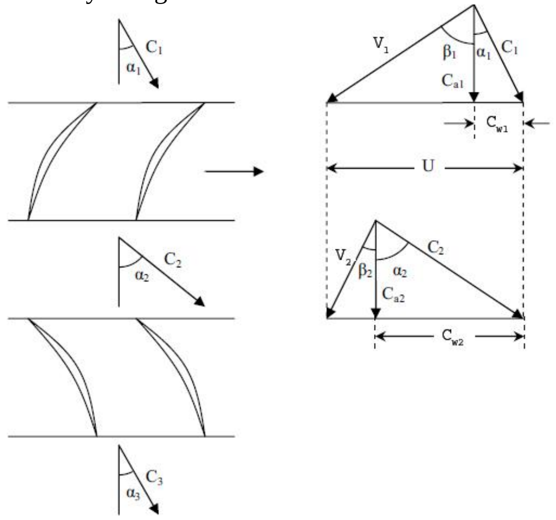

Fig 1 Velocity triangles for a single stage

For a free vortex design,
C r = constant

The air and blade angles calculated are shown in Fig 2.

Table 2 Air angles and Blade angles

Air/ Blade angles	α_h	α_m	α_t	β_h	β_m	β_t
1	0	0	0	49.4	60.2	66.7
2	46.2	36.09	29.2	7.07	45.0	60
3	17.4	12.84	10.1	55.1	56.6	57.5
4	46.1	44.71	38.5	35.3	37	43.5
5	66.2	60.46	55.2	0	0	0

3.4. Calculation of Blade Geometry

Blade nomenclature and blade angles are shown in Fig 2.

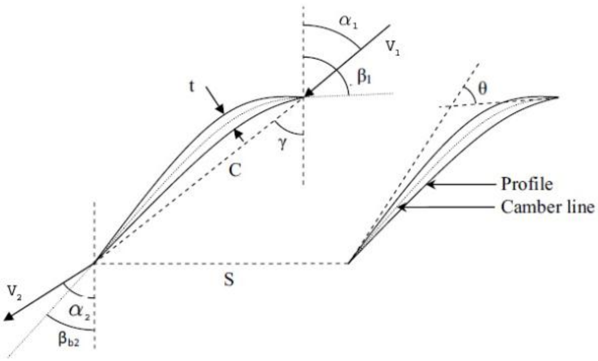

Fig 2 Blade geometry

There are many airfoils suitable for designing low pressure transonic axial compressors. Of them, NACA 65010 is selected here.

For 1st rotor:

For the mean line design,
Let s/c = 0.9 and the aspect ratio be 3,
Then, c = h/AR = 0.07997/3 = 0.02665m
Pitch, s = 0.9x0.02665 = 0.02398 m
Number of blades, n = $2\pi r_m/s$ = 31.3
Rounding to next integer as 32, the corrected pitch, chord and aspect ratio is
S = 0.02344m
C = 0.02604m
h/c = 3.07. Now, the deviation,
$\delta = m\theta\sqrt{s/c}$
And m = $0.23(2a/c)^2+0.1(\alpha_2/50)$ = 0.2360
$\delta = 0.223\theta$
$\theta = \beta_1 - \alpha_2 + \delta$ =30.93^0
So, the stagger angle $\boldsymbol{\gamma} = \beta_1 - \frac{\theta}{2}$ = 44.783^0
The above procedure is repeated at the tip and hub for all the four cascades.

4. **Results and Discussion**

The values tabulated below are the geometrical parameters of the various blades in this compressor. As the tables 3(a), 3(b) depicts, there is a variation from blade to blade regarding every geometrical aspect. As the hub diameter goes on decreasing, the pitch decreases and the number of blades increases. And the three different stagger angles play a major role in the blade twist.

Table 3 (a) Blade geometry parameters

S.No.	Cascade	Height (m)	Chord (m)	Number of blades
1	Rotor 1	0.07997	0.02665	32
2	Stator 1	0.0726	0.0242	35
3	Rotor 2	0.0653	0.0217	39
4	Stator 2	0.0595	0.0198	43

Table 3(b) Blade Pitch and Angles

S.No.	Pitch(m)	γ_h (degrees)	γ_m (degrees)	γ_t (degrees)
1	0.02398	47.407	44.785	43.480
2	0.02178	30.815	25.795	13.325
3	0.01950	49.330	48.970	45.500
4	0.01780	56.390	52.730	51.290

Three dimensional model of a two stage axial compressor has been generated based on the calculated data. Model has been generated in CATIA. Figure3 illustrates the two stage axial compressor in isometric view.

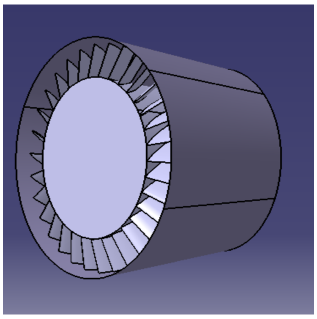

Fig 3 Two stage axial Compressor

5. Conclusion

This study demonstrated the design of a two first-stage axial- flow compressor using NACA airfoil. The case study provides the valuable information about the design methodology. The geometry created can be used as model either to carry fluid flow analysis or the structural analysis**.**

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