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Design and Analysis of Low-Pressure Axial Flow Compressor at Single Stage

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ABSTRACT: The purpose of this work is to provide a method for the design of an axial flow compressor stage. This latter represents the front stage of a multistage compressor of industrial gas turbine. The proposed blade design approach is based on the mean line concept which assumes that mean radius flow conditions prevailing at all other radial stations. The different conservation equations of fluid mechanic; mass, momentum, and energy with ideal gas state equation are applied in conjunction with the NASA empirical relations to compute both incidence and deviation angles at design condition. A excel sheet computer software was implemented with the inputs such as overall pressure ratio, design speed, tip diameter, hub-tip ratio, operating absolute Mach no, mass flow rate, tip speed, pressure ratio, ambient temperature and pressure under larger flow coefficient and high reaction ratio. Design process begins with calculation of the channel geometric form from inlet to outlet stage. Thermodynamic properties of the working fluid are determined at rotor inlet, stator outlet and the intermediate station. Detailed geometry of the cascade; chord, pitch, camber and stagger angle is identified for both rotor and stator.

KEYWORDS: Axial flow compressor, Single stage, Rotor, Stator, Annulus Dimensions, Vortex Angle, Stagger angle, Flow direction.

I. INTRODUCTION

The necessity of industrial gas turbine power plants increases with the rise demand of electricity. It represents one of the most important plants in the energetic engineering field. Generally, gas turbine incorporates three main parts: compressor, combustion chamber and turbine. These parts form the Brayton thermodynamic cycle used in either electricity production in residential and industrial areas or thrust generation in the aviation field. Compressor has the primordial role because high efficiency of the gas turbine is strongly connected to the pressure ratio delivered by the compressor. A lot of works were carried out in the field of design and analysis of axial flow compressors. However, to understand all details and ideas, from these works, leading to design the axial compressor and give the detailed geometric form of blading, hub, casing and its different parts, a tedious work will be devoted. The purpose to model compressor stage, on the basis of empirical correlations and thermodynamic relations, is to provide the detailed geometry in order to use it in the analysis process and avoiding the CFD approach which consumes a long time during the calculation in the design operation.

Axial compressors are high-speed, large-volume compressors but are smaller and somewhat more efficient than comparable centrifugal compressors. The axial compressor's capital cost is higher than that of a centrifugal but may well be justified by energy cost savings in an overall evaluation. The pressure ratio per stage is less than that of the centrifugal. In a general comparison, it takes approximately twice as many stages to perform the same pressure ratio as would be required by a centrifugal. The characteristic feature of this compressor, as its name implies, is the axial direction of the flow through the machine.

The energy from the rotor is transferred to the gas by rotating blades—typically, rows of unshrouded blades. Before and after each rotor row is a stationary (stator) row. The first stator blade row is called the guide vane. The volume range of the axial compressor starts at approximately 30,000 cfm. One of the largest sizes built is 1,000,000 cfm, though this size is certainly not common. The common upper range is 300,000 cfm.



II. RELATED WORK

The fluid enters and leaves in the axial direction, The centrifugal component in the energy equation does not come into play. Here the compression is fully based on diffusing action of the passages. The diffusing action in the stator converts the absolute kinetic head of the fluid into a rise in pressure. The relative kinetic head in the energy equation is a term that exists only because of the rotation of the rotor. The rotor reduces the relative kinetic head of the fluid and adds it to the absolute kinetic head of the fluid. Analysis using CFD or Analysis of flow through the compressor Cascade.

III. METHODOLOGY

Consider mass flow rate, Pressure ratio, Axial velocity. Calculate Annulus dimensions using continuity equation. Select appropriate adiabatic efficiency. Calculate stagnation pressure and temperature at inlet and outlet. Calculate inlet and outlet pressure and temperature for stage. Calculate radius at hub, Tip and Mid for stage. Determine variation of Air angles from root to tip. Calculate number of blades for stage. Calculate Blade height, Pitch, chord and Stagger angle. Generate blade coordinates for stator and rotor for stage. Study its compressibility effects. Check whether all requirements are satisfied.

IV. EXPERIMENTAL RESULTS

Design Specifications:

Specifications	Values
Design speed	10000
Pressure ratio	1.2 bar
Ambient conditions	Atmospheric conditions
Working fluid	Air
Constant axial velocity	170 s

TABLE 1: Inlet condition

V. CALCULATION

DETERMINATION OF ANNULUS DIMENSIONS:

The required annulus area at entry is obtained from the specified mass flow, axial velocity and ambient condition

From given Hub-tip radius we can find radius at hub as

$$r_h = r_t * 0.5$$

because the area of the compressor is varying along the length, so the tip radius and hub are also changing, so it is not convenient to calculate annulus dimensions with respect to hub and tip radius. So, we take a mean radius passing through mid of compressor area and it is given by,

$$r_m = (r_h + r_t) / 2$$

At static condition assuming no loss in intake we now calculate inlet temperature, pressure and density.



Inlet temperature:

$$T_1 = T_{01} - C_a^2 / 2C_p$$

Inlet pressure:

$$P_1 = P_{01} [T_1 / T_{01}]^{Y/Y-1}$$

Density at exit:

$$P_2 = P_2 / RT_2$$

From these inlets and the outlet parameters we can find the mass flow rate through axial flow compressor inlet and exit annulus area as,

Mass flow rate

$$M = \rho_1 \pi r_m [1 - (r_r / r_t)^2] C_a$$

Exit annulus area:

$$A_2 = m / \rho_2 * C_a$$

From exit area blade height can be found as,
Blade height:

$$H = A_2 / 2\pi r_t$$

Now as from inlet parameters we know the inlet hub tip radius, now we can find the hub and tip radius at exit of stage Radii at the exit from the last stator are then:

$$r_t = r_m + h/2$$

$$r_t = r_m - h/2$$

At this point we have established the rotational speed and the annulus dimensions at inlet and outlet, on the basis of a constant mean diameter.

Radial location	Stage inlet	Stage outlet	Units
Hub	0.1875	0.211	m
Tip	0.375	0.351	m

TABLE 2: Radii at inlet and outlet of compressor stage

Overall temperature rise = $\Delta T = T_2 - T_1 = 20.37 \text{ K}$

2. Stage design at mid radius:

At this point we have to find the rotational speed at mean radius



$$U_m = 2\pi r_m N / 60$$

From the velocity triangle we can see that

$$C_{w1} = C_a \tan \alpha_1$$

Absolute velocity at rotor inlet is

$$C_1 = \sqrt{C_a^2 + C_{w1}^2}$$

Relative velocity and its whirl component can be given as

$$V_{w1} = U - C_{w1}$$

$$V_1 = \sqrt{C_a^2 + V_{w1}^2}$$

Now to find the relative velocity at stator, it must satisfy De Haller's criterion that is

$$V_2 / V_1 < 0.72$$

From this diffusion criteria by De Hailer we can find V_2

Now the whirl velocity component at stator inlet can be obtained by

$$V_{w2} = \sqrt{V_2^2 - C_a^2}$$

$$C_{w2} = U - V_{w2}$$

Absolute velocity at stator inlet is

$$C_2 = \sqrt{C_a^2 + C_{w2}^2}$$

We obtain the absolute and relative velocity and their whirl components at rotor and stator inlet, from these data we can now find the variation of air angle at rotor and stator inlet as follow,

$$\alpha_2 = \tan^{-1}(C_{w2}/C_2)$$

$$\beta_1 = \tan^{-1}(V_{w1}/V_1)$$

$$\beta_2 = \tan^{-1}(V_{w2}/V_2)$$

VARIATION OF ANGLE AT MID RADIUS.

Quantity	Mid	Units
α_1	18	Degree
α_2	44.82	Degree
β_1	59.97	Degree
β_2	36.36	Degree

TABLE 3: Variation of air angle at midradius.



There are four vortex conditions as,

1. Free vortex condition
2. Forced vortex condition
3. Exponential condition
4. Constant reaction

These conditions are discussed earlier. For our design calculations we will use free vortex condition

$$C_{w1}r_m = C_{w1}r_t = \text{Constant}$$

From above equation whirl velocity at hub and tip can be obtained, and we have the blade rotational speed at hub and tip, so by repeating above procedure we will get variation of air angles at hub and tip.

STAGE DESIGN PARAMETER RESULT

Quantity	Hub	Mid	Tip	Units
α_1	18	18	18	Degree
α_2	44.85	44.82	44.45	Degree
β_1	52.41	59.97	65.16	Degree
β_2	16.91	36.36	49.71	Degree

TABLE 4: Stage design parameters results.

Mach no. at tip:

$$M_1 = V_1 / \sqrt{\gamma R T_1}$$

We get 1.22 Mach no. at tip that is the flow at rotor tip is transonic.

Blade section:

Analysis of the values of the nominal deflection ξ determined from a large number of tests covering different forms of cascade has shown that its value is mainly dependent on the pitch/chord ratio and the air outlet angle α_2 .

On this basis the whole set of test results is reducible to the form shown in fig in which the normal deflection is plotted against air outlet angle which pitch/chord ratio as parameter. This set of master curves, as they may well be called, is of great value to the designer

Because, having fixed any two of the three variation involved, an appropriate value for the third may be determined. For instance, if the air inlet and outlet angles have been fixed by the air angle design, a suitable pitch/chord ratio can be read from the diagram.

For our project consider the design of rotor blade required for the free vortex carried out in previous section. At the mean radius of 0.281 m where $\beta_1 = 59.97$ and $\beta_2 = 36.36$, $\epsilon = 35.32$ and from figure with air outlet angle of 43.83° , $s/c = 0.5$

$$S = 0.5 * 0.625 = 0.03125 \text{ m}$$

Number of blades:

$$n = 2\pi r_m / s$$



$$n = 58$$

Now for design of blade air inlet and outlet angles has to be determine. Air inlet angle α_1 is equal to blade outlet angle α_2 . But air outlet angle is blade outlet angle α_2 subtracted by deviation δ . The deviation angle is given as

$$\delta = m\theta\sqrt{s/c}$$

where,

$$m = 0.23(2a/c)^2 + 0.1(\alpha_2/50)$$

a is distance of maximum chamber from leading edge of the blade. For our blade we used circular arc chamber line blade so the $(2a/c) = 1$.

On the assumptions of a circular arc chamber-line, the deviation will be,

$$\begin{aligned} \delta &= (0.23 + 0.1(\alpha_2/50)) * \theta * \sqrt{s/c} \\ &= 0.214\theta. \end{aligned}$$

Now,

$$\theta = \alpha_1 - \alpha_2$$

$$\theta = \alpha_1 - \alpha_2 - \delta$$

we get,

$$\theta = -34.64$$

stagger angle that is blade setting angle is given as

$$\begin{aligned} \xi &= \alpha_1 - \theta/2 \\ &= 35.32 \end{aligned}$$

Similarly, we can find stagger angle for both rotor and stator at hub, mid and tip section.

STAGGER ANGLE FOR ROTOR AND STATOR

For rotor:

Stagger angle	Hub	Mid	Tip
ξ	35.4	35.32	31.33

TABLE 5: Stage angle for rotor

For stator

Stagger angle	Hub	Mid	Tip
ξ	46.72	40.887	49.65

TABLE 6: Stager angle for stator

VI. CONCLUSION

From this we can conclude that we are done with the introduction, working, advantages and calculated the values of required parameters like temperature, pressure and blade angles, blade heights, stagger angle etc. And we have also gone through the mechanical features and inlet parameters and dimensions to find out the required parameters. from this we will continue the design and analysis through the following information we got now.



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