Aerodynamic design of Axial Flow Compressor

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***Abstract*— axial flow compressor is used to get the compressed pressurized air as an input the gas turbine. Generally it is used in the aircraft engine and the industrial application. Compressor is composed with the rotor and stator blades at each stage. Aerodynamic shapes of the blades play important role to improve the efficiency of the compressor.**

Keywords: compressor, axial flow, stator, rotor.

I. INTRODUCTION

The development of gas turbines has in the recent years come a long way. Mainly development began during the Second World War with the key interest of shaft power, but attention was shortly transferred to the turbojet engine for aircraft propulsion. The gas turbine began to compete successfully in other fields in the mid-1950s, since then it has made a successful impact in an increasing variety of applications. When combining a gas turbine with a heat recovery steam generator the heat, that otherwise would be wasted from the gas turbine outlet, can be extracted. Together with a conventional steam generator this will form a combined cycle. The efficiency of a combined cycle power plant is far better than regular gas turbine power plants. The question is than, how could we improve the efficiency of a gas turbine? One can either focus on the compressor, the combustion chamber or the turbine. In this thesis the compressor, especially the axial flow compressor, will be investigated.[1]

When designing a new compressor, a good start is to create a base design for the compressor. By just a handful of design specifications an accurate model can be generated. The modeling techniques used are based on combinations of thermodynamic and aerodynamic correlations. This base design will make up for about 60-70 % of the finished design. In this first stage in designing a new compressor, designs that would not work or have pore efficiency can be avoided. Further on in the process powerful CFD (Computational Fluid Dynamics) simulation programs are being used.

The principal type of compressor being used in aircraft gas-turbine power plants is the axial-flow compressor [figure 1.1]. Although some of the early turbojet engines incorporated the centrifugal compressor, the recent trend, particularly for high speed and long-range applications, has been to the axial-flow type. This dominance is a result of the ability of the axial-flow compressor to satisfy the basic requirements of the aircraft gas turbine. These basic requirements of compressors for aircraft gas-turbine application are well-known. In general, they include high efficiency, high airflow capacity per unit frontal area, and high pressure ratio per stage. Because of the demand for rapid engine acceleration and for operation over a wide range of flight conditions, this high level of aerodynamic performance must be maintained over a wide range of speeds and flows. Physically, the compressor should have a minimum length and weight. The mechanical design should be simple, so as to reduce manufacturing time and cost. The resulting structure should be mechanically rugged and reliable.

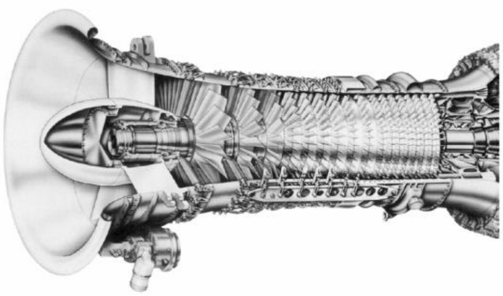

Figure 1.1: Axial Flow Compressor

Because axial-flow compressors are most extensively used in the field of aircraft propulsion, and because this field requires the highest degree of excellence in compressor design and performance, the attention in this over-all report has been focused primarily on the problems pertinent to the axial-flow compressor of turbojet or turbo-prop engines. The results, presented, however, should be applicable to any class of axial-flow compressors.[2]

II. AXIAL FLOW COMPRESSOR OPERATIONS

A typical axial flow compressor consists of a series of stages; each stage has a row of moving rotor blades followed by a row of stator blades which is stationary. The rotor blades accelerates the working fluid thus gaining energy, this kinetic energy is then converted into static pressure by decelerating the fluid in the stator blades. The process is then repeated as many times as necessary to get the required pressure ratio. The number of stages in a compressor is important especially when the engine will be used in an aircraft. The main reason is that too many stages will result in an increase in weight and a large core engine length. For land based gas turbines the main reason is the cost, which will increase when adding more stages. Some different compressors used in aircrafts are shown in Table 2.1, and here one can see how compressor improvement has come along over the years.

Table2.1: Compressor Evolution in Aircraft Engine [3]

Engine	Date	Thrust (KN)	Pressure Ratio	Stages
Avon	1958	44	10	17
Spey	1963	56	21	17
RB211	1972	225	29	14
Trent	1995	356	41	15

As discussed earlier all the power is absorbed in the rotor and the stator transforms the kinetic energy which has been absorbed by the rotor into an increase in static pressure. The stagnation temperature remains constant throughout the stator since there is no work feed into the fluid. Figure 1.2 shows a sketch of a typical compressor stage. The stagnation pressure rise occurs wholly in the rotor, but in practice, there will be some losses in the stator due to fluid friction which will result in a decrease in stagnation pressure. There are also some losses in the rotor and the stagnation pressure rise will be less than of an isentropic compression.

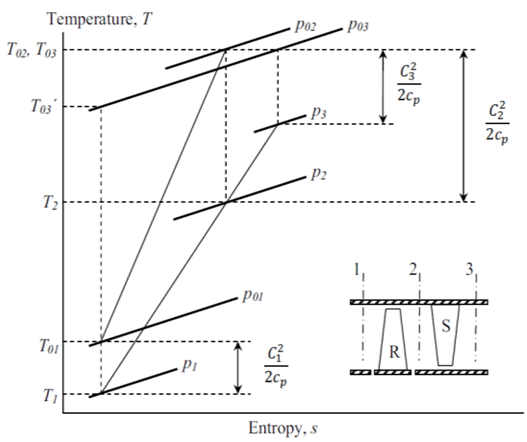

Figure 1.2: Compressor Stage and T-S diagram

In an axial flow compressor, air passes from one stage to the next, each stage raising the pressure slightly. By producing low pressure increases on the order of 1.1:1 to 1.4:1, very high efficiencies can be obtained as seen in table 2.2. The use of multiple stages permits overall pressure increases of up to 40:1 in some aerospace applications and a pressure ratio of 30:1 in some Industrial applications. [4]

Table 2.2: Axial Flow Compressor Characteristics

Type of application	Type of Flow	Inlet Mach number	Pressure ratio per stage	Efficiency per sage
Industrial	Subsonic	0.4-0.8	1.05-1.2	88%-92%
Aerospace	Transonic	0.7-1.1	1.15-1.6	80%-85%
Research	Supersonic	1.05-2.5	1.8-2.2	75%-85%

III. FLOW DIRECTION IN COMPRESSOR

Flow direction analysis in compressor is most fundamental part to know how compressor works. The velocity components of the working fluid can be expressed in two velocity vectors, absolute and relative velocity.

The fluid enters the rotor with an absolute velocity, $C1$, and has an angle, $\alpha 1$, from the axial direction. Combining the absolute velocity with the blade speed U, gives the relative velocity, $W1$, with its angle $\beta 1$. The mechanical energy from the rotating rotors will be transferred to the working fluid. This energy absorption will increase the absolute velocity of the fluid. After leaving the rotor the fluid will have a relative velocity, $W2$, with an angle, $\beta 2$, determined by the blade outlet angle. The fluid leaving the rotor is consequently the air entering the stator where a similar change in velocity will occur. Here the relative velocity, $W2$, will be diffused and leaving the stator with a velocity, $C3$, at an angle, $\alpha 3$. Typically the velocity leaving the stator will be the same as the velocity entering the rotor in the next row, $C3 = C1$ and $\alpha 3 = \alpha 1$. (Figure1.3) show the velocity triangles for one stage.

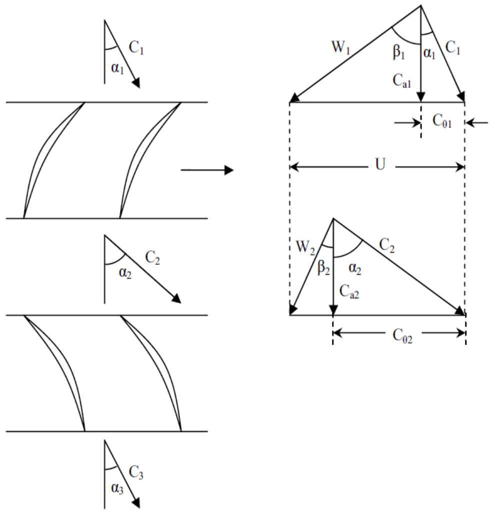

Figure 1.3: Velocity triangle for one stage

IV. DESIGN ALGORITHM

Several methods developed for the design for the design analysis of axial flow compressor

- **The mean-line Method**
  In this method the different flow properties as temperature, pressure, velocity and the dimensions of the equipment are determined at half blade height. The fact of just using one stream line makes necessary the use of empiric correlation for loss correlation originating from experimental data.
- **The Radial Equilibrium Method**
  In this method the analysis of the compressor is based on the meridian plane that contains the axis of rotation of the equipment. Here this plane is divided in control surfaces from hub to tip if the blades. The flow properties are determined using conservation laws and empirical correlations. This method allows obtaining the geometry of the blades in several points from hub to tip of each row according to the division done in the Meridional plane.
- **The Streamline Curvature Method**
  In this method the calculation is based on the resolution of the Euler's equation aided by empirical correlation of experimental data. Many streamline are adopted from hub to tip of blade and divided in sections applied in the inlet and outlet each row so all flow properties can determined at each point of the intersection among the sections of the cascade and streamlines forming control surfaces.
- **The Finite Differences Method**
  In This Method the flows as well as its properties are resolved through the finite different equations. It is necessary to define the mesh type to be used since it determines the detailing level of the solution.
- **The Finite Element Method**
  In this method the domain is divided into a group of finite elements and the properties are calculated in the resulting nodal point. This method used for non-structured grids.
- **The Finite Volume Method**
  In this method flow passes channel divided into several elementary volumes. The equation is integrated in each control volume and solved all equation by mathematical algorithms until the results are obtained as in other methods. [4]

V. STAGE LOSSES

Figure 1.4; show the energy flow diagram for an axial flow compressor stage. Figures in the brackets indicate the order of energy loss corresponding to 100 units of energy supplied at the shaft.

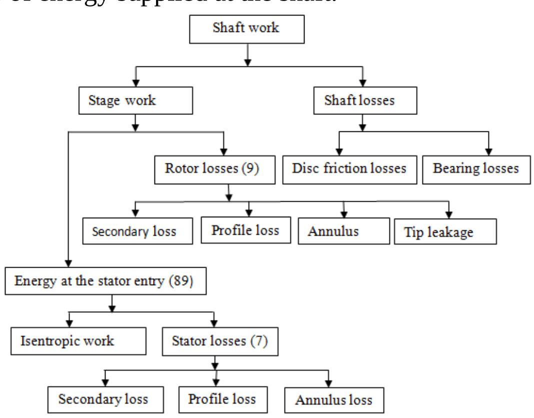

Figure 1.4: Energy Flow Diagram for an Axial Flow Compressor Stage

VI. BLADE TERMINOLOGY

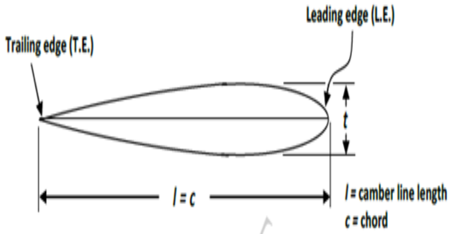

Figure 1.5: Simple aerofoil blade profile [5]

- Base profile: It is defined by dividing the major axis into equally spaced station designated as a percentage of the blade length and specifying from axis to profile at each station.
- Camber line: A blade section of Infinitesimal thickness is a curved line known as 'camber line'. This line the backbone line of a blade of finite thickness. Its shape is specified by x-y coordinates. The shapes of profiles of the upper and lower surfaces of the blade are specified by the thickness distribution along the chord or the camber line.
- Maximum thickness: It is expressed as a percentage of the blade length.
- Leading edge: It is usually a circular arc blended into the main profile and specified by its radius as a percentage of the maximum thickness.
- Trailing edge: It is ideally sharp of zero radius but as this is impossible from strength considerations, it also a circular arc specified as a percentage of the maximum thickness.[6]

Lingen Chen et al.[7], 2004, presented a model for the optimum design of a compressor stage with assuming fixed axial velocities distribution. The absolute inlet and exit angles of the rotor are taken as design variables. The optimum relation between the isentropic efficiency and the flow coefficient, the work coefficient, the flow angles and the degree of reaction of the compressor stage has been obtained using one-dimensional flow-theory .they provided a numerical example are provided to illustrate the effects of various parameters on the optimal performance of the compressor stage. The calculation of the stage performance is performed using one-dimensional flow-theory. The analysis begins from the energy equation and continuity equation. The other assumptions take as the working fluid is compressible, non-viscous and adiabatic and it flows stably relative to stator and rotor, which rotates at a fixed speed. The mass-flow rate of the working fluid is also constant.

Jun Luo et al. [8], 2007, in this paper present a model for the optimal design of a multi-stage compressor, with assuming a fixed configuration of the flow-path. The absolute inlet and exit angles of the rotor, the absolute exit angle of the stator, relative gas densities at the inlet and exit stations of the stator, of every stage, are taken as the design variables. They obtain analytical relations of the compressor elemental stage and the multi-stage compressor .for better understanding one numerical example are also provided to illustrate the effects of various parameters on the optimal performance of the multi-stage compressor. In this paper first analysis begins with the energy and continuity equations. The axial flow velocities of working fluids are not constant. First select the original value of rotor and stator flow coefficient and then calculate the stage parameters. Further by iterative method calculate value of flow coefficient and repeat the first step until the differences between the calculated values and the original ones are small enough.

Akin Keskin et al. [9], 2006, in this paper shows how to automate a given Rolls-Royce preliminary design process in order to find Pareto-optimal trade-off solutions for design conditions. The aspect of process acceleration is also an important goal to release the design engineer from time-consuming parameter studies. Essential elements for speeding up the design process are the use of modern process integration tools, multi-criterion decision concepts, and nonlinear programming algorithms. Results will be shown based on a given Rolls-Royce compressor design for multi-objective optimization with respect to maximum efficiency, maximum surge margin, and maximum overall pressure ratio, where different deterministic and stochastic algorithms are used.

Lingen Chen et al. [10], 2004, in this paper optimised design of a subsonic axial flow compressor stage with the objective of minimizing the aerodynamic losses and the weight of the stage and maximizing the compressor's stall margin. A design optimization program for a subsonic axial-flow compressor stage has been developed by applying the numerical optimization techniques to a simulation algorithm which consists of thermodynamic compression relations, cascade geometric variables, empirical loss-correlations, and simple stress relations.

J. R. Aguinaga,[11], 2011, in this paper show the method of find axial space between rotating blade rows and the stationary one in an axial-flow compressor. Here axial distance is determined by the mathematical model obtained using design variables of rotating blade rows and the stationary blade rows or determined axial distance values. the analysis also consider the transition plane between rotating blade row and the stationary one, where outlet values for the rotating blade row is considering the same for the inlet values in the stationary blade row. The results were obtained using the mathematical model and compared with data of axial flow compressor in one plane. After comparison they were founded approximations of 19% for stage number four and 16% for stage number nine of axial distance. from this method for any axial turbo machinery possible to obtain the axial distance of different stages.

B.T. Lebele-Alawa [12], 2008, in this paper presented the relationship between changes in the incident rotor-blade angle due to compressor blade profile distortions and the required compressor power. In the paper the parameters relevant for the study at the inlet and outlet of the compressor, as well as the inlet and outlet of turbine were measured at the Ugheli Power Station, Delta State, Nigeria. Performance data were obtained from the pertinent daily log-sheets for the turbine. Design parameters were taken from the design manuals. Theoretical predictions from computer simulations have been compared with the corresponding measurements. Some data shown as in table: 2.3.

Table 2.3: Measured variation of the compressor power with change in rotor blade angle

m(kg/s)	($\beta1^{o}$)	($\beta2^{0}$)	($\beta2^{0}$- $\beta1^{o}$)	W(kw)
122.1	48	21.6	26.4	12470
121.8	48.2	20.1	28.4	13304
121.3	49	18.6	30.4	14107
120.9	49.5	17.1	32.4	14913
120.5	50	15.6	34.4	15712

The graph plotted experimental and theoretical power required to rotor's inlet blade angle as shown below.

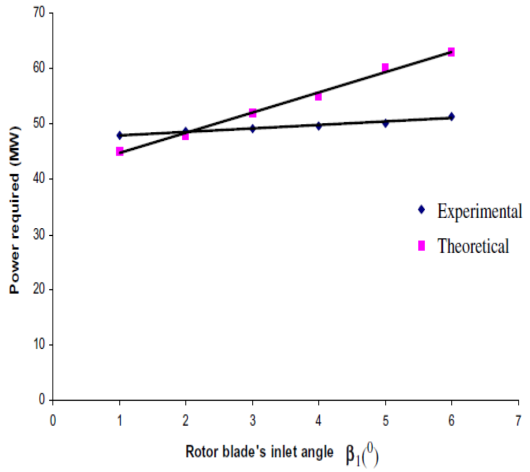

Figure 1.6: Dependence of compressor power required upon the rotor-blade inlet angle.

The graph plotted experimental and theoretical power required to rotor's outlet blade angle as shown below

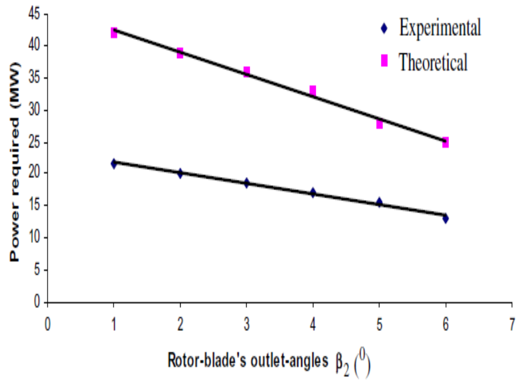

Figure 1.7: Dependence of compressor power required upon the rotor-blade outlet angle.

From the graph it conclude that Compressor blade profile distortions can result in significant increases in the compressor power required per stage as well as the decreases in the gas turbine's isentropic efficiency. Compressor blade deteriorations and distortions lead to significant reduction in the gas-turbine's performance.

I.A. Hamakhan et al [13] .in this paper presented the aerodynamic performance effects of leading-edge geometry in gas-turbine blades. They were used the Hodson–Dominy blade as an example to show the ability of this blade-design method to remove leading-edge separation bubbles in gas turbine blades and other airfoil shapes that have very sharp changes in curvature near the leading edge. This gas turbine blade example has inlet flow angle 0°, outlet flow angle -64.3°, and tangential lift coefficient 1.045, in a region of parameters where the leading edge. This gas turbine blade example has inlet flow angle 0°, outlet flow angle -64.3°, and tangential lift coefficient 1.045, in a region of parameters where the leading edge shape is critical for the overall blade performance. Computed results at incidences of -10°, -5°, +5°, and +10° are used to illustrate the complete removal of leading edge flow-disturbance regions, thus minimizing the possibility of leading edge separation bubbles, while concurrently minimizing the stagnation pressure drop from inlet to outlet.

Vyas P.B et al[14]. In this paper show the CFD analysis of the multistage axial flow compressor. In this analysis focus on this blade's geometry by changing various standard Profiles of NACA and RAF (royal air force) to get the maximum output in terms to get the higher pressure ratio and number of stages. Here first assumed some basic data, compressor pressure ratio, Air mass flow rate and compressor inlet temperature and then find root, tip and mean radius and made mathematical model. After make model and do CFD analysis. Final the result founded by theoretical model and CFD analysis is 10 % difference. From this paper conclude that by changing the various blade geometry or say by changing the aerodynamic shape of the blade we can improve the pressure ratio of the compressor and reduce the number of stages which is huge beneficial. It may be possible to increase the efficiency of the compressor by changing the various blade profiles.

X.C. Zhu et al. [15].2013, in this paper present the off-design performance of axial Compressor based on a 2d approach. In this paper used streamline curvature method based on the deviation and loss models and two compressors are simulated in detail at both design and off-design conditions and find that the pressure ratio trends for the two compressors follow the overall shape of the equivalent mass flow speed lines. Adiabatic efficiency versus speed lines are in a good qualitative and agreement as well, although the overall shape of the lines does not exactly match the experimental data.

VII. CONCLUSION

In this paper an axial flow compressor design process has been studied using different method. Also the efficiency optimization of single stage and multistage axial-flow compressor has been studied using different flow-theory. Different universal characteristic relation for an axial-flow compressor stage is obtained. Some basic knowledge about axial flow compressor design also given.

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