Preliminary Design Optimization of a Gas Turbine Multi-Stage Compressor

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ABSTRACT
In the present study, the preliminary design optimization of an axial-compressor is considered. The axisymmetric flow solver is developed to simulate the flow field within the compressor to predict quickly some important drawbacks of the conceptual design and treat them in preliminary design phase where the conceptual design is made by the mean-line method and the free-vortex assumption is utilized to find the radial distribution of the flow deflection angles. The finite volume scheme is used in this numerical procedure of the inviscid flow simulation where the advection upstream splitting method is used to calculate the fluxes.

The focus is on the axial velocity changes along the compressor, and the optimization target of the preliminary design is to increase the minimum axial velocity with the criteria of keeping the mass flow rate and the total pressure ratio constant. The results demonstrate that this target is achieved by minor modification in flow deflection angles to improve the variation of the axial velocity, which can be more important especially in off-design performance of the compressor.

Keywords
Axial compressor; Axisymmetric; Multi-stage; Mean-line; Optimization; Preliminary design.

Nomenclature

| Symbol | Definition                  |
|--------|----------------------------|
| \(c_v\) | constant volume specific heat          |
| \(E\)  | total specific energy            |
| \(H\)  | total specific enthalpy          |
| \(p\)  | local pressure                  |
| \(T\)  | local temperature               |
| \(u\)  | axial velocity                  |
| \(v\)  | radial velocity                 |
| \(w\)  | circumferential velocity        |
| \(\beta\) | relative flow angle             |
| \(\Omega\) | rotor angular velocity         |
| \(\rho\) | local density                   |

Introduction
The multi-stage axial-compressor is a low aspect ratio turbomachinery which operates at high air mass flow rate to pressurize the air stream. It is an essential component of a gas turbine which is used in power plants and transmission pipelines. A compressor design is a multi-objective problem involving different targets such as maximization of total pressure ratio, mass flow rate and efficiency, and minimization of the weight. Like any other system design process, these objectives are under consideration in all conceptual, preliminary, and detail design phases. In addition, it is a main target of the gas turbine design to improve the off-design performance which is directly affected by the compressor performance map. Although the optimizations are often made in the final steps of design process, considering the flow field’s behavior and some important criteria at the conceptual or preliminary design phase may be helpful to achieve an acceptable result with lower time and cost of the detail design process. The compressor design starts generally by the mean-line design and proceeds by through-flow procedures like the streamline-curve method and finishes by three-dimensional flow simulations and optimizations. Lots about these procedures are published in the literature which some of them are reviewed here.

Lim and Chung [1] utilized the mean-line method to maximize the efficiency and minimize the weight of the compressor. They estimated the total pressure loss by integrating the empirical coefficients of loss mechanisms along the blade. Reddy et al. [2] developed a tool for maximizing the isentropic efficiency of a multi-stage compressor. They utilized the streamline-curve through-flow technique beside the optimization procedure. Egorov and Kretinin [3] presented a method of the compromise region formulation for the multi-stage axial flow compressor optimization problems using two-dimensional axisymmetric model. Lee and Kim [4] combined an optimization technique with a three-dimensional thin-layer Navier-Stokes solver to optimize the blade shape of the axial-compressor. Dong et al. [5] utilized the code to design a new multi-stage compressor for a modern turbofan engine. They started with a mean-line model, and then used the streamline-curve flow solver. Buche and Stoll [6] developed an automated optimization process for gas turbine compressor blades using blade-to-blade flow solver. Benini [7] developed a three-dimensional optimization method using the CFX code to maximize the efficiency and pressure ratio with a constraint on the mass flow rate of the transonic compressor. Chen et al. [8] formulated the design of axial compressor as a mathematical program with the objective minimizing the aerodynamic losses.

Taghavi and Afzali [9] presented a numerical technique for preliminary design optimization of the multi-stage axial-compressor using the mean-line method. Tournier and Genk [10] developed a design model for axial flow compressor based on a mean-line analysis for free vortex flow. Wang and He [11] applied the adjoint method in design optimization for turbomachinery.
Ikeguchi et al. [13] designed a 14-stage new compressor for circumferential casing grooves to increase the overall efficiency. Kim et al. [12] performed a multi-objective optimization technique beside blading aerodynamics using three-dimensional flow solver. Kim et al. [15] optimized a transonic axial-compressor to improve operating stability using numerical solution of three-dimensional RANS equations.

Luo and Liu [16] introduced a new approach for multi-objective optimization to overcome the drawbacks of traditional gradient-based optimization techniques, and improved the aerodynamic performance of the aircraft engine by increasing the total pressure ratio and efficiency of the compressor. Yang et al. [17] designed a 4-stage compressor by a conventional one-dimensional method and using empirical correlations and then redesigned the last stage to reduce the flow losses at the operation condition near stall through modifying the aerodynamic shape and stagger angle of the stator blade. Lu et al. [18] employed an integrated optimization design system to optimized a 5-stage industrial compressor and improve the adiabatic efficiency maintaining the total pressure ratio. They used the commercial flow solver to simulate the three-dimensional flow field in optimization process. Li at al. [19] optimized the stator vane settings of highly loaded 5-stage axial flow compressor via combination of artificial neural network and genetic algorithm. They used the mean-line computation method as a flow solver. Luo [20] presented a hybrid model method based on proper orthogonal decomposition for aerodynamic optimization of the axial-compressors where the three-dimensional flow solver was used.

With the advances in computing hardware and computational fluid dynamics, the detailed design and optimization of the multi-stage axial-compressors are performed by three-dimensional flow solvers and multi-objective optimization techniques. Therefore, the detailed designs are very time-consuming, on one hand, and the outcomes of preliminary design may be so different from the outcomes of detailed design which has some side effects in design process, on the other hand. Hence, using fast techniques in preliminary design process considering some minor optimizations based on important physical behavior of the flow field in the axial-compressors may help to decrease the time of the detailed design process. Two-dimensional axisymmetric flow solvers are appropriate tools for this purpose. The common methods to perform such studies in the field of the compressor design are based on the streamline-curvature scheme which has some differences in numerical procedure and domain decomposition with the common CFD programs. In the present study, the axisymmetric flow solver is developed based on the common techniques for simulation of the two-dimensional flow fields to study the axial flow compressor and optimize the outcomes of a mean-line design by consideration of the span-wise distribution of the flow parameters to have better results in preliminary design phase.

**Governing Equations and Numerical Procedure**

In the present study, the compressible inviscid flow is considered and the three-dimensional axisymmetric equations for continuity, momentum, and energy are used to develop a numerical program for preliminary design optimization of the multi-stage axial-compressor. Although no change in θ direction is assumed, the circumferential momentum equation must be solved. The five governing equations are presented here in conservation form:

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho u}{\partial x} + \frac{\partial \rho w}{\partial y} + \frac{\partial \rho v}{\partial z} = S + S' \quad (1)$$

Where,

$$Q = \begin{bmatrix} \rho \\ \rho u \\ \rho v \\ \rho w \\ \rho E \end{bmatrix}, \quad R = \begin{bmatrix} \rho u \\ \rho u^2 + \rho + \rho w^2 \\ \rho u v + \rho w \\ \rho v^2 + \rho w \\ \rho E \end{bmatrix}, \quad S = \begin{bmatrix} \rho v \\ \rho v u \\ \rho v w + \rho + \rho v v \\ \rho w \\ \rho w v \end{bmatrix}$$

$$0 \quad 0 \quad f_c$$

$$S' = -p - \rho w v$$

$$G = 0$$

$$f_r$$

$$r\Omega$$

In which:

$$E = C_r T + \frac{u w + v v + w w}{2}$$

$$H = E + \frac{p}{\rho}$$

$$\frac{\partial f}{\partial \tau} = -\rho \sin \beta \left( \rho w - \rho v \Omega \cos \beta \right) \quad (4)$$

Where Ω is an angular velocity of the rotor. First, the axial-compressor is designed by the one-dimensional mean-line method and the radial distribution of the flow deflection angle is computed by free-vortex assumption. Then, the resulting geometry is used in the flow solver to analyze the flow field more accurately and optimize the design. The mentioned governing equations are discretized by the finite-volume and cell-centered method and the fluxes are calculated by the Advection Upstream Splitting Method (AUSM+) [22]. The developed computer program has been validated using experimental and numerical data of different flows which are presented in Ref. [23-26], and the comparison of results with the mean-line solution for the axial-compressor is presented in the next section.

**Results and Discussion**

A preliminary design optimization of the 16-stage axial-compressor is under consideration in the present study. The target mass flow rate is 300 kg/s, the target pressure ratio is 12, and the compressor is working with an angular velocity of 3000 rpm. First, the compressor is designed in mean-line, and the span-wise variation is computed by the free-vortex flow assumption. The first row tip Mach number is considered during the mean-line design procedure and the inlet hub to tip ratio is 0.7. The axial velocity is kept constant and all rows' degree of reaction is 0.5 at the mean-line. The total pressure ratio and total temperature variation in different rows of the compressor is shown in Fig. 1 for the mean-line design method. This is the outcome of the conceptual design phase, and is an ideal design that is the target of the final design. Each stage has 4 locations, rotor inlet and outlet, and stator inlet and outlet. The compressor geometry is shown in Fig. 2. In this ideal design, there is no radial velocity, and the axial velocity is constant and is uniform as well as total properties, but the real flow features are completely different. The flow field is three dimensional and the effect of boundary layer and turbulence is important and the final design or optimization must consider these features, but in the preliminary design, using the axisymmetric simulation may illustrate some drawbacks of the conceptual design.
Using developed flow solver to simulate the axisymmetric flow within the axial-compressor demonstrates the radial variation of the flow properties more accurately, and helps the designer to make some optimization in preliminary design phase. The considered 16-sage axial-compressor which is designed in the previous phase by the mean-line method is studied here. Assuming too much length for the compressor, the results are compared with the mean-line data. Their differences are expected to be small because assuming the high length for the compressor decreases the radial variations and brings the results of the axisymmetric flow solver towards the results of mean-line design. This is illustrated in Fig. 3 where also confirms the validity of both schemes. The radial non-uniformity of an axial velocity is less than five percent in simulation results.

Now, the length of the compressor is assumed 3 m based on its inlet diameter to study the flow field better in preliminary design phase. The grid resolution study shows that the 1300*50 grid points are proper for the numerical simulations as shown in figures 4, 5. Although the free-vortex assumption is made in conceptual design, the real flow field is different and the radial distribution of flow properties may be considerable in some rows. One of the most important things that should be investigated in axial-compressor design and optimization is the axial velocity profile. The axial velocity reduction especially at hub or shroud increases the probability of separation and so decreases the surge margin of the compressor. Observing such behavior in design point of the compressor which is caused by radial variations in the flow field demonstrates that it may be amplified in off-design points. Therefore it should be considered in preliminary design phase and controlled by proper optimization methods.

Fig. 1: The total pressure ratio and total temperature variation for mean-line design.

Fig. 2: Compressor annulus geometry from mean-line design.

Fig. 3: Pressure ratio and axial velocity change along the compressor from mean-line design and axisymmetric simulation.

Fig. 4: Axial velocity profile at compressor exit for different grids.
Fig. 5: Total pressure profile at compressor exit for different grids.

It is mentioned before that the optimization process in detail-design phase is performed by three-dimensional simulations which are very time-consuming, so using fast schemes in preliminary design may help to find better solutions in this phase. For the considered compressor, the axial velocity is compared in Fig. 6 with one for an unreal long compressor. The difference is due to the rate of an annulus area change which illustrates intense axial velocity reduction at the hub of some rows in design point, which is shown in Fig. 7 too. Radial velocity change is also shown in Fig 8. The axial velocity reduction is about 20 percent in some rows which may be worse in off-design operations too. It is treated in the present study by the developed fast axisymmetric flow solver to get better preliminary design. The goal is to push the behavior of the flow field from the bright points on the figures 6, 8 towards the dark points.

Suppose that the annulus geometry is fixed, the designer can adjust the flow angles to reduce the axial velocity reduction in the flow field that is caused by radial variations. Indeed, the radial distribution of the deflection angle of flow in different rows must be changed somehow that the axial velocity decrement at the hub of this compressor is improved. But the important thing here is that changing the flow deflection angles may changes the compressor’s pressure ratio and mass flow rate. So this is an optimization problem with some criteria; improving the axial velocity profile keeping annulus geometry, mass flow and the pressure ratio constant. The optimization scheme is not the focus of this study, and the results show that proper change of flow deflection angles which have been computed by the mean-line method is effective and increases the minimum axial velocity in the flow field without significant impact on the mass flow rate and pressure ratio. This is finally achieved here by applying two degrees change on the deflection angles of all rows of all stages except the first one, where it doesn’t change the tip flow deflection angle but has maximum change at hub and is a second order function of the blade radius (as illustrated in Fig. 9). The result is shown in Fig. 10.

The results demonstrate that the minimum axial velocity in the flow field is increased about 10 percent (14 m/s) during preliminary design optimization using the axisymmetric flow solver, and can be used in the blading process and optimizations in detail design phase as a better base-design. The developed fast axisymmetric flow solver allows for more accurate study of the flow field especially for preliminary design optimization of the axial-compressor.
which can be used then in the blading process and optimizations in velocity about 10 percent in preliminary design optimization, order function of the blade radius, increases the minimum axial the deflection angles of all stages except the first one, as a second deflection angles is effective and applying two degrees change on compressor. The results show that the minor modification in flow increasing the minimum axial velocity of the flow field without using the flow solver before the blading process. The target is time-consuming, the problem is treated in preliminary design phase as a better base-design.

**Conclusion**

The preliminary design optimization of an axial-compressor is made using the axisymmetric flow solver to simulate the flow field. In the conceptual design phase, the mean-line method is used to compute the axial variations where the free-vortex assumption helps calculating the radial distribution in the flow field. So the axial velocity profiles in different rows are uniform as well as the total pressure and total temperature. But this is altered due to the flow area change and advent of radial velocities. Using developed fast axisymmetric flow solver, the flow field of the compressor is simulated more accurately which shows some drawbacks as axial velocity decrease at the hub of most rows, which should be improved especially that it may be intensified in the off-design performance and leads to lower surge margins. Since the optimization process of the detail design phase is very time-consuming, the problem is treated in preliminary design phase using the flow solver before the blading process. The target is increasing the minimum axial velocity of the flow field without significant change in total pressure ratio and mass flow rate of the compressor. The results show that the minor modification in flow deflection angles is effective and applying two degrees change on the deflection angles of all stages except the first one, as a second order function of the blade radius, increases the minimum axial velocity about 10 percent in preliminary design optimization, which can be used then in the blading process and optimizations in detail design phase as a better base-design.

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