Performance Analysis of Axial Flow Compressor and Part Load Consideration in a Gas Turbine Application*

Majid ASLI** and Abolghasem M. TOUSI **

**Department of Aerospace Engineering, Amirkabir University of Technology
No. 424, Hafez Ave., Tehran, Iran
E-mail:m_asli@aut.ac.ir

Abstract

A performance analysis of a three stage axial flow compressor with part load consideration in gas turbine application regarding surge margin is presented in this paper. A three stage compressor was designed and numerical simulation as well as a modified stream line curvature analysis was applied to validate the design model. These two analyses verified the design which was based on a test case model and showed an acceptable accuracy. Performance was predicted by introducing a modified one dimensional method for the compressor. A typical gas turbine with specific components was considered to evaluate the effect of compressor early design parameters on its behavior at part loads. Four compressors were designed with change in reaction ratio and flow coefficient as the input data of the design process. The analysis showed that an increase of 8% and decrease of 5% in reaction ratio and flow coefficient respectively, compared with the base design model, would lead to getting away the running line from the surge line by an amount of 5% approximately and working in a safer margin as well as higher efficiency working range, so the energy conversion can be done efficiently.

Key words: Performance, Gas Turbine, Compressor Characteristics, Part Speed, Working Line, Pressure Ratio

1. Introduction

In a jet engine, the compressor operates with different rotational speed to produce a variety of demanded thrust, ranging from taxing to take off conditions. Similarly, in an industrial gas turbine the same process produces a variety of demanded loads. Therefore, in both aero and industrial gas turbines, the engines operate under part load conditions in considerable period of time and a proper action should be taken to raise the engine efficiency and also preventing the compressor from exceeding the surge line. The operation of such compressors is limited by instabilities causing the full breakdown of flow termed as surge. These instabilities caused by high incidence occurring due to different phenomena at part- and full-speed operation. Once in flight the inlet pressure drops, but the inlet speed increases (due to the forward motion of the aircraft) to recover some of this pressure, and the compressor tends to run at a single speed for long periods of time. Clearly, this large variation in rotor RPM and mass flow rate both need to be considered while designing an axial flow compressor. On the other hand, a gas turbine works under part load conditions most of the time. In these kinds of situations, the efficiency falls. So part load remedies should be considered before designing a gas turbine (1). The problem at part load speed operation is that the front stages are often heavily stalled and rear stages choked; whereas at
high speeds, the front stages are operating close to choke and the rear stages tend to stall. Variable inlet and nozzle guide vanes and bleeding are common techniques for part load remedies. A large number of gas turbines make use of a combination of control strategies throughout the operating range whether they run standalone (2, 3). Haglind (4) studied these two methods in combined cycles for industrial applications, with and without variable geometry gas turbines.

Optimization of the design to high-speed conditions may provide problems at part-speed and to achieve the acceptable performance at part-speed operation, variable inlet guiding vanes and the front-stage variable stator vane settings are sometimes used to modify the flow angles to avoid stage stall and subsequent compressor surge (5,6,7). Sun et. al (8) presented a numerical methodology for optimizing a stator stagger setting in a multistage axial-flow compressor. According to his results, variable stagger is a powerful method to rematch stages which can be used to improve desired overall performance. In addition, some researches were done on compressor sensitivity to its design parameters. Vincent (9) and Lavannie (10) at MIT consider variation of geometric parameters on a repeating stage compressor performance. A three dimentional analysis using Reynolds Average Navier Stokes solver conducted by Lavannie revealed a large effect of tip clearance on mean performance characteristics. There are several works were done on design consideration in axial flow compressor characteristics, stall margin and optimum design (11,12,13,14).

In the present paper a typical three stage axial flow compressor is designed using a computer code by the writer and the effect of its design parameters was studied on part load performance. Validation of the design code is done based on NASA 74-A data (15, 16), CFD tool and a modified stream line curvature developed by the writer (17). However, the aim of the design is to investigate primary design parameters at early design stages and their effects on full and part load performance of a gas turbine.

2. Results and Discussion

2.1. Design

The model studied in this paper is geometrically similar to NASA 74-A three stage axial flow compressor. The compressor overall pressure ratio and its efficiency are 3.94 and 85.4% respectively. Mass flow rate is 29.71 kg/sec and blade profiles are of double circular arc type. A computer code is prepared to design the compressor using overall scale sizes as inputs. The code is able to determine the flow properties at each blade leading and trailing edge and blade profile in details.

Blade design is done in the basis of obtaining the required velocity diagram at the blade inlet and outlet. It can be defined by incidence, deviation and camber angles and also the blade thickness distribution along the blade profile chord as well. In this design multiple circular arc blade profile is used.

The correlations applied in the design have been selected on the basis of simplicity and appropriateness and have been modified where necessary to give better agreement with the reference compressor model. Lieblein correlation (18) is used to calculate blades profile and relevant angles. Applying this correlation leads to obtain incidence, deviation, chamber and other blade angles. Loss sources considered in this design include profile loss (distinguished as the most significant source of loss in compressor) and end wall loss. Aungier (19) defines profile loss based on equivalent diffusion ratio ($D_{eq}$) and Reynolds effect coefficient ($k$), see Eq. (1). Equivalent diffusion ratio is the ratio of maximum possible relative velocity and the exit relative component of velocity. Lieblein developed the correlation of Eq. (2) to calculate the equivalent diffusion ratio for operation at minimum loss.
\[
\frac{\omega_p \cos \beta_2}{2\sigma} \left( \frac{W_1}{W_2} \right)^2 = 0.004 \left( k + 3.1(D_{eq} - 1)^2 + 0.4(D_{eq} - 1)^8 \right)
\]

(1)

\[
\frac{W_{max}}{W_1} = 1.12 + 0.61 \frac{\cos^2 \beta_1}{\sigma} \left( \tan \beta_1 - \tan \beta_2 \right)
\]

(2)

End wall loss \((\omega_{ew})\) is defined as a function of diffusion ratio and tip clearance by Eq. (3)\(^{(20)}\). So the total loss coefficient is the sum of these two coefficients. See Eq. (4).

\[
\omega_{ew} = \frac{h}{c} \left( \frac{W_1}{W_2} \right)^2 = f \left( \frac{c}{c}, D \right)
\]

(3)

\[
\omega = \omega_p + \omega_{ew}
\]

(4)

It can be shown that how much the designed compressor stages are near to stall condition by comparing the static pressure raise coefficient based on pitch line dynamic head and its maximum allowable value. The maximum static pressure raise coefficient defined in the basis of Reynolds effect coefficient, tip leakage and axial spacing of blade rows defined by Koch \(^{(21)}\) as Eq. (5).

\[
C_{p, Max} = C_{PD} F_{ef} \frac{C_p}{C_{PD}} \left( \frac{C_p}{C_{PD}} \right)_{Re} \left( \frac{C_p}{C_{PD}} \right)_{e} \left( \frac{C_p}{C_{PD}} \right)_{DZ}
\]

(5)

“\(F_{ef}\)” is the effective dynamic pressure factor, was defined as the effective dynamic head divided by the pitch line free stream dynamic head.

Running the code with several loops resulted in obtaining the compressor flow properties and blade geometry. A number of specifications are listed in Table 1. Fig. 1 (a) shows the 3D model of the first compressor rotor blades. Variation of loading factor regarding to flow coefficient is plotted for each stages in Fig. 1 (b).

<table>
<thead>
<tr>
<th>Table 1 Compressor design parameters and specification.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stage</td>
</tr>
<tr>
<td>-------</td>
</tr>
<tr>
<td>Efficiency</td>
</tr>
<tr>
<td>Pressure ratio</td>
</tr>
<tr>
<td>Total temp. ratio</td>
</tr>
<tr>
<td>D-Haller No.</td>
</tr>
<tr>
<td>Diffusion ratio</td>
</tr>
<tr>
<td>Reaction ratio</td>
</tr>
<tr>
<td>Flow coefficient</td>
</tr>
<tr>
<td>Blade profile</td>
</tr>
<tr>
<td>Rotational speed (RPM)</td>
</tr>
<tr>
<td>Mass flow rate (kg/sec)</td>
</tr>
</tbody>
</table>

* Multiple Circular Arc
2.2. Validation

To validate the design code, a through flow analysis based on stream line curvature method was applied. Early research and development of this method was done by Denton (22). The SLC method involves solutions for the equations of continuity, motion, energy, and state in the compressor flow field on each stream line. The method solves the discrete equations on a computational domain (grid) which is constructed in the meridional plane bounded by the compressor annulus geometry. Grid nodes are defined at the intersections of the streamlines and the blade edges. The initial boundary conditions are set by the ambient conditions of the incoming air, the mass flow, and the compressor rotational speed. After transforming the discrete equations, the result is a non-linear partial differential equation solved by a finite difference approximation method following an iterative approach (23). It is generally accepted that any streamline curvature solution technique will yield satisfactory flow solutions if the deviation, losses, and blockages are accurately applied in the code. The loss correlations used in the SLC code modified by the writers, calculate the relative total pressure loss coefficient, by Eq.(6). The SLCC (Stream Line Curvature Code) calculates loss at a given compressor operating point by developing a "loss bucket". A loss bucket is the graphical representation of loss as a function of incidence for constant Mach number. The curve normally has a V-shape or bucket shape with the middle of the bottom of the bucket being the minimum loss incidence point.

Profile loss (blade section boundary layer and wake) represented by $\omega_{min}$, is determined using the method of Koch and Smith (24). Shock losses and Mach number effect is modelled using normal shock and isentropic relations developed by Miller et al. (25). The effects of secondary loss are considered in the last two terms of Eq. (6), $\omega_{hub}$ and $\omega_{shroud}$ defined by Hearsy (26). According to Casey (27) span-wise mass transfer is considered because, due to some mechanisms that lead to an apparent span-wise transport of fluid relative to flow on the mean streamlines such as non-axisymmetric blade-to-blade stream surfaces as result of stream-wise vortices being shed by the blades, secondary flows in the

$$\omega = \left[ (\sigma_{min} + \sigma_M + \sigma_{hub} + \sigma_{shroud}) \left[ 1 + \left( \frac{i - i_w}{W} \right) \right]^2 \right]$$

(6)
end-wall boundary layers and in the blade boundary layers, wake momentum transport downstream of blade rows, tip clearance flows with tip clearance vortices and turbulent diffusion span-wise mixing are inevitable phenomena. Fig. 2 shows the pressure contour of the designed compressor in meridional plane obtained by stream line curvature code.

In addition, an algebraic multi-grid method is applied to the coupled linearized discrete equations arising from an implicit pressure based finite volume discretization system of equations. The whole methodology is modelled by using CFX solver which mesh study is considered to obtain satisfactory results. The compressor modelled in the program and the three stages were solved simultaneously. The computational domain and the first stage of the compressor are indicated in Fig. 3. Pressure contour obtained from CFD analysis by CFX solver is shown in Fig. 4 and good matching with that of SLC method shown in Fig. 2 is obvious. Fig. 5 shows the results of design code, stream line curvature analysis, CFD analysis versus experimental data of Steinke and compares efficiency and pressure ratio of each stage obtained by the mentioned methods.

![Pressure contour in meridional plane obtained by SLC analysis](image1)

**Fig. 2.** Pressure contour in meridional plane obtained by SLC analysis

![The first rotor of the compressor and computational domain in CFD tool](image2)

**Fig. 3.** The first rotor of the compressor and computational domain in CFD tool
Fig. 4. Compressor annulus pressure contour in meridional plane obtained by CFX

Fig. 5. Comparison of efficiency and pressure ratio of each stage between Design code, Stream line curvature, CFD analysis and Steinke experimental results.

It is evident that a good matching exists between the two analysis methods of SLC and CFD with the design code results. Just there is an average difference of maximum 5 percent in efficiency which is negligible in design model accuracy discussion. In addition, Steinke results have an acceptable consistent with the design model with a maximum error of 6 percent in pressure ratio.

2.3. Performance Analysis

The method used for performance prediction of the compressor is the mean line method using stage stacking procedure based on Steinke (28) method. Some investigators (29,30,31) used this kind of stacking method widely, but Steinke’s code availability make it more
preferred to others. This one dimensional method is fast, general, and easy to use. Determining the power level for a given stage is essentially a 1D problem; the Euler equation shows that one must get the flow angles corrects (deviation or slip) to set power levels properly.

In this model, the annulus and stage/blade geometry for all stages, the rotor rotational speed and the overall conditions at the inlet of the first stage obtained from design code are specified as input data to the 1D code.

The method involves a relatively simple individual stage characteristics prediction procedure, in which a one dimensional analysis of velocity vectors at root mean square radius $r_{rms}^2 = (r_{hub}^2 + r_{tip}^2)/2$ is used. Values of physical flow quantities at this radius are taken to represent the compressor under consideration. The overall compressor characteristic is established by ‘stacking’ stages at a given rotational speed (16). The stage stacking code applied here is presented and validated in ref. (32). This analysis is performed for four rotational speeds; 100, 90, 80 and 70 percent of design speed. The 1D performance prediction code results were compared against the experimental data which is provided by Steinke in Fig. 6 in which the compressor pressure ratio characteristic versus experimental data of NASA compressor 74-A at four part speed are presented. As it is shown, the performance prediction code results for the specific compressor which is under consideration meet the Steinke’s results. The differences are acceptable in accordance with the differences between the two design details and one dimensional simulation.

![Fig. 6. Compressor performance characteristic and experimental data](image)

Early design parameters which are considered in preliminary design steps have effects on compressor characteristics undoubtedly. Here, some design parameter distributions are studied and the effects are observed. Four cases are designed in which their differences with the base design (case 0) are as follows.

- Case 1: reduction in flow coefficient by 5%;
- Case 2: increase in flow coefficient by 5%;
- Case 3: reduction in reaction ratio by 8%;
Case 4: increase in reaction ratio by 8%.

Each case leads to a different compressor design considering the changes in mean line preliminary design parameters mentioned. The compressors characteristics are shown in Fig. 7. According to Fig. 7 (a), the characteristic lines at design and part speeds are shifted lower in case 1, so the compressor is designed with lower mass flow rate and pressure ratio at the same design rotational speed while an increase in flow coefficient causes the new compressor to operate at a higher mass flow rate and pressure ratio as well which is illustrated in case 2. The variation of flow coefficient can change the rotational speed of the compressor if the mass flow rate is kept constant but, as far as this kind of compressor naturally works in transonic criteria and to avoid shock waves and other related effects due to transition, constant rotational speed is considered by variation of flow coefficient. Reduction and increase in reaction ratio in cases 3 and 4 respectively, cause the new compressor to work in a higher and lower pressure ratio at part speeds. According to Fig. 7 (b), reduction of the reaction ratio, change characteristics lines to higher mass flow rate and pressure ratio, but around the choke conditions, characteristic lines reach those of the design. It is shown that reduction in reaction ratio may not improve the characteristics around the choke condition. Also it is shown that in each case, the surge line limits are changed as well.

These types of compressor design are considered in a typical gas turbine application to investigate the working line. A simple single shaft gas turbine is studied with the exit burner temperature of 1450K and turbine pressure ratio of 1.6, so the five compressors are studied as part of this kind of gas turbine. Then off-design calculation is done and the working lines of the gas turbine are presented as results in Fig. 8.

![Fig. 7. (a), (b) The comparison of compressor performance characteristics in case0, case1, case2 and case3, case4 respectively](image-url)
The main scope of off design calculation is to investigate how near the gas turbine working line is to surge condition. As it can be seen through the Fig. 8, each design has a distinct working line. The working line of the case 0 is the upper one and it’s closer than other four cases to its surge line. According to Fig. 9, increasing the flow coefficient and reducing the reaction ratio in preliminary design consideration of the compressor lead to a change in running line of the gas turbine and cause to work in a safer margin from surge condition in full and part speed. In addition, decreasing the flow coefficient and increasing the reaction ratio cause the gas turbine to work closer to the surge condition.

Fig. 8. Working lines of the five typical gas turbines

Fig. 9. Working lines in compressor pressure ratio characteristics for the case number 0, 1, 2 and 3, 4.
3. Conclusion

Here a three stage axial compressor design was introduced and the improved mean line analysis for compressor performance prediction was developed and applied, then the effects of early design parameters on compressor characteristics and gas turbine performance at part load were studied as well. Five compressor designs were investigated in gas turbine application and the results were discussed. Finally, it is concluded that selecting a wise non dimensional parameters of reaction ratio and flow coefficient for multistage axial compressor early design have considerable effects on gas turbine application and its operating criteria regarding surge phenomenon. In this case, small increase in flow coefficient and decrease in reaction ratio in early design steps have positive effect on working line position in compressor characteristics map. There are some other methods for part load remedies of gas turbines like variable inlet guide vane in which the running line is kept fix and just compressor characteristics are shifted by variation of vanes angle. However, by changing the early compressor design parameters, both characteristics and working line are displaced. These kinds of changes must be within the compressor performance limitations and some parameters like loading factor, diffusion ratio and efficiency must be checked continuously. This analysis relates the early design considerations to final gas turbine application and an optimization process can be done to select optimized early design parameters regarding the specific multistage compressor application.

In conclusion, further study required on optimization of the parameters regarding to a specific gas turbine application. Some optimization method such as neural network may be applicable and useful to find optimum compressor design early parameters to get higher level of energy efficiencies and pressure ratio regarding safer margin in gas turbine application.

References


(9) A. Vincent; *Impact of Geometric Variability on Compressor Repeating-Stage Performance*, (2003), Master Thesis, Massachusetts Institute of Technology.

(10) J. Lavannie; *Sensitivity of a Compressor Repeating-Stage to Geometric Variation*, Master Thesis, (2003), Massachusetts Institute of Technology.


