Uniport Journal of Engineering and Scientific Research (UJESR)

Vol. 5, Issue 2, 2021, Page 6-12
ISSN: 2616-1192

(www.ujesr.org)

Preliminary Re-design of a Twelve Stage Axial Compressor in an Existing Single Shaft Industrial Gas Turbine Engine

Agbadede, R*[1] and Kainga, B[2]
[1]Department of Electrical Engineering, Nigeria Maritime University, Okerenkoko Warri, Delta State, Nigeria.
[2]Department of Mechanical Engineering, Nigeria Maritime University, Okerenkoko Warri, Delta State, Nigeria
*Corresponding author's email: roupa.agbadede@nmu.edu.ng

Abstract
This study presents a preliminary design of an axial compressor with a view to meeting the increased load requirement from the gas turbine output. The compressor is redesigned to match the turbine effectively, so the increased power demand of the mini-city harbouring an oil rig location from the gas turbine engine can be achieved. The existing gas turbine engine with a power output of 42MW could not supply the desired power thus calls for the need to embark on a research for possible improvement. Using proprietary gas turbine performance software called TURBOMATCH and a computer program written in Microsoft Excel, a redesign of the axial compressor was achieved. From the study, it was observed that most of the aerodynamic performance parameters are within acceptable limits, hence the preliminary design is deemed feasible. Though the hub-to-tip ratio at the inlet annulus is acceptable, the one at the outlet annulus is above the stipulated limit of 0.91. Also, the loading coefficient at the tip of blade which is 0.243 is some fractions below the minimum required level of 0.25.

Keywords: *Axial Velocity, Aerodynamic Performance, Hub-to-Tip Ratio, Inlet Annulus, Outlet Annulus*

Received: 28th August, 2020 *Accepted: 12th October, 2020*

1. Introduction

The performance of axial compressor is sensitive to fine details of its aerodynamic design. As a result, it is imperative to apply aerodynamic design theory when designing the compressor so as to prevent wasteful losses and stall problems. Designing an axial compressor to optimize its performance requires enormous effort and time. This is because slight changes in any of the design parameters may result in significant variation in another. In addition, each step of the design and performance analysis of any component is quite time consuming. This study presents a preliminary design of an axial compressor with a view to meeting the increased load requirement from the turbine end. The main drive for this redesign is to generate a power output within the range of 45 – 60 MW of an existing 42MW gas turbine which was not available in the parent engine. As part of the entire processes, this report is aimed at performing a critical design and conduct a well-designed feasibility study in line with acceptable aerodynamic limitations.

Al-Druby et al. (2008) developed a numerical calculation algorithm to design thermodynamic and aerodynamic parameters of an axial compressor. The authors reported that when the designed algorithm was tested on a compressor cascade, the results show that lift coefficient decreased with increased mean flow angle. Similarly, the total drag to lift coefficient ratio increased with mean flow angle. Massardo and Satta (1989) presented a technique where pitch-line analysis was used to evaluate the objective function and constraints facing compressor designs. An optimized design with multivariable objective function was reportedly achieved in the study. Three variables selected, namely stage efficiency, stall margin and the inlet specific area were all evaluated using the pitch-line analysis.

Schnoes et al. (2018) conducted a detailed blade row design with 3D CFD. A large number of airfoil shapes was optimized to meet design requirements of the new class of airfoils developed. In addition, the authors employed machine learning to predict the optimum airfoil shape and performance. Obanor et al. (2015) conducted a preliminary design of an axial compressor for power generation application using an algorithm that was developed by the authors. The design procedure and parameters obtained were validated by comparing the results from the design to the field engine performance data. Xiaoqing et al. (2008) proposed a method of designing axial compressor using low reaction, highly loaded and boundary layer suction. The feasibility of the design concept was validated by conducting analysis on a low reaction stage. In addition, the authors stated that the new concept can be used to design high pressure stages of aero engines, high pressure ratio blower and transonic stages of industrial gas turbines. Design optimization of an axial flow compressor for industrial gas turbine was presented in Salunke et al. (2014). A set of parameters were used for the preliminary design of the compressor. Also, the design parameters were tested for stall at different mass flow rates for a given pressure ratio. The study showed that the optimized design was in close agreement with theoretical design data. Kumar and Alone (2019) employed Steady RANS 3D CFD simulations to investigate a three stage axial compressor design, to ascertain the interaction among stages and the behaviour of different flow parameters at design point, choke point and stall conditions. From the literature search conducted, it is obvious that there has never been any preliminary re-design of an existing axial compressor in a gas turbine so as to meet the increased load demand. This study therefore presents the preliminary redesign of a twelve stage axial compressor in order to meet the increased load demand from the gas turbine power output.

2. Materials and methods

To carry out the redesign of the twelve stage axial compressor, an existing gas turbine specification data obtained from open domain were used to model the new proposed engine. The modelling and simulation of the proposed engine was done using a gas turbine performance simulation software, TURBOMATCH. The modelling was done by altering some of the design point performance parameters of the old engine until the desired power output of the new proposed engine was achieved. Also, a computer program written in Microsoft Excel was used to design the new optimized axial compressor. The program written in Microsoft Excel incorporates thermodynamic and aerodynamic equations to carry out the redesigned of the axial compressor. After obtaining the design performance specifications of the new proposed engine from the simulations, the parameters such as mass flow rate, compressor inlet pressure and temperature were used as input into the code written in Microsoft Excel to design the new axial compressor for the proposed gas turbine engine. In addition to these input parameters mentioned, few assumptions were made in order to design the new axial compressor. The assumptions made are presented below:

The overall diameter of the engine was assumed to be fixed; hence the new design of the compressor should be in line with the baseline engine. With this assumption the core of the old/existing engine is maintained in the new design. An axial velocity of 210m/s was chosen for the preliminary design because Ramsden (2009) stated that the inlet axial velocity experienced in a compressor is in the range of 150 to 210m/s. Also, it was assumed that the axial velocity will be constant throughout all the stages of the compressor. A polytrophic efficiency of 0.88 was chosen for the initial calculation of the compressor isentropic efficiency of the design. Also, it was assumed that there are no inlet guide vanes; therefore, there will be no whirl component at the inlet velocity (Saravanamutto et al., 2009). A constant outside diameter was assumed for the annulus configuration for the preliminary design of the compressor in order to have lower blade height at the outlet. Also, the value of the inlet hub to tip ratio of 0.68 was chosen (Ramsden, 2009), and a free vortex flow was assumed as the design choice. The flow angle at the inlet of the compressor was assumed to be zero throughout the stages of the compressor since the compressor has no inlet guide vane. This implies that the inlet axial velocity (Va) equals Vo. A mean blade speed of 289m/s was chosen for this design. This is because according to Saravanamutto et al. (2009) that tip blade speed should not exceed 350m/s. Consequently, the input data from the design specifications and assumed parameters were implanted into the code, after which several iterations were carried out to arrive at the design parameters deemed satisfactory for the new axial compressor design. This was achieved by using the aerodynamic limiting factors such as the De Haller number, stage loading coefficient, flow coefficient and the static pressure rise, to certify if the new designed axial compressor is aerodynamically feasible. Some of the thermodynamic and aerodynamic equations used in formulating the computer code employed for the axial compressor design are presented below:

Equation (1) shows the relationship between inlet axial velocity and inlet temperature, and was used to obtain the values of flow coefficient from the compressible flow tables (Ramsden, 2009).

$$Va_1 / \sqrt{T_1} \quad (1)$$

where Va_1 is the axial velocity and T_1 is the inlet temperature.
Consequently, annulus inlet area of the compressor stage was calculated using Equation (2).

$$Q_0 = W\sqrt{T_1/K_B}.A_1.P_1 \quad (2)$$

where Q_o is the flow coefficient, W is the mass flow, K_B is the blockage factor, A_1 is the annulus inlet area, and P_1 is the inlet pressure.
Since inlet hub/tip ratio is known, the tip diameter can be calculated using Equation (3).

$$A_1 = \frac{\pi}{4}{D_{t1}}^2\left(1-\left(\frac{D_{h1}}{D_{t1}}\right)^2\right) \quad (3)$$

where D_{t1} is the tip diameter and D_{h1} is the hub diameter.
Using Equation (4), the compressor overall efficiency was calculated.

$$\eta_c = \frac{PR^{\left(\frac{\gamma-1}{\gamma}\right)}-1}{PR^{\left(\frac{\gamma-1}{\gamma\eta p}\right)}-1} \quad (4)$$

where PR is the pressure ratio, η_p is the polytropic efficiency, η_c is the compressor isentropic efficiency, and γ is the specific heats ratio.
The overall temperature increase was obtained from Equation (5).

$$\Delta T = \frac{T_1}{\eta_c}\left(PR^{\left(\frac{\gamma-1}{\gamma}\right)}-1\right) \quad (5)$$

Hence, compressor stage outlet temperature was obtained using Equation (6).

$$T_2 = T_1 + \Delta T \quad (6)$$

where ΔT is the temperature increase and T_2 is the compressor stage exit temperature.
These equations provided above can be used to obtained inlet and outlet conditions of any number of stages in a compressor where the stage parameters can be changed accordingly. The rotational speed of the compressor was calculated using Equation (7), having obtained the assumed blade mean speed and hub-to-tip ratio; while the De Haller number was calculated using Equation (8).

$$U_m = \pi\ x\ D_m\ x\ \frac{N}{60} \quad (7)$$

where U_m is the mean speed, D_m is the mean diameter, and N is the rotational speed.

$$De\ Haller\ Number,\ \frac{\Delta H}{U^2} = \frac{V_1}{V_2} \quad (8)$$

where V_1 is the rotor inlet velocity and V_2 is the rotor outlet velocity.
Using Equation (9), the pressure rise coefficient was calculated.

$$\frac{\Delta p}{D} = 1-\left(\frac{V_2}{V_1}\right)^2 \quad (9)$$

3. Results

Table 1 presents preliminary design specifications of the new axial compressor.

Table 1: Preliminary design specifications

Specifications	Value
Power (MW)	45
Mass flow (kg/s)	141
Mean blade speed (m/s)	289 (chosen)
Pressure ratio	12.380
Inlet hub to tip ratio	0.68 (chosen)
Inlet temperature(K)	288
Inlet pressure (Pa)	101325
Axial velocity (m/s)	210 (chosen)
Polytrophic efficiency	0.88 (chosen)
Number of stages	12
Rotational speed (rpm)	5166

Table 2 shows the results for annulus geometry of the axial compressor designed, while Table 3 and 4 show the compressor design results and the aerodynamic performance analysis respectively.

Table 2: Calculated results for compressor annulus geometry

Parameters	First Stage			Last Stage		
	Rotor Inlet	**Intermediate**	**Rotor Outlet**	**Rotor Inlet**	**Intermediate**	**Rotor Outlet**
Inlet Annulus Dimension						
Annulus Area (m2)	0.683		0.557			
Tip Diameter (m)	1.272		1.272			
Mean Diameter (m)	1.068		1.112			
Hub Diameter (m)	0.865		0.953			
Annulus (Blade) Height (m)	0.203		0.159			
Intermediate Annulus Dimension						
Annulus Area (m2)		0.623			0.033	
Tip Diameter (m)		1.272			1.272	
Mean Diameter (m)		1.090			1.239	
Hub Diameter (m)		0.909			1.207	
Outlet Annulus Dimension						
Annulus Area (m2)				0.132		0.119
Tip Diameter (m)				1.272		1.272
Mean Diameter (m)				1.234		1.241
Hub Diameter (m)				1.204		1.211
Annulus(Blade) Height (m)				0.034		0.031

Table 3: Calculated Axial Compressor Design Results

Design Parameters	Design Values				Design Limit
	First Stage			Last Stage	
	Mean	**Tip**	**Hub**	**Mean**	
		Rotor Inlet			
Alpha α_0 (degrees)	0.00	0.00	0.00	0.00	
Alpha α_1 (degrees)	53.99	58.59	48.22	58	
Va_1(m/s)	210.00	210.00	210.00	210.00	$130<Va<250$
V_1(m/s)	357	403.07	314.38	395.24	
U_1(m/s)	289	344.04	245.84	335.31	
		Rotor Outlet			
Alpha α_2 (degrees)	43.22	51.11	31.52	49.91	
Alpha α_3 (degrees)	24.90	21.71	29.12	22.22	
Va_2(m/s)	210.00	210.00	210.00	210.14	
V_2 (m/s)	288.22	334.55	246.37	326.14	
V_3(m/s)	231.54	210.00	240.39	226.84	
Vw_2(m/s)	197.41	260.44	128.83	249.53	
Vw_3(m/s)	97.52	83.61	117.00	85.78	
U_2(m/s)	294.94	344.05	245.84	335.31	

Table 4: Axial compressor aerodynamic performance results

Design Parameters	First Stage			Last Stage	Design Limit	Comments
	Mean	**Tip**	**Hub**	**Mean**		
De Haller Number (V_2/V_1)	0.81	0.83	0.78	0.83	> 0.65	Within design limit
Flow coefficient (Va/U_2)	0.57	0.46	0.74	0.49	0.4 - 0.9	Within design limit
Stage loading coefficient ($\Delta H/U_2^2$)	0.33	0.24	0.47	0.26	0.25 – 0.5	Tip is below the acceptable limit
Static pressure rise ($\Delta P/D$)	0.35	0.31	0.38	0.32	Max. 0.5	Within design limit

Rotor inlet absolute Mach number	0.64	0.64	0.64	0.42		
Rotor inlet relative Mach number	1.088	1.22	1.04	0.78	Tip < 1.3	Within design limits
Rotor exit absolute Mach number	0.67	0.66	0.70	0.45		

Figures 1 and 2 show the schematic of the first and last stage of the compressor annulus respectively. Figures 3, 4, 5 and 6 show the combined velocity triangles for first stage mean, tip and hub diameters and last stage mean diameter respectively. The velocity triangles are presented to provide good knowledge of the stage loading and axial velocity, once the flow angles are decided (Gallimore, 1999).

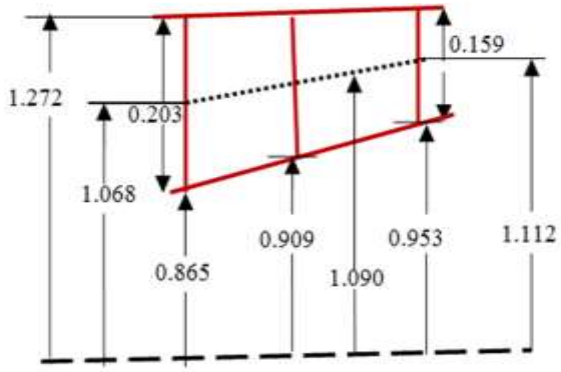

Fig. 1: Schematic of first stage annulus of the axial compressor

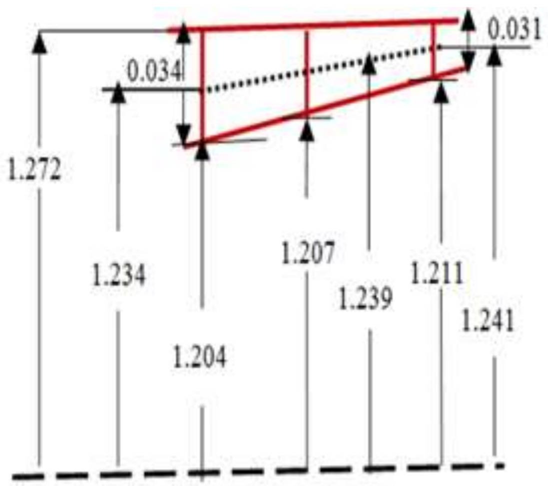

Fig. 2: Schematic of last stage annulus of the axial compressor

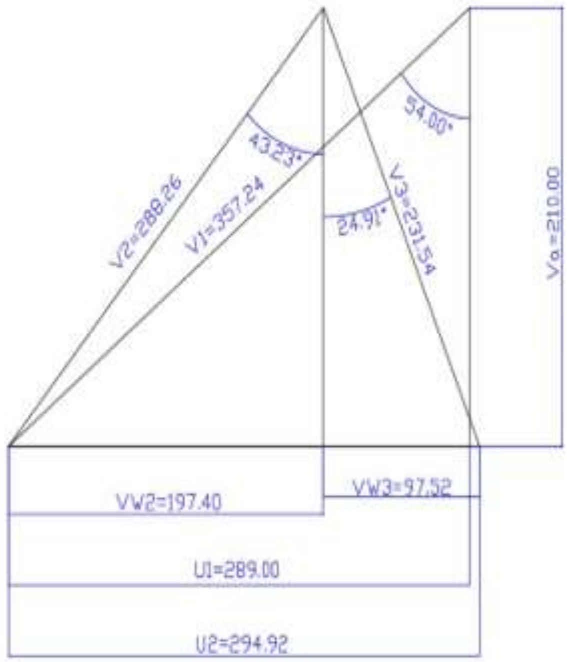

Fig. 3: First stage combined velocity triangle at mean diameter

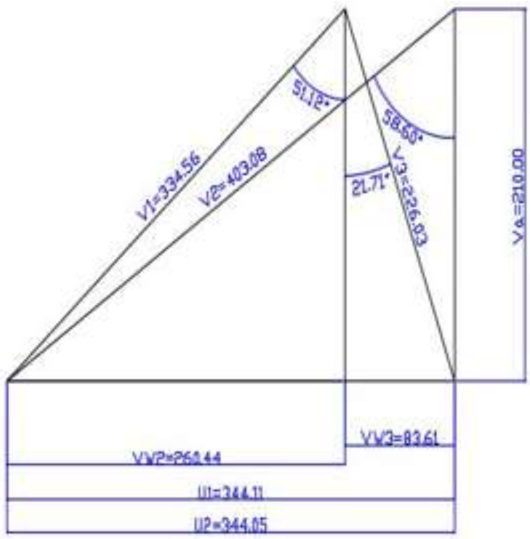

Fig. 4: First stage combined velocity triangle at tip diameter

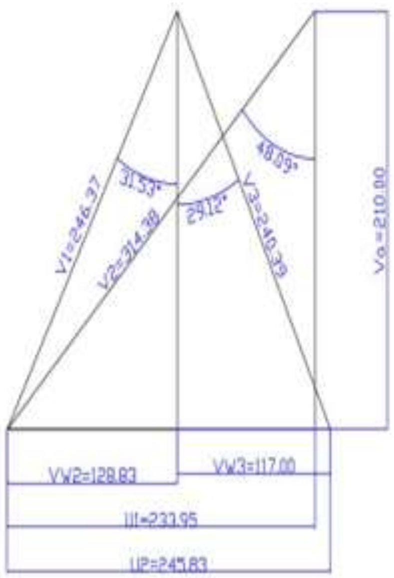

Fig. 5: First stage combined velocity triangle at hub diameter

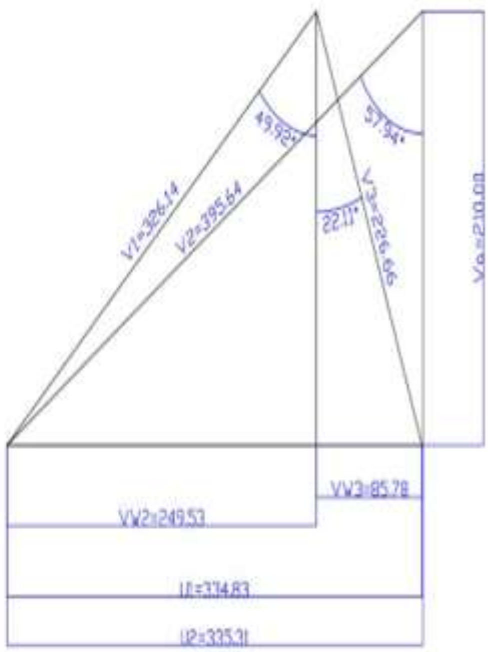

Fig. 6: Last stage combined velocity triangle at mean diameter

4. Discussion

The annulus geometry shown in Figures 1 and 2 seem satisfactory. From Table 3, it was observed that most of the aerodynamic performance parameters are within acceptable limits, hence the preliminary design is deemed feasible. Though the hub-to-tip ratio at the inlet annulus is acceptable, the one at the outlet annulus is above the stipulated limit of 0.91. The hub-to-tip ratio of 0.68 at the inlet is satisfactory, while the hub-to-tip ratio of 0.95 at the outlet is above the specified limit of 0.91. Despite the high hub-to-tip ratio obtained at the first stage outlet, if the suggested recommendations are implemented, the blade heights are not expected to present any mechanical integrity problems vis-à-vis high pressures and rotational speed encountered by the blade. Though it is desirable to satisfy all the aerodynamic limitations, experience has shown that not all the factors considered meet the desired limit at the preliminary design stage (Gallimore, 1999). The high hub to tip ratio of 0.95 which result from very short blade heights at out let of the last stage can be corrected by reducing the inlet axial Mach number with a view to reducing the axial velocity. Secondly, a constant annulus height should be maintained from the stage that has a hub-to-tip ratio of 0.92. Also, the loading coefficient at the tip of blade which is 0.243 is some fractions below the minimum required level of 0.25. The blade tip loading coefficient of 0.243 obtained from the design results is not expected to pose any negative effect on the design because the highest loading usually occurs at hub. Therefore, maintaining the number of compressor stages at 12 is in order.

5. Conclusions

This study presents a preliminary design of a modified twelve-stage compressor of a single shaft industrial gas turbine engine originally design by GE for power generation. A design feasibility analysis was carried out at the mean, hub and tip of the blade for a free vortex application. From the aerodynamic analysis, most of the design parameters met the design targets such as the De Haller number, stage loading coefficients, static pressure rise, and flow coefficient. These parameters are all within the acceptable limits for all the stages in the compressor except the stage loading at the blade tip, which falls below expected range of 0.25-0.5. But this does not pose any negative effect on the design because the maximum stresses occur at the root of the blade. From the results of the preliminary compressor design, the hub-to-tip ratio of 0.68 at the inlet annulus geometry is within the acceptable limits while the first stage outlet hub-to-tip ratio of 0.95 is above the specified limit of 0.91. This implies that the height of the blade would be very small, which could result in blading and performance problems, making the handling of the tip leakage flow to be difficult. Furthermore, this means that a high hub-to tip ratio will not be acceptable in the design of the last stage of the compressor. Therefore, corrective measures such as reducing the axial inlet Mach number which in turn reduces the axial velocity thereby leading to increase in blade heights at the exit of the last stage of compressor or to redesign the compressor in such a way by keeping a constant annulus height from the point where the hub to tip ratio reaches 0.9 are recommended.

Acknowledgement

The authors are exceedingly grateful to Nigeria Maritime University for the technical support.

Conflict of interest

The authors can confirm that there is no conflict of interest in this research work.

References

Al-Druby, I.M., Hussain, Y.I. and Al-Ruyee, I.R. (2008) Preliminary design of a multi-stage axial compressor. Journal of Engineering, 14(2): 2534-2551.

Gallimore, S.J. (1999) Axial compressor design. Proceedings Institution of Mechanical Engineers, 213:437-449.

Kumar, L. and Alone, B.D. (2019) Design and CFD analysis of a multi-stage axial flow compressor. 21st Annual CFD Symposium, CFD Division Aeronautical Society of India, 08-09 August, 2019, Bangalore, India.

Massardo, A. and Satta, A. (1989) Axial compressor design optimization. part i: pitchline analysis and multivariable objective function influence. ASME Gas Turbine and Aero-engine Congress and Exposition, June 4-8, 1989, Toronto, Ontario, Canada.

Obanor, A.I., Unuareokpa, O.J. and Egware, H.O. (2015) An Algorithm for the design of axial flow compressor of a power generating gas turbine. Nigeria Journal of Technology, 34(2): 314-324.

Ramsden, K.W. (2009) Gas turbine fundamentals and turbomachinery. MSc Lecture Notes, Department of Thermal Power, School of Engineering, Cranfield University, UK.

Salunke, P.N., Channiwala, A.S. and Juned, A.R.A. (2014) Design optimization of an axial flow compressor for industrial gas turbines. International Journal of Research in Engineering and Technology, 3(5): 458-464.

Saravanamutto, H.I.H., Rogers, G.F.C., Cohen, H. and Straznicky P.V. (2009) Gas turbine theory. 6th edition Pearson Education Limited, Edinburgh Gate Harlow, Essex, England.

Schnoes, M., Vob, C. and Nicke, E. (2018) Design optimization of a multi-stage axial compressor using through flow and database for optimal airfoils. Global Power and Propulsion Society 18th Conference, May 7-9, Monteal, Canada.

Xiaoqing, Q., Songtao, W., Quotai, F. and Zhongqi, W. (2008) Aerodynamics design and analysis of a low reaction axial compressor stage. Chinese Journal of Aeronautics, 21: 1-7.