Preliminary Re-design of an Axial Turbine in an Existing Engine to Meet the Increased Load Demand

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Abstract — This work presents a preliminary design of an axial turbine section in an industrial gas turbine. The design was necessitated following the need to provide a gas turbine of a power output in the range of 48 to 60MW for a mini-city harbouring an oil rig, which was not possible with the old engine. The turbine section is designed to produce a power capable of driving the compressor as well as produce a useful power for electricity. Using proprietary gas turbine performance simulation software called TURBOMATCH and a computer program written in Microsoft Excel, a redesign of the axial turbine component was achieved. Consequently, a preliminary analysis was carried out to ascertain the new turbine stages introduced. The analysis revealed that when one or two turbine stage(s) was used for new engine, it proved unsatisfactory as the blade loading coefficient and the flow efficiency were both beyond the limit acceptable for an optimum performance. A three stage turbine was finally employed having provided a loading coefficient of 2.1, 1.9 and 1.7 for the first, second and the last stages respectively.

Index Terms — Flow Coefficient, Computer Program, Blade Loading Coefficient, Blade Aerodynamics, Performance Simulation Software.

I. INTRODUCTION

The overall performance of gas turbine depends on how efficiently its turbomachinery components (compressor and turbine) are designed. Literature search has revealed that there are several techniques to improve the design of turbomachinery components in gas turbines. However, designing any component to optimize its performance requires enormous effort and time. This is because slight changes in any of the design parameters may result in large 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 turbine with a view to meeting the increased load requirement from a mini-city harbouring an oil rig. Ajoko [1] conducted a study with a view to reducing the weight and cost, while still maintaining the same weight-to-thrust ratio of a gas turbine. This was achieved by designing to reduce the number of multiple stages of the High Pressure Turbine (HPT) component. The author reported that a single stage pressure turbine was able to produce the required aerodynamic performance expected of the multi-stage one. In addition, it was able to drive the corresponding compressor. Furthermore, it was reported the designed single stage HPT satisfied all the aerodynamic performance conditions set out.

Brighenti et al. [2] used a mean-line technique to design a Free Power Turbine (FPT). The designed FPT geometries were obtained using a simulation software. Consequently, the design and off-design conditions of the modelled FPT were simulated using 3D Computational Fluid Dynamics (CFD) tool. The operational conditions at design and off design conditions were verified by comparing the results from the meanline approach to the 3D numerical simulations. Maia et al. [3] developed a program which uses main data from engine thermodynamic cycle calculations to design the Axial Flow Turbines (AFT). A general automated methodology for the design of axial turbines was proposed and validated [4]. Agromayor and Nord [5] proposed a mean-line and optimization methodology approach for the design of axial turbine of any number of stages. Sequential quadratic algorithm was used to execute the axial turbine preliminary design, which was formulated as a constraint optimization problem. An optimal mean line design of a small axial turbine with over 2million configurations was performed by Franchini [6] using different tools. Preliminary design and modelling of axial turbine suitable for small and low temperature, based on organic rankine cycle application was presented by Situmbeko and Inambao [7].

II. MATERIALS AND METHODS

To carry out the design of the new axial turbine which is meant to drive a twelve stage axial compressor as well as produce the necessary increased power demand from a mini-city harbouring an oil rig, an existing engine data obtained from open domain was modelled and simulated using a gas turbine performance simulation software, TURBOMATCH. The modelling was done by altering some of the Design Point (DP) performance parameters of the old engine until the desired power output of the new proposed engine was achieved. TABLE I shows the DP specification for the new engine. A computer program was written in Microsoft Excel based on mean line approach. The Code written in Microsoft Excel to design the axial turbine was formulated using thermodynamic and aerodynamics equations. After obtaining the design performance specifications of the new proposed engine from the simulations, parameters such as mass flow rate, turbine inlet pressure and temperature were used as input into the code written in Microsoft Excel to design the new axial turbine 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 discussed below:

### TABLE I: TURBINE OVERALL SPECIFICATION

| Parameter            | Value   |
|----------------------|---------|
| TET(K)               | 1400    |
| Mass Flow Rate (kg/s)| 144.0   |
| Power Output (MW)    | 45.4    |
| Rotational Speed (rpm)| 5166   |
| Pressure Ratio       | 12.38   |

A. Assumptions

1. **Constant Blade Mean Velocity**
   
   The mean velocity was assumed constant in line with the compressor delivery mean velocity [8]. Also, since the compressor and the turbine are in tandem to one another, the rotational and mean blade speed were assumed to be constant.

2. **Constant Mean Diameter**
   
   The annular mean diameter describes the flow behaviour of the inlet compressed air and the hot gas as they pass from one stage to another. It is basically described to allow the preliminary design process to be initiated [9]. Also, constant rotational speed and blade mean speed imply a constant mean diameter as required.

3. **Degree of Reaction at Blade Mid-Height**
   
   Reaction is the ratio of the static temperature drop at the rotor to the total temperature drop at a given stage. The reaction varies from the root to tip of the blade but has to be retained at 50% of the mean diameter, to ensure evenly distribution of flow across the stages [9].

4. **Free vortex flow distribution**
   
   As required, the whirl velocity is inversely proportional to the radius from the root to the tip of the blade [9]. With the constant blade speed and constant specific heat capacity across the stages, a radial equilibrium condition is established.

5. **Straight sided Annulus**
   
   For ease of design at this preliminary stage, the annulus of the turbine was assumed to be straight sided [10].

III. RESULTS AND DISCUSSION

A. **Combined Velocity Triangles**
   
   Velocity triangle is a vital tool in the evaluation of the flow direction and thus the power produced by the turbine. In this case of constant blade speed and axial velocity at a given stage, the rotor inlet and the outlet triangles can be superimposed to analyze their combine effects. The different angles describing the direction of flow of the gas is adequately illustrated in the velocity triangle in Fig. 1. The product of the blade speed, U, and the apex vector, ΔVw is the stage work input [8]. Turbine specific power developed is obtained using the performance analysis. The value obtained is then compared to the product of the base vector and apex velocity. The absolute and the relative values are obtained and the velocity triangle for a constant mean diameter at the root of the turbine shown in Fig. 2 and 3.

At the blade mid height, the following are evaluated:

\[
\text{Stage work input} \quad \Delta H = U \Delta Vw
\]

From the cycle analysis, we have:

\[
\Delta H = \frac{\text{(Stage power output)}}{\text{Stage Mass flow}}
\]

i.e.,

\[
W \times Cp \times \Delta T = \text{Power output}
\]

where \( \Delta H = Cp \times \Delta T \).

The close approximation of the cycle calculation confirms the reliability of the velocity triangle in describing the work balance in the turbine section. Several assumptions were made which may be responsible for the little variation observed.

![Fig. 1. Velocity triangle at the root of the turbine for a constant mean diameter.](image1)

![Fig. 2. Velocity triangle at the blade mid height for a constant mean diameter.](image2)

![Fig. 3. Velocity triangle at the tip for a constant mean diameter.](image3)

B. **Design Evaluation**
   
   Three stages of a turbine were design to accommodate the blade loading and the flow coefficient which would have been impossible with the desired output. The figures
presented above represent the velocity diagram of the hub, mid height and the tip of the first stage and the velocity diagram for the mid height for the last stage. These are meant to drive twelve stages of compressors and other ancillary components of the gas turbine in a single spool while providing sufficient useful power. The design is basically guided by the philosophy of maintaining to the same shaft rotational speed and mean diameter of the compressor exit rotor. Having maintained these conditions, the axial Mach number, the rotor exit swirl angle, NGV leaving gas angle, the deflection angles, the rotor acceleration and the hub-tip ratio and thus the stage loading coefficient have to fall within acceptable limit.

All the design limitations were taken into consideration to ensure that the new industrial gas turbine becomes feasible and viable. TABLE II presents design limiting parameters

| Parameters | First Stage Turbine | Second Stage Turbine | Third Stage Turbine |
|------------|---------------------|----------------------|---------------------|
| Flow Area (m²) | 0.499 | 0.62 | 0.64 |
| Blade Height (m) | 0.128 | 0.156 | 0.165 |
| Tip Diameter (m) | 1.37 | 1.40 | 1.42 |
| Hub Diameter (m) | 1.11 | 1.07 | 1.07 |
| Hub-to-Tip Ratio | 0.81 | 0.773 | 0.765 |
| Blade Loading Coefficient | 2.15 | 1.93 | 1.73 |
| Degree Reaction | 0.47 | 0.3 | 0.27 |
| Flow Coefficient | 0.8 | 0.52 | 0.47 |

1. Design Assessment
From the algorithms provided by Ramsden [9], an excel spreadsheet was developed from where all the data are altered with ease and decision made accordingly.

2. Viability of the Annulus Diagram
Fig. 4 shows the constant mean diameter of the designed turbine last stage. The limit for an ideal hub-tip ratio should not exceed 0.9 [8]. The inlet and the outlet annulus presented seem viable since there is a divergence in the annulus for a constant mean speed. TABLE III shows the overall stage hub-to-tip ratio, while TABLEs IV and V show the design calculations of the turbine inlet and outlet geometries respectively.

| TABLE III: OVERALL STAGE HUB TO TIP RATIO |
|-------------------------------------------|
| Overall Stage Specification | Data |
| Inlet hub to tip ratio (first Stage) | 0.87 |
| Outlet hub to tip ratio (last Stage) | 0.77 |

Fig. 4. Constant Mean Diameter of the of the Last stage turbine.

Tables VI and VII show the turbine efficiency prediction parameters and turbine free vortex design respectively.

| TABLE IV: TURBINE INLET ANNULUS GEOMETRY |
|------------------------------------------|
| Turbine Inlet Annulus Geometry |
| TET(K) | 1400 |
| Turbine Inlet Pressure | 119314 |
| ΔW= | 1.24 |
| Dmean (m) | 1.32 |
| Dhub | 1.15 |
| H | 0.088 |
| Dhub/Dtip | 0.866 |
| Va (m/s) | 0.469 |
| Vhub(m/s) | 268 |
| Vmean(m/s) | 272.13 |

| TABLE V: TURBINE OUTLET ANNULUS GEOMETRY |
|------------------------------------------|
| Turbine Outlet Annulus Geometry (First Stage) |
| TET(K) | 1400 |
| Turbine Drop = Power/ W.Cp | 261.549 |
| Blade Speed, U= RPM x π x D/60 | 335 |
| ∆H= Cp ∆T.U²/2 | 2.81 |
| V3 (m/s) | 268 |
| Vf/U | 0.8 |
| ηturb(smith chart) | 0.9 |

| TABLE VI: TURBINE EFFICIENCY PREDICTION |
|------------------------------------------|
| Turbine Inlet Annulus Geometry |
| TET(K) | 1400 |
| Turbine Drop | 261.549 |
| Q3 (from chart) | 0.288 |
| Q3 (from chart) | 0.018936 |
| Pressure Ratio, PR | 0.562 |
| P3 = PRx P1 | 671084.5 |
| A3 = W3/√T3/P3Q3 | 0.4 |
| Ain = A3 Cosα3 | 0.505 |

| TABLE VII: TURBINE FREE VORTEX DESIGN |
|------------------------------------------|
| Parameters | ROOT | BMH | TIP |
| D (NGV Exit) | 1.16 | 1.24 | 1.31 |
| D (Rotor Exit) | 1.16 | 1.24 | 1.31 |
| Va (m/s) | 268 | 268 | 268 |
| Vmean (m/s) | 192.2 | 192.2 | 192.2 |
| Vhub(m/s) | 527.2 | 527.2 | 527.2 |
| Vax(m/s) | 563.6 | 563.6 | 498.3 |
| αx(degrees) | 64.6 | 63.0 | 61.7 |
| Vax(m/s) | 205.5 | 192.2 | 181.7 |
| αx(degrees) | 37.5 | 36.6 | 34.1 |
| U(m/s) | 313 | 335 | 354 |
| Va(m/s) | 624 | 591 | 566 |
| αx(degrees) | 27.0 | 25.6 | 24.3 |
| Vax(m/s) | 300 | 297 | 294 |
| V1 (m/s) | 367 | 329 | 294 |
| αx(degrees) | 36.2 | 39.1 | 41.4 |
| V2(m/s) | 584 | 591 | 599 |

3. Number of stages
To satisfy the requirement for the turbine blade loading coefficient of less than 2.5 at the first stage and a flow coefficient of less than 0.8 at all stages, the turbine was split
into three distinct stages with their blade loading coefficient of 2.15, 1.93 and 1.73 for the first, second and third stages respectively.

C. Design Feasibility

With the introduction of more turbine stages, a feasibility analysis was conducted on the new engine design configuration. Having satisfied all the design limiting criteria, the configuration was considered feasible. Certain diagnostic trends were observed which also serve as a precursor in monitoring the design procedure and for adequate decision. The blade speed, \( U \) at the exit of the each stage of the turbine was found to be increasing from the root to the tip. This results from the increasing diameter of the blade away from the root and is as expected since the centrifugal stress on the blade tends to concentrate at the root. The choice of the blade mean speed is restricted to the rotor exit mean diameter of the compressor since they are been driven by a common shaft of equal rotational speed. Also, the nozzle acceleration increases while the rotor acceleration decreases from the hub to the tip. With these trends, the feasibility of the design can be deemed to be satisfactory.

D. Aerodynamic Feasibility

Tables VIII and IX present the turbine design assessment 1 and 2 respectively.

| TABLE VIII: TURBINE DESIGN ASSESSMENT 1 |
|-----------------------------------------|
| Hub-to-Casing | ROOT | BMH | TIP |
|---------------|------|-----|-----|
| \( \alpha_{in} \) | 40.22 | 37.79 | 32.78 |
| NGV Exit Gas Angle, \( \alpha_g \) | 72.49 | 71.0 | 67.47 |
| Nozzle Deflection, \( \alpha_g + \alpha_{in} \) | 112.71 | 108.79 | 100.45 |
| Rotor Deflection, \( \alpha_1 + \alpha_2 \) | 112.06 | 108.86 | 74.16 |
| Nozzle Acceleration, \( \frac{V_2}{V_1} \) | 2.535 | 2.43 | 2.19 |
| Rotor Acceleration, \( \frac{V_2}{V_1} \) | 1.83 | 2.42 | 3.27 |
| Exit Swirl, \( \alpha_3 \) | 41.39 | 37.79 | 33.95 |

1. Rotor swirl angle

The swirl angle \( \alpha_3 \) in all the cases are unacceptable. As is shown in Table VIII, all the swirl angles are more than the maximum allowable value [9]. A low swirl angle becomes important as reduces the risk of jet pie losses. The excessive swirl angle is as result of the high whirl velocity at the rotor exit at the operating axial velocity. An increase in the shaft rotational speed can correct this problem but may subject the component to more stresses. Table IX shows the design assessment 2.

| TABLE IX: TURBINE DESIGN ASSESSMENT 2 |
|---------------------------------------|
| Turbine Design Assessment at Blade Mid Height |
| Parameters | NGV Exit | Blade Exit |
| Static Temperature (K) | 1142 | 1061 |
| Speed of Sound (m/s) | 700 | 722 |
| Absolute Mach Number | 0.48 | 0.46 |
| Axial Mach Number | 0.284 | 0.275 |

2. Rotor deflection angle

A rotor deflection angle of less than 110° is acceptable [8]. As shown in Table VIII, this limit is exceeded at the root of all the stages but not at the tip. The obtained valued can be accepted as this does not constitute any severe problem in this design.

3. Flow coefficient

As earlier discussed, this is one of the key factors controlling the performance of the turbine. The limit acceptable is \( Va/U<0.8 \). For this design, the value of 0.8, 0.52 and 0.47 has been obtained from stages 1, 2 and 3 respectively.

4. NGV Gas angle

As is shown in TABLE VIII, the NGV gas angles at the hub, blade mid height and the tip are all within 70° which is the limit acceptable for this design [9]. With this, the need for cooling mechanism is not actually necessary.

5. Axial Mach numbers

As shown in Table IX, the axial Mach number at the blade exit was 0.275. This is clearly below the design limit of <0.5 thus ensuring a no breakdown of flow at downstream of the turbine.

6. Blade Loading Coefficient

As one of the key factors in the design of the turbine, stage loading coefficient, \( \Delta H/2U \) for this design is acceptable having not exceeded the limit of <2.5 for the first stage and 1.8 for the last stage. The values obtained for this design are 2.15, 1.93 and 1.73 for the first, second and last stages respectively. These agrees with the mean blade diameter of 1.234 been imposed by the compressor exit rotor.

7. Hub to Tip Ratio

For the three stages, the obtained hub/tip ratio as shown in Table III is within the acceptable limit of 0.5<hub/tip<0.9 [9].

8. Reaction

The degree of reaction is essentially the ratio of the temperature drop across the rotor to the total temperature drop across the entire stages of the turbine. The results obtained for the reaction at the rub, blade mid height and the tip are 0.27, 0.5 and 0.61 respectively. These values are acceptable for a preliminary design of this kind. The compressor rotational speed controls the reaction at the turbine.

9. Weight

Low stage loading in this design necessitated the introduction of more stages which would have caused a significant effect on the weight. For the intended application, size in not that of a problem and will be safely ignored.

10. Blade Cooling

The NGV gas angle at the hub, blade mid height and the tip which is within a limit of 70° [9] thus making blade cooling irrelevant for this design. With the increase in blade speed at the rotor, the annulus height is reduced at the constant mass flow confirming the unimportance of blade cooling in this design.
IV. CONCLUSION

This study presents a preliminary design of an axial turbine section in an industrial gas turbine. The design was necessitated following the need to provide a gas turbine of a power output in the range of 48 to 60MW for a mini-city harbouring an oil rig, which was not possible with the old engine. Using proprietary gas turbine performance simulation software called TURBOMATCH and a computer program written in Microsoft Excel, a redesign of the axial turbine component was achieved.

All the factors essential for confirming the feasibility of the prescribed performance with respect to the turbine has been adopted. The design analysis suggest that at least a three stage turbine will be required to conveniently drive the twelve-staged compressor at the rotational speed of 5166 rpm and the exit rotor mean speed of 335m/s. Having carried out the geometrical, free flow vortex and the stage loading analysis, it became apparent that this design will be feasible since these designs controlling factors are within the acceptable limit. The verification of the existing gas turbine for these design constraints proved unacceptable since such factor as the stage loading coefficient was severely exceeded signifying an apparent tip leakage and a possible failure due to high centrifugal stresses on the rotor. Though the introduction of more turbine stages may contribute to an additional weight on the entire engine and perhaps extra cost, it will not pose any serious setback for the intended application since weight is not factors for land base turbines.

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NOMENCLATURE

A Flow Angle
ΔH Stage Work Input
ΔT Temperature Change
Cp Specific Heat at constant Pressure
D Hub Diameter
D, Mean Diameter
D, Tip Diameter
η Efficiency
Wt Fuel Flow
AFT Axial Flow Turbine
BMH Blade Mean Height
CFD Computational Flow Dynamics
DP Design Point
EGT Exhaust Gas Temperature
FPT Free Power Turbine
HP High Pressure
HPT High Pressure Turbine
IGV Inlet Guide Vanes
LP Low Pressure
LPT Low Pressure Turbine
M Mach number
NGV Nozzle Guide Vanes
RPM Revolutions per Minute
TET Turbine Entry Temperature
V Axial Velocity
Vw Whirl Velocity