PERFORMANCE PREDICTION OF A TURBOSHAFT ENGINE THROUGH THE USE OF LOSSES MODELS

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ABSTRACT

The aim of present research is performance simulation of axial flow turbine by using of one-dimensional modelling method. As respects the importance of modelling for performance prediction and optimization in initial designs level, in following work, that is based on one-dimensional modelling method, after the presentation of solution algorithm by trial and error method and introduction of different loss models for modelling, this method were examined for a turbo shaft engine and compared with experimental results. Comparison of the results with experimental data shows so good adaptation. Also according to these results, Kacker and Okapuu's developed model gave the closest results to the reference data.

Keywords

Modelling; Two-stage axial flow turbine; Total loss coefficient; Loss model; Performance curve

1. INTRODUCTION

Gas turbines have an important role in power generation and propulsion units and day to day with technology progress and increase the efficiency and performance, their utilization rises in aerospace and electricity generation plants and Power Generation systems [1].

Due to the very wide scope that exists for gas turbines, must show accurate and reliable performance in the vast range of environment temperature, pressure and also flight Mach and rotational speed.

According to flow pattern complexity in gas turbines, different calculating and modelling methods are required in order to their effective design. Meanwhile, one-dimensional modelling is a simple and fast method due to the performance prediction and optimization of initial design.

2. PRINCIPLES OF ONE DIMENSIONAL ANALYSIS

One-dimensional modelling is a simple, accurate and fast method for obtaining gas turbine performance condition. In this method, the averages of aerodynamic and thermodynamic properties are considered on midstream line [2].

In this method, to simplify the equations and having a faster access to performance characteristics, the following assumptions are applied:

1-The inlet gas is considered as a perfect gas.

- 2-The flow is steady.
- 3-Heat transfer effects are ignored.
- 4-The flow is one-dimensional. Therefore various parameters changes are regardless in the radial and angular direction, and the values at mean radius, are considered as the average values of the whole blade passage.
- 5-Since the viscosity of air changes with temperature, these changes are taken into consideration.

2.1. One Dimensional Modelling by Using of Suggested Algorithm

The suggested algorithm is based on flow field equation in turbine blades.

By applying the continuity equation on stator outlet section and replacement of Mach number relation in it, and by using other base equations, the dimensionless mass flow rate equation will be obtained by considering the losses term as [1,3].

$$\frac{\dot{m} \sqrt{\frac{RT_0}{\gamma}}}{A_{out} P_0} = \sigma \cos(\alpha_{out}) M_{out} \left(1 + \frac{\gamma-1}{2} M_{out}^2\right)^{\frac{\gamma+1}{2(1-\gamma)}} \quad (1)$$

It can be seen that the left hand side of this equation is a function of the mass flow, $\dot{m}$, the outlet flow area, A_{out} , the stagnation temperature, T_0 , and the stagnation pressure, P_0 . And the right hand side is a function of the outlet Mach number, M_{out} , and outlet flow angle, α_{out} . The special symbol σ is a function of entropy change of actual process [4]:

$$\sigma = e^{(-\Delta s / R)} \quad (2)$$

To do the calculations, rather than turbine geometry, amount of turbine rotational speed, stagnation temperature and pressure conditions of input flow to turbine are defined. Calculation will be started from turbine inlet. The solution of the flow equation through the suggested algorithm for modeling is as follows:

First, the proper Mach number is determined in blade cascade inlet. Then by using of continuity equation in turbine inlet, the mass flow rate is calculated. By guessing an initial Mach number and flow angle in blade outlet, losses coefficients and total loss coefficient will be determined by using of loss model. After determination the total loss coefficient, the only unknown parameter in the flow equation is outlet Mach number that is calculated. By calculating of outlet Mach number, the outflow angle will be modified and this repetition continued until intended precision achieved. By determining final outlet Mach number, another parameter like outlet stagnation pressure and temperature will be achieved.

Calculating the flow condition after choke, is one trait of done modeling. In blades, choking phenomenon occurs when Mach number or critical velocity ratio is equal one. For values of Mach number greater than 1.0, the outlet flow angle of the blade row is calculated by using of choking mass flow.

By using of flow equation and placement $M=1$, we reach to Eq. (3) for critical mass flow:

$$\dot{m}_{cr} = \cos(\alpha_{out}) \cdot \sigma \cdot A \cdot P_0 \cdot \left(\frac{1+\gamma}{2}\right)^{\frac{-1}{(\gamma-1)}} \cdot \sqrt{\frac{2\gamma}{RT_0(\gamma+1)}} \quad (3)$$

The critical mass flow that is calculated in Eq. (3), is the choking mass flow, that its value is constant for $M>1$.

$$\dot{m}_{choke} = \dot{m}_{cr} \quad (4)$$

For calculating the outlet flow angle in choking region, the choking mass flow is used [5, 6]:

$$\cos(\alpha) = \frac{\dot{m}_{choke}}{\sigma A \cdot P_0 \cdot M \left(1 + \frac{\gamma-1}{2} M^2\right)^{\frac{1+\gamma}{2(1-\gamma)}} \sqrt{\frac{\gamma}{RT_0}}} \quad (5)$$

After calculation the Mach number, choking mass flow and flow angle, other required quantities are obtained like conditions before choke.

2.2 Loss Coefficients

Different energy losses that usually are considered in turbine, include profile loss, secondary loss, tip leakage loss and annulus loss. These losses in axial flow turbines are expressed in terms of loss coefficients. The loss coefficients manifested by a decrease in the stagnation enthalpy, and a variation in the static pressure and temperature, compared to the isentropic flow [7].

Enthalpy loss coefficient, entropy loss coefficient and pressure loss coefficient are three usual loss coefficients in turbines. We used the pressure loss coefficient and the enthalpy loss coefficient in our modeling, that their equations are presented in this part.

The pressure loss coefficient for stator and rotor blades is defined as [7, 8]:

$$Y_N = \frac{P_{01} - P_{02}}{P_{02} - P_2} \quad \text{Stator (6)}$$

$$Y_R = \frac{P_{02,rel} - P_{03,rel}}{P_{03,rel} - P_3} \quad \text{rotor (7)}$$

The enthalpy loss coefficient for stator and rotor blades is:

$$\zeta_N = \frac{h_2 - h_2'}{h_{01} - h_2} = \frac{T_2 - T_2'}{\frac{1}{2} C_2^2} \quad (8) \text{ stator}$$

$$\zeta_R = \frac{h_3 - h_3''}{h_{02,rel} - h_3} = \frac{T_3 - T_3''}{\frac{1}{2} V_3^2} \quad (9) \text{ rotor}$$

The parameters and subscripts that are used in these equations are based on diagram (Temperature- Entropy) in Fig. 1.

Figure 1. T-S diagram for a reaction stage [8]

There are different loss models for predicting the amount of loss in the axial flow turbines. In this section three models of these losses have been presented.

2.3 Soderberg's model

This model is useful for obtaining quick and preliminary estimates of turbine performance. Soderberg gave the total loss coefficient as:

$$\zeta_N = \left(\frac{10^5}{Re} \right)^{1/4} \left[(1 + \zeta^*) \left(.993 + .075 \frac{l}{H} \right) - 1 \right] \quad (10)$$

$$\zeta_R = \left(\frac{10^5}{Re} \right)^{1/4} \left[(1 + \zeta^*) \left(.975 + .075 \frac{l}{H} \right) - 1 \right] \quad (11)$$

In these equations, H is the blade height, l is the blade actual chord, ζ is the enthalpy loss coefficient and ζ^* , is the nominal loss coefficient and main function of the blade deflection. Soderberg's model only includes profile and secondary flow loss but not tip clearance loss [7].

2.4 Kacker and Okapuu's developed model (Tournier & Genk's model)

This model is the latest refinements proposed by Benner et al. of Kacker and Okapuu's model and Tournier & Genk developed it, that we used in this research.

The total pressure loss coefficient in this model is given as:

$$Y = (Y_p + Y_s) + Y_{TE} + Y_{Tc} \quad (12)$$

They suggested the bottom equations for calculating profile and secondary losses:

$$(Y_p + Y_s) = \left(1 - \frac{Z_{TE}}{H}\right) \times Y_p' + Y_s' \quad (13)$$

Z_{TE} , is the spanwise penetration depth.

The profile loss coefficient, based on recent turbine cascade experimental data, is given by:

$$Y_p' = .914 \times \left[K_{in} Y_{p,AM}' + K_p + Y_{shock} \right] \times K_{Re} \quad (14)$$

K_{in} in Eq. (14), has a constant value and represented by different people. Also, K_{Re} , is the Reynolds number correction factor.

The factors K_p and Y_{shock} in Eq. (14) is identical to that introduced by Kacker and Okapuu [9, 10] to account for the gas compressibility. $Y_{p,AM}$ is the same profile loss that presented by Ainley & Mathieson, and its equation was mentioned in previous part.

In Eq. (13), the spanwise penetration depth (Z_{TE}) of the separation line between the primary and the secondary regions, is calculated by:

$$\frac{Z_{TE}}{H} = \frac{.10 \times |F_t|^{.79}}{\sqrt{\cos \alpha_{in} / \cos \alpha_{out}} \times (H/l)^{.55}} + 32.7 \left(\frac{\delta^*}{H} \right)^2 \quad (15)$$

δ^* , is the boundary layer displacement thickness at the inlet end wall.

In Eq. (15), the tangential loading parameter, F_t , is given by:

$$F_t = 2 \frac{s}{l \times \cos \phi} \times \cos^2(\alpha_m) \times (\tan \alpha_{in} + \tan \alpha_{out}) \quad (16)$$

In Eq. (20), ϕ is the blades stagger angle.

The secondary loss coefficient in Eq. (13), is given by:

$$Y_s' = F_{AR} \times \frac{.038 + .41 \times \tanh(1.2\delta^*/H)}{\sqrt{\cos \phi} \times (\cos \alpha_{in} / \cos \alpha_{out}) \times (l \cos \alpha_{out} / l_x)^{.55}} \quad (17)$$

In this equation, l_x , is the blade chord length in rotating axial direction.

The aspect ratio factor, F_{AR} , is a function of the blade aspect ratio (Chord/Height).

In this model trailing edge loss coefficient, Y_{TE} , is a function of outlet Mach number and kinetic energy loss coefficient in trailing edge that their equations are presented by Kacker & Okapuu [9].

Also the tip leakage loss coefficient, Y_{tl} , is calculated by using the approach of Yaras and Sjolander

The Abstract section begins with the word, “Abstract” in 13 pt. Times New Roman, bold italics, “Small Caps” font with a 6pt. spacing following. The abstract must not exceed 150 words in length in 10 pt. Times New Roman italics. The text must be fully justified, with a 12 pt. paragraph spacing following the last line.

3. RESULTS AND DISCUSSION

In order to have a better understanding of a two stage axial flow turbine, behaviour and verify the suggested algorithm accuracy, modelling has been done on this engine by using of two different loss model. Experimental data have been obtained from Ref. [12]. Also turbine blades geometry has been measured by authors.

Figs. 2 and 3 show variation of the turbine mass flow versus pressure ratio at design speed. The design speed is 38100 rpm for compressor turbine (First Stage turbine) and 33000 rpm for power turbine (Second Stage turbine). The conditions are the same as the experiments. The comparison of achieved results of modelling with experimental results shows that the theoretical values agree with the experimental values and good adaptation exist between these results.

Figure 2. Mass flow vs. Pressure ratio by using of Kacker and Okapuu's developed model

Figure 3. Mass flow vs. Pressure ratio by using of soderberg's model

In Figs. 4 and 5 efficiency curves of modelling were brought in design speed. Efficiency rises as increase of pressure ratio until it reaches its maximum measure. The reason for these changes is that in special cases, the incidence angle and energy losses due to it, reach its minimum value, so that in this condition the efficiency will maximize, and after this condition, the losses will increase again.

Figure 4. Turbine efficiency vs. Pressure ratio by Kacker and Okapuu's developed loss model

Figure 5. Turbine efficiency vs. Pressure ratio by Soderberg's loss model

In Figs. 6 and 7 the losses predictions have been shown. Fig. 6 is related to power turbine rotor blade and Fig. 7 is related to power turbine stator one.

In these Figures distribution of the profile loss, secondary loss, tip leakage loss, trailing edge loss, shock wave and total loss coefficient are presented. When the pressure ratio rises, the profile loss and total loss coefficient decrease and reach to a minimum point, and from there begin to increase again. But the Secondary loss, tip leakage loss and trailing edge loss almost exhibit linear behaviour. Also this behaviour reported by Ning Wei [7].

The profile loss has greatest value among another losses coefficient. The value of this loss, which is calculated with Eq. (14), give the lowest value near the pressure ratio 2.54 that this pressure ratio is related to about zero incidence.

Figure 6. Loss coefficients vs. Pressure ratio

Figure 7. Loss coefficients vs. Pressure ratio

In Fig. 8 Enthalpy loss coefficient is shown for rotor and stator of power turbine. As can be seen the measure of this loss for stator blade is greater than rotor blade. This is because, according to Eqs. (10) and (11) the enthalpy loss is a main function of the blade deflection.

Figure 8. Loss coefficients vs. Pressure ratio

In Table 1 a comparison between outlet pressure and temperature from turbine blades and experimental data are presented.

Table 1. Comparison between outlet pressure and temperature in theoretical and experimental Results

	First Stage at 38100 rpm		Second Stage at 33000 rpm	
	Pout(Kpa)	Tout(k)	Pout(Kpa)	Tout(k)
Experimental Data	282.6	1058.15	111.00	901.15
soderberg's loss model	302.8	1077	171.2	924.7
Kacker and Okapuu's developed loss model	278	1070	124	866

4. CONCLUDING REMARKS

In this paper, one-dimensional modelling was done by using of suggested algorithm and different loss models on a typical two stage axial flow turbine, in order to validate the suggested method and loss models. According to the results of modelling, it is clear that this modelling and suggested algorithm, predict the turbine performances acceptably.

Comparing the results of these models show that Kacker and Okapuu's developed model gave the closest results to the reference data because this model estimated the loss coefficients with greater accuracy.

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