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Analytical Modeling of Performance Characteristics of Axial Flow Two-Stage Turbine Engine Using Pressure Losses Models and Comparing with Experimental Results

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Abstract: The main objective of this paper is to make a detailed systematic analysis of two-stage, axial flow turbine by using of different losses models and a new suggested algorithm based on one-dimensional simulation. The suggested method is found to be effective, fast and stable, in obtaining performance characteristics of multi-stage axial flow turbines. In one-dimensional modeling, mass flow rate, pressure ratio and efficiency are unknown, with define turbine geometry, inlet total pressure and temperature the turbine performance characteristics can be modeled. This modeling is based on common thermodynamics and aerodynamics principles in a mean stream line analysis under steady state condition. Finally, in order to have a better understanding of the loss models behavior and verify the suggested algorithm accuracy, the analytical results from modeling were compared with experimental results and the comparison shows that good adaptation is obtained.

Key words: Axial flow turbine · Total loss coefficient · Design point · Experimental results

INTRODUCTION

Gas turbines have an important role in power generation and propulsion units [1]. The operation of a gas turbine depends on the characteristics of its major components such as the compressor, turbine and combustor [2]. Among these, the turbine is known as one main components of the gas turbine. The fundamental idea with a turbine is to extract work from the incoming airflow and convert it into mechanical work at a rotating axis.

The flow pattern through the turbine is fully three-dimentional and complex and not yet fully understood, so experimental method is best way to investigate the machine performance but it takes time and cost. Thus the analytical modeling is a suitable method for predicting the performance in design and matching procedure [3].

Turbine flow performance can be predicted using mean stream line analysis that identifies the losses occurring along a meanline flow path through the turbine. Futral and Wasserbauer [4], Abidat *et al* [5] and Mamat and Martinez-Botas [6], showed the capability of mean streamline modeling to predict the steady flow performance of different turbines. Recently, more research has started to focus on analyzing the flow performance of axial flow turbines.

During the recent years, one-dimensional modeling technique is utilized by a number of researchers. Ning WEI [7], studied the significance of loss models and their applications in simulation and optimization of axial turbines. He presented useful guides for applying the models properly in turbines aerothermodynamic simulation and optimization. Dahlquist [8], described the physical flow phenomena in a blade row that creates losses in an axial gas turbine and extracted the correlation to estimate these losses in a mean line calculation. Abed [9], presented an algorithm for one-dimensional modeling that was considered as a main algorithm in many studies after it. Tournier & Genk [10], used one-dimensional modeling that was based on a mean-line flow analysis for

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performance prediction of axial flow turbines. They developed the latest refinements proposed by Benner *et al*. of Kacker and Okapuu's model.

This paper describes a method of prediction the performance of two-stage axial flow turbine at both the design and off-design conditions. This model is based on one-dimensional performance prediction.

Flow Field and Loss Mechanisms in Axial Turbine Blades: The flow in turbine blades is characterized by a three dimensional, highly unsteady motion with random fluctuations due to the interactions between the stator and rotor rows [1]. Because of this, different losses are created in turbine cascade.

Profile loss is a main loss that is created due to blades boundary layers or wake which will take place with a uniform two-dimensional flow across a cascade of blades [11]. There is a primary flow field through the blade row which describes the mean path of the flow. Overlapping on this primary flow will produce a secondary flow field [8].

A leakage flow across the tip clearance at a blade disturbs the primary flow. This flow is highly dependent upon the size of the tip gap and strongly influences the other end wall losses. Typically, this flow is ejected as a strong jet which mixes with the main stream on the suction side, usually rolling up to form a vortex. This strong jet and vortex cause entropy change. [12, 13].

Annulus loss that is created with boundary layer growth on the inner and outer walls of the annulus, is a part of secondary loss [11].

Principles of One-Dimensional Analysis in Turbine: One-dimensional modeling is an accurate and fast method for obtaining gas turbine performance condition. In this method, the mean flow parameters are solved along a mean str eamline on key stations (inlet and the exit of each section) [14, 15].

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

- The inlet gas is considered as a perfect gas.
- The flow is steady.
- Heat transfer effects are ignored.
- 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.
- Since the viscosity of air changes with temperature, these changes are taken into consideration .

Modeling: The governing equations of one-dimensional isentropic flow along a channel include continuity, energy survival and perfect gas relations. By using of Mach number equation and above relations, the flow field equation will be obtained by considering the losses term as [3, 14, 2]:

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

The suggested algorithm is based on this equation in turbine blades. The special symbol, ó, is the entropy production function and a function of entropy change of actual process:

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

For the calculation of this parameter, ó, that is called total pressure loss coefficient, we used equation (3) and equation (4) that follow, immediately, from its definition (Equation (2)) [3]:

$$\sigma = \left\{1+Y\left[1-(1+\frac{\gamma-1}{2}M_{out}^2)^{\frac{\gamma}{1-\gamma}}\right]\right\}^{-1} \qquad (3)$$

$$\sigma = (1-\frac{\gamma-1}{2}\zeta M_{out}{}^2)^{\gamma/\gamma-1} \qquad (4)$$

The method for flow equation solution by using the proposed algorithm is as follows:

First, the turbine geometry and the gas property and stagnation temperature, pressure and Mach number at the entrance of the blade row put as known parameter. Then the mass flow is calculated by using of continuity equation. And by guessing an initial Mach number and flow angle in the blade outlet, losses coefficients and total loss coefficient will be determined and outlet Mach number calculated again from flow equation. By calculating of Mach number, the outflow angle will be modified and this repetition continued until intended precision achieved. This algorithm is shown in Fig. 1. After determining the final outlet Mach number, other quantities like outlet stagnation pressure and temperature will be achieved.

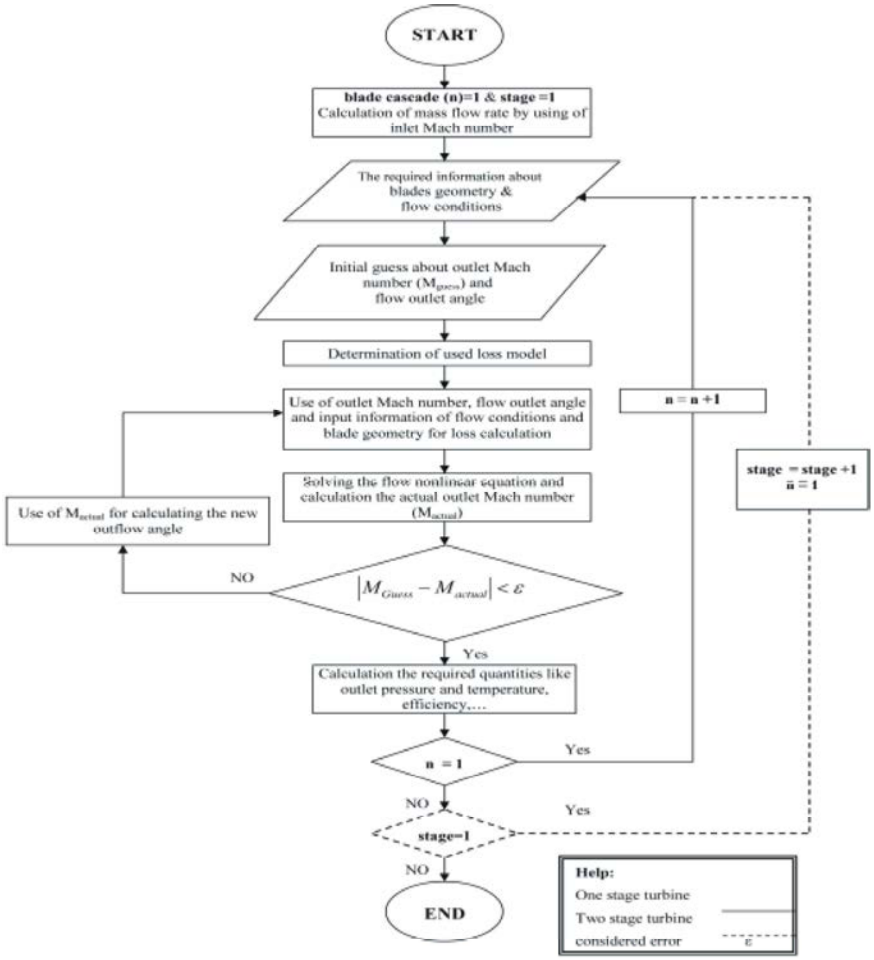

Fig. 1. Suggested algorithm for turbine 1D modeling

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

By using of flow equation and replacement of M=1, we reach equation (5) for critical mass flow:

$$\dot{m}_{cr} = \cos(\alpha_{out}).\sigma.A\ .P_0.\left(\frac{1+\gamma}{2}\right)^{\frac{-1}{(\gamma-1)}}.\sqrt{\frac{2\gamma}{RT_0(\gamma+1)}} \tag{5}$$

The critical mass flow that is calculated in equation (5), is the choking mass flow whose value is constant for M>1.

$$\dot{m}_{choke} = \dot{m}_{cr} \tag{6}$$

For calculating the outlet flow angle in choking region, the choking mass flow is used [16-18]:

$$\cos(\alpha\ ) = \frac{\dot{m}_{choke}}{\sigma.A\ .P_0.M\ \left(1+\frac{\gamma-1}{2}M^{\ 2}\right)^{\frac{1+\gamma}{2(1-\gamma)}}\sqrt{\frac{\gamma}{RT_0}}} \tag{7}$$

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

Losses: 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 [7, 11].

In one-dimensional modeling different kind of losses are considered and for successful modeling, understanding the cause of these losses is essential. In this section four models of these losses have been presented as follow :

Soderberg's Loss Model: This model is useful for obtaining quick and preliminary estimates of turbine performance.

Soderberg gave the total loss coefficient as:

$$\zeta_N = \left(\frac{10^5}{\text{Re}}\right)^{1/4}\left[\left(1+\xi^*\right)\left(.993+.075\frac{l}{H}\right)-1\right] \tag{8}$$

$$\zeta_R = \left(\frac{10^5}{\text{Re}}\right)^{1/4}\left[\left(1+\xi^*\right)\left(.975+.075\frac{l}{H}\right)-1\right] \tag{9}$$

ζ^* is the nominal loss coefficient. The profile loss in this model depends on the ζ^*, that is a function of the blade deflection. Also the secondary loss was considered as a function of aspect ratio*, l/H* [7].

Soderberg's model only includes profile and secondary flow loss but not tip clearance loss. The neglect of tip clearance loss, inlet boundary layer and the most of blade geometry are the greatest infirmity of this model.

Ainley and Mathieson's Loss Model: This model is the most comprehensive model for simulation and based on assumptions and experimental data that can be used to predict the performance of axial flow turbines with conventional blades over a wide part of their full operating range.

The total losses coefficient in a turbine cascade by Ainley & Mathieson [7] consists of profile loss, secondary loss and tip leakage loss.

$$Y = (Y_p + Y_s + Y_{TI})X_{TE} \tag{10}$$

In this equation, X_{TE} is the trailing edge correction factor. This variable parameter can be obtained from the figure that given by Ainley & Mathieson [7].

Ainley & Mathieson gave profile loss model based on a series of experimental graphs of the total pressure losses versus pitch/chord ratio for nozzle and impulse blades.

$$Y_{p(i=0)} = \left\{ \begin{array}{l} Y_{p(\alpha'_{in}=0)} + \left(\dfrac{\alpha'_{in}}{\alpha_{out}}\right)^2 \\ \left[Y_{p(\alpha'_{in}=\alpha_{out})} - Y_{p(\alpha'_{in}=0)}\right] \end{array} \right\} \times \left(\frac{t'_{\max}/l}{.2}\right)^{\alpha'_{in}/\alpha_{out}} \tag{11}$$

In this equation the value of $Y_{P(i=0)}$ refer to blades operating at zero incidence.

The final profile loss is equal to the profile loss at zero incidence, $Y_{P(i=0)}$, multiplied by an incidence coefficient, x_i.

$$Y_p = x_i .\ Y_{p(i=0)} \tag{12}$$

x_i can be obtained from figures that was presented by Ainley & Mathieson [7].

The secondary loss coefficient in this model is calculated based on the blade loading which is considered as a main function of the blade turning. This loss equation is:

$$Y_s = \lambda\left(\frac{c_l}{s/l}\right)^2\left(\frac{\cos^2\alpha_{out}}{\cos^3\alpha_m}\right) \tag{13}$$

In this equation, ë is a parameter that is a function of the flow acceleration through the blade row and given in a figure by Ainley & Mathieson [7].

The flow mean angle, α_m, is given by:

$$\tan(\alpha_m) = \frac{1}{2}\left(\tan\alpha_{in} - \tan\alpha_{out}\right) \tag{14}$$

The tip leakage loss is also considered, with the same principle as the secondary loss, as a function of the blade loading supplemented with the ratio of tip clearance to the blade height. It can be calculated with bottom equation:

$$Y_{TL} = B\frac{\tau}{h}4\left(\tan\alpha_{in} - \tan\alpha_{out}\right)^2\left(\frac{\cos^2\alpha_{out}}{\cos\alpha_m}\right) \tag{15}$$

In equation (15), the flow absolute angle is used for stator blades and relative angle is used for rotor blades. The constant *B* is 0.25 for a shrouded blade and 0.5 for an unshrouded blade.

One defect in this model is that the effects of incidence variation was not considered by Ainley & Mathieson in the tip leakage loss [7, 19].

Came and Dunham's Loss Model: In this model, the total loss is based on the Ainley & Mathieson's loss model and computed by considering of the influence of Reynolds number on the profile and secondary loss.

$$Y = [(Y_p + Y_s)\left(\frac{\text{Re}}{2\times10^5}\right)^{-.2} + Y_{TI}]X_{TE} \tag{16}$$

They developed the profile loss model from Ainley and Mathieson by taking the factor of Mach number into account.

$$Y_p = \left[1 + 60\left(M_{out} - 1\right)^2\right] x_i Y_{p(i=0)} \tag{17}$$

x_i and $Y_{P(i=0)}$ are the same as the Ainley and Mathieson's loss model.

Dunham and Came found that the Ainley and Mathieson secondary loss model was not correct for blade of low aspect ratio, as in small turbines. They modified the Ainley & Mathieson model to include a better correlation with aspect ratio and also simplified the flow acceleration parameter, ë. They presented the secondary loss as:

$$Y_s = .0334 \left(\frac{l}{H}\right) \left[4\left(\tan\alpha_{in} - \tan\alpha_{out}\right)^2\right] \left(\frac{\cos^2\alpha_{out}}{\cos\alpha_m}\right)\left(\frac{\cos\alpha_{out}}{\cos\alpha_{in}^{'}}\right) \tag{18}$$

The tip leakage loss given in this model is based on Ainley and Mathieson's model, but Came and Dunham calculate this loss coefficient as the power function of the tip clearance instead of the linear function in Ainley and Mathieson's loss model.

$$Y_{TL} = B\frac{l}{h}\left(\frac{\tau}{l}\right)^{.78} 4\left(\tan\alpha_{in} - \tan\alpha_{out}\right)^2 \left(\frac{\cos^2\alpha_{out}}{\cos\alpha_m}\right) \tag{19}$$

In equation (19), The constant B is 0.37 for a shrouded blade and 0.47 for an unshrouded blade [7, 11].

Kacker and Okapuu's Developed Loss Model: This model is the latest refinements proposed by Benner *et al*. [20, 21] of Kacker and Okapuu's model [22].

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

$$Y = (Y_p + Y_s) + Y_{TE} + Y_{TL} \tag{20}$$

Benner *et al*. suggested a loss scheme for the breakdown of the profile and secondary losses as:

$$(Y_p + Y_s) = \left(1 - Z_{TE}/H\right) \times Y_p^{'} + Y_s^{'} \tag{21}$$

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

$$Y_p^{'} = .914 \times \left[K_{in}\ Y_{p,AM}^{'}\ K_p + Y_{shock}\right] \times K_{\mathrm{Re}} \tag{22}$$

K_{in}, in equation (22), has a constant value and represented by different people. Also, K_{Re}, is the Reynolds number correction factor [10].

The factors K_p and Y_{shock} in equation (22) is identical to that introduced by Kacker and Okapuu [22] to account for the gas compressibility.

Y_{shock} is the loss coefficient concerning the loss caused by shocks. It is calculated as [7]:

$$Y_{shock} = .75\left(M_{in,hub} - 0.4\right)^{1.75}\left(\frac{r_H}{r_T}\right)\left(\frac{\rho_{in}}{\rho_{out}}\right) \frac{1 - \left(1 + \frac{\gamma - 1}{2} M_{in}^2\right)^{\gamma/\gamma - 1}}{1 - \left(1 + \frac{\gamma - 1}{2} M_{out}^2\right)^{\gamma/\gamma - 1}} \tag{23}$$

$Y^{'}_{p,AM}$ is the same profile loss that presented by Ainley & Mathieson.

In equation [21], the spanwise penetration depth (Z_{TE}) of the separation line between the primary and the secondary regions, is calculated by [20]:

$$\frac{Z_{TE}}{H} = \frac{.10 \times \left|F_t\right|^{.79}}{\sqrt{\cos\alpha_{in} / \cos\alpha_{out}} \times \left(H / l\right)^{.55}} + 32.7\left(\frac{\delta^*}{H}\right)^2 \tag{24}$$

In equation (24), the tangential loading parameter, F_t, is given by:

$$F_t = 2\frac{s}{l \times \cos\phi} \times \cos^2(\alpha_m) \times \left(\tan\alpha_{in} + \tan\alpha_{out}\right) \tag{25}$$

The secondary loss coefficient in equation (21), is given by [8, 21]:

(a) For $H/l \leq 2/0$

$$Y_s^{'} = \frac{.038 + .41 \times \tanh\ (1.2\delta^* / H)}{\sqrt{\cos\phi} \times (\cos\alpha_{in} / \cos\alpha_{out}) \times (\frac{H}{l})^{.55} \times (l\cos\alpha_{out} / l_x)^{.55}} \tag{26a}$$

(b) For $H/l > 2/0$

$$Y_s^{'} = \frac{.052 + .56 \times \tanh\ (1.2\delta^* / H)}{\sqrt{\cos\phi} \times (\cos\alpha_{in} / \cos\alpha_{out}) \times \frac{H}{l} \times (l\cos\alpha_{out} / l_x)^{.55}} \tag{26b}$$

In this model, the trailing edge loss coefficient, Y_{TE}, is a function of the outlet Mach number and kinetic energy loss coefficient in the trailing edge, whose equation is presented by Kacker & Okapuu as:

$$Y_{TE} = \frac{\left\{1-\frac{\gamma-1}{2}M_{out}^2 \times \left(\frac{1}{1-\Delta\phi_{TE}}-1\right)\right\}^{-\gamma/\gamma-1}-1}{1-\left(1+\frac{\gamma-1}{2}M_{out}^2\right)^{-\gamma/\gamma-1}} \quad (27)$$

The coefficient for the trailing edge kinetic energy loss is presented as [7]:

$$\Delta\Phi_{TE} = \Delta\Phi_{TE}^{\alpha'_{in}=0} + \left|\frac{\alpha'_{in}}{\alpha_{out}}\right|\left(\frac{\alpha'_{in}}{\alpha_{out}}\right)\left[\Delta\Phi_{TE}^{(\alpha'_{in}=\alpha_{out})} - \Delta\Phi_{TE}^{(\alpha'_{in}=0)}\right] \quad (28)$$

(a) For an axial entry nozzle:

$$\Delta\Phi_{TE}^{(\alpha'_{in}=0)} = .59563\times\left(\frac{t_{TE}}{o}\right)^2 + .12264\times\left(\frac{t_{TE}}{o}\right) - 2.2796\times10^{-3} \quad (29a)$$

(b) For an impulse blading :

$$\Delta\Phi_{TE}^{(\alpha'_{in}=\alpha_{out})} = .31066\times\left(\frac{t_{TE}}{o}\right)^2 + .065617\times\left(\frac{t_{TE}}{o}\right) - 1.4318\times10^{-3} \quad (29b)$$

The tip leakage loss coefficient, Y_{TL}, is calculated by using the approach of Yaras and Sjolander [10] as:

$$Y_{TL} = Y_{tip} + Y_{gap} \quad (30)$$

$$Y_{tip} = 1.4\times K_E \times \frac{l}{s} \times \frac{\tau}{H} \times \frac{\cos^2\alpha_{out}}{\cos^3\alpha_m} \times C_L^{1.5} \quad (31)$$

$$Y_{gap} = 0.0049\times k_G \times \frac{l}{s} \times \frac{l}{H} \times \frac{\sqrt{C_L}}{\cos\ \alpha_m} \quad (32)$$

Where K_E is a quite insensitive constant that take care to the load distribution of the blade and its amount is [8]:

$$K_E = \begin{cases} .5 & \text{If mid-loaded blade} \\ .566 & \text{If front-or aft loaded blade} \end{cases} \quad (33)$$

And also K_G is:

$$K_G = \begin{cases} 1.0 & \text{If mid-loaded blade} \\ .943 & \text{If front-or aft loaded blade} \end{cases} \quad (34)$$

Turbine Characteristics: The performance of a two-stage turbine that is described in this section has been simulated by different loss models. The main input data for calculations are the stage inlet stagnation pressure and temperature, mass flow, turbine speed and geometric parameters of each cascade. The turbine geometry, experimental data and flow conditions are obtained from a NASA report [23].

Results and Analysis of Turbine Performance Simulation with Different Loss Models: Figs.2, 3, 4 and 5 show variation of the turbine mass flow versus pressure ratio at the design speed (5041 rpm) that were compared with the experimental data. The conditions are the same as the experiments.

The pressure ratios range is 1/17 to 3/8. This range of pressure ratio covers the large off-design region.

The comparison of achieved results of modeling with experimental results shows that the theoretical values agree well with the experimental values and a very good adaptation exists between these results.

In off-design points at lower pressure ratios, the theoretical curves tend to deviate from experimental data. Because flow phenomenas are far from being fully understood. Therefore, there is a risk that not all the most important parameters will be included in the today's used loss models. This can explain why experiments and measurements do not always show the same trends and it is difficult to find any general correlation for 1D modeling at off-design loads. And also because of the design point importance in done modeling, the initial guess for outflow angle is chosen near the design outflow angle. This issue also intensified the deviation at lower pressure ratios.

In Table 1, the percent errors of pressure ratio in modeling and its experimental value, in design point with rotational speed 5041 rpm and mass flow 19.95 kg/sec, are presented. Also a comparison between some off-design points is shown in Table 2.

Soderberg performance curve have the greatest error in respect to experimental curve in design and off design points. This is because of soderberg's model estimates the loss processes lower than actual measures.

Whereas the Kacker & Okapuu's developed model has exhibited the best result, in Fig. 6 performance curves of two stage axial flow turbine have been shown in four rotating speed by using of this model.

By increase of pressure ratio, the mass flow rate also rises, but in a special pressure ratio, this increase is stoped and mass flow remains constant; that can be seen in Fig. 6. In this situation, in a section of blade, Mach number or critical velocity ratio is equal to one.

In Fig. 7, efficiency curves of modeling are brought for different speeds. These curves are also based on Kacker and Okapuu's developed loss model. In each

Table 1: Percent error of pressure ratio toward experimental data

	Soderberg	Came &Dunham	Ainley & Mathieson	Kacker & Okapuu's developed model
Percent of error	%9	%6	%1.6	%0.1

Table 2. percent error of pressure ratio toward experimental data in off design points

	Loss Model			
Point	Soderberg	Came & Dunham	Ainley & Mathieson	Kacker & Okapuu's developed model
1	%12.9	%10.3	%11.5	%11.2
2	%7.1	%-./6	%5	%4.6
3	%6	%-9	%.5	%1
4	%15.4	%-8	%10	%3
Choking mass flow difference (Kg/Sec)	.44	.07	.2	.15

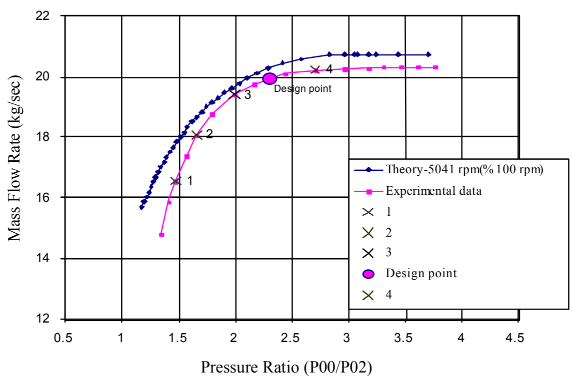

Fig. 2: Mass flow vs. Pressure ratio at design speed by using of soderberg's model

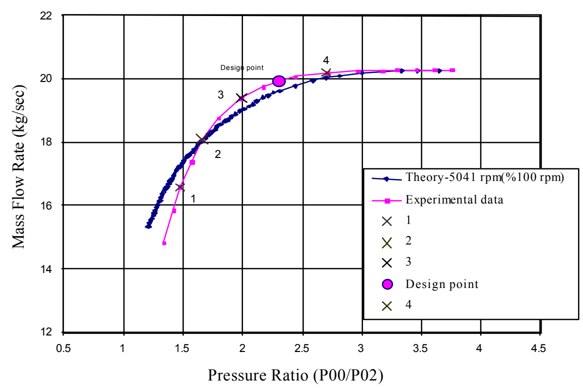

Fig. 4: Mass flow vs Pressure ratio at design speed by using of Came & Dunham's model

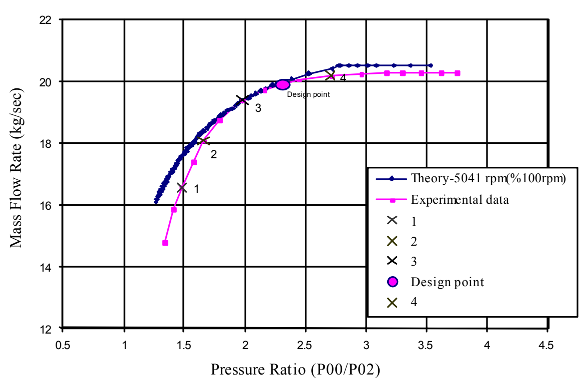

Fig. 3: Mass flow vs. Pressure ratio at design speed by using of Ainley & Mathieson's model

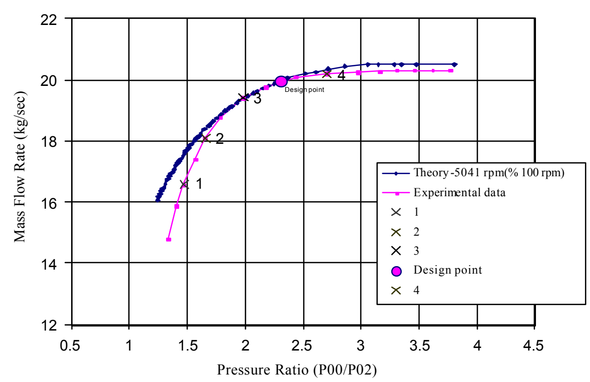

Fig. 5: Mass flow v s. Pressure ratio at design speed by using of Kacker and Okapuu's developed model

rotational 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 efficiency will decrease again. Since the efficiency is usually defined as the ratio of the actual work output to the isentropic work output, only rises in entropy or losses can reduce the efficiency.

A comparison between predicted efficiencies and experimental data in 5041 rpm and 4030 rpm is shown in Figs. 8 and 9. Efficiency prediction by using of Soderberg and Kacker & Okapuu's developed model has better conformity with experimental data.

It can be seen from this figures that the efficiency values predicted in the Kacker & Okapuu's developed model a gree well with the experimental

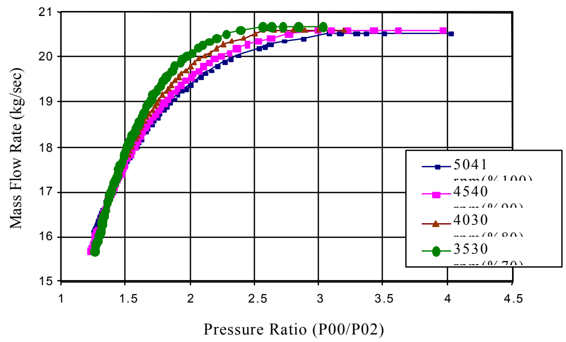

Fig. 6: Mass flow vs. Pressure ratio at different rpm

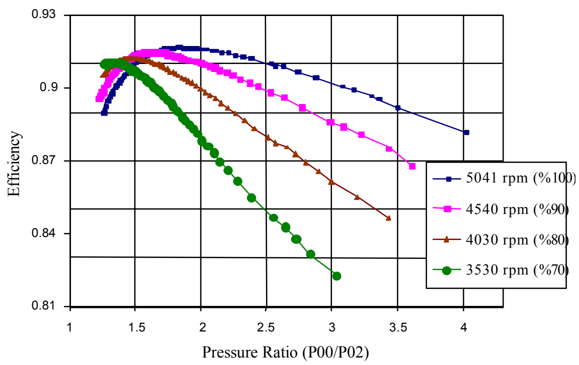

Fig. 7: Turbine efficiency vs. Pressure ratio at different rpm

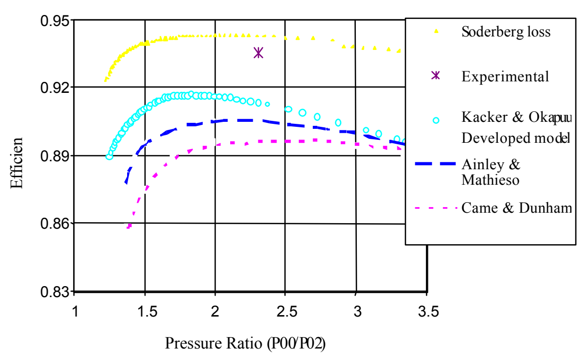

Fig. 8: Comparison between predicted efficiencies and experimental data at 5041 rpm

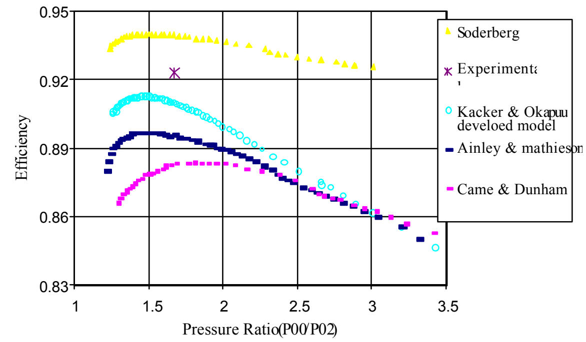

Fig. 9: Comparison between predicted efficiencies and experimental data at 4030 rpm

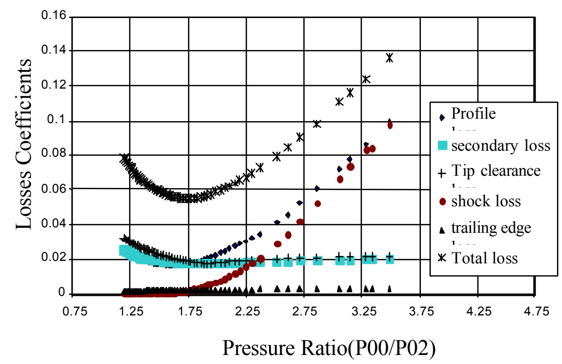

Fig. 10: Loss coefficients vs. Pressure ratio

value in the design point, about %2.1 unit smaller than the experimental result in 5041 rpm and %1.3 unit in 4030 rpm.

Soderberg's model underestimated the losses therefore the value of predicted efficiency by using of this model is about %1 unit greater than the reference data in 5041 rpm and %1.6 unit in 4030 rpm.

In Fig. 10 the losses prediction by using of Kacker & Okapuu's developed model, over the second stage rotor of turbine and design speed, have been shown. This figure shows five different components of losses.

The profile loss has greatest value among another losses coefficient. The value of this loss, which is calculated with equation (22), give the lowest value near the pressure ratio 1.75 that this pressure ratio is related to about zero incidence.

In high pressure ratio that related to high incidence angle and large absolute value of the ratio of the flow inlet to outlet angles, which imply the high turning of the blade shape, will easily induce flow separation on the blade surfaces and therefore produce high off-design profile loss.

Another important loss is shown in Fig. 10 is the secondary loss. This loss is calculated by equations (26a) and (26b). This loss is correlated to the blade loading, which is in terms of flow inlet and outlet angles, as well as the blade aspect ratio. From the predicted results in Fig. 10, the secondary loss rises with the increase of pressure ratio. Because according to increase of pressure ratio, incidence angle also rises and comes to positive range, whereupon the difference between flow inlet and outlet angles becomes large, flow has high turning and blade loading increases. But the secondary loss increase isn't salient among another loss in the researched turbine geometry.

The secondary loss, tip clearance loss and trailing edge loss almost exhibit linear behavior. Also this behavior reported by Ning Wei [7].

The shock loss is a component of the profile loss that is calculated by equation [23]. In this two stages turbine, this loss influence is observed in pressure ratios grater than 1/9. Entropy is generated by shock waves in the flow field at high Mach numbers. The shock waves occurs at the highly curved leading edges. This is normally the smallest loss component [10, 13].

CONCLUSIONS

In this paper one dimensional flow model is presented for performance prediction of two-stage axial flow turbine. Steady flow model, based on loss correlation is used to predict turbine performance. Results of this model are compared with experimental data, which are in reasonable agreement.

According to the modeling results, it is clear that this modeling and suggested algorithm for solving the flow equation, predict the turbine performances acceptably at both the design and off-design conditions, but the results in design point have greater accuracy.

Also, it was found that all these losses models give the same trend of overall performance compared with the trend of experimental results on the turbine stages. The Kacker & Okapuu's developed model give close results to the reference data because this model estimated the loss coefficients with greater accuracy, especially, the profile loss coefficient that is the main loss component in the researched turbine geometry.

Nomenclature:

A = Cross-sectional flow area (m2)
C = Gas absolute velocity vector (m)
C_L = Blades lift coefficient
H = Blade height
h = Blade annulus height (m)
l = Actual chord length of blade (m)
M = Gas Mach number
m = Mass flow rate (kg/s)
O = Throat width between blades in cascade (m)
P = Pressure (Pa)
Re = Gas constant (kj/kg.k)
r = Reynolds number
s = Radius (m)
T = Pitch or distance between blades in cascade (m)
t = Temperature (K)
t_{max} = Blade thickness (m)
Y = Blade maximum thickness (m) pressure loss coefficient
Z_{TE} = Spanwise penetration depth between primary and secondary loss regions (m)

Greek symbols:

α = Angle between C and meridional plane (°)
α' = Blade angle relative to meridional plane (°)
γ = Ratio of specific heat capacities
δ^* = Boundary layer displacement thickness (m)
$\Delta\phi$ = Kinetic energy loss coefficient
ζ = Enthalpy loss coefficient
ρ = Density (kg/m3)
ϕ = Blades stagger angle measured from axial direction (°)
τ = Blades clearance gap (m)

Subscript:

0 = Stagnation parameter
H = Hub of blade
in = Inlet
out = Outlet
p = Profile losses
rel = Relative parameter
s = Secondary losses
T = Tip of blade
TE = Trailing edge of blades
x = Rotating axial direction

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