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DEVELOPMENT OF A NUMERICAL TOOL FOR PRELIMINARY SIZING OF AXIAL TURBINES

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Abstract. *The turbine designer uses the data from engine cycle analysis and based on these engine requirements the designer starts the preliminary turbomachine sizing. For turbomachinery design process the Euler equations are applied with the addition of loss models to include the internal machine losses related with the fluid along the turbine. In the turbine preliminary design it is possible to determine the main operational characteristics and dimensions as power, diameters, efficiency, blade and flow angles, rotational speed, blade dimensions and so on. In this work, a numerical tool written in FORTRAN 90 was developed to calculate the turbine preliminary sizing using the Kacker-Okapuu loss model. A single-stage axial turbine was designed and the results were compared with a commercial turbomachinery design software at design point. The results obtained showed good are in agreement with the commercial software when a comparison of the main loss coefficients as tip, profile, secondary and trailing edge losses are made. The numerical implementation issues and the calculation procedure are presented and discussed.*

Keywords: *Axial Turbine, Turbomachines, Simulation, Preliminary Design.*

1. INTRODUCTION

The internal losses determination during an axial turbine design procedure is crucial to obtain the required performance. Each loss source should be analyzed by the designer to improve the machine operational characteristics and sizing.

The choice of a good loss model in the turbine conceptual design, makes the results obtained with calculation process more realistic when compared with test data. These loss model are based on several turbine test data, including data from cascade tests that are 2D results from airfoil tests, in which total pressure loss are determined.

For many years several authors as Ainley and Mathieson (1957); Traupel (1966); Craig and Cox (1970) developed empirical correlations from test results to better understanding the flow losses mechanisms within turbomachines. Ainley and Mathieson (1957) developed their models through many correlations through turbine test data. The loss sources consist of profile, secondary and tip leakage losses. As all modeling, there are some constraints for: Mach number, Reynolds number, aspect ratio (blade height to chord ratio), trailing edge thickness and blade chord. This model is valid at the design and off design point from axial flow turbines, for design and performance prediction.

The Dunham and Came model (1970) was based on the Ainley and Mathieson correlations (1957). The authors revised some parameters which the Ainley and Mathieson (1957) model did not regard. The new considerations allowed the model to be used on axial turbine with lower aspect ratio and higher velocities.

Kacker and Okapuu (1982) improved the Ainley and Mathieson (1957) and Dunham and Came (1970) models through a restructuration of some loss source. They authors used experimental data of 33 turbines from 80's. The model included the compressibility and shock losses effects into the estimation of profile and secondary loss sources. The total loss coefficient account profile, secondary, tip leakage and trailing edge.

The aim of this work is to estimate the efficiency of a single stage axial flow turbine using the loss model developed by Kacker and Okapuu (1982). In addition, the results were compared with a commercial turbomachine design software named AXIAL, developed by CONCEPTS NREC®.

2. THE INTERNAL LOSS MODEL

The numerical tool developed can be used to determine the main operational characteristics and dimensions of axial turbines. Some information about axial flow turbine design constraints and its sizing are described by Saravanamuttoo, *et al.*, 2001.

The loss model implemented in this work was developed by Kacker and Okapuu (1982). The authors have made the implementation of Ainley and Mathieson and the Dunham and Came (AMDC) model, already was implemented into the computational program. The main differences are the introduction of compressibility and shock losses into the calculation of profile and secondary loss coefficients. New correlations from curves obtained by Kacker and Okapuu experiments should be fitted and implemented into the loss calculation subroutines.

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The method can be used to estimate the design and off design-point operation conditions. In this work, only the design-point was evaluated.

The total loss coefficient (Y), by Kacker and Okapuu model (1982), is defined in Eq. 1.

$$Y = \chi_{Re} Y_p + Y_s + Y_k + Y_{Te} \tag{1}$$

The coefficients are: Y_p: profile loss; Y_s: secondary loss; Y_k: tip leakage loss; Y_{Te}: trailing edge thickness; χ_{Re}: Reynolds number correction.

2.1 Profile loss

The profile loss coefficient depends on the correlation from Ainley and Mathieson loss model, adding the terms related with compressibility and shock waves effects. In addition, Kacker and Okapuu (1982) introduced two other modifications:

- the factor of 2/3 was applied to account for the advances in turbine aerodynamics;
- in the model developed by Dunham and Came, the profile loss is multiplied by a correction factor if the blade trailing edge thickness to pitch ratio (t_e/s) is different of 0.02 . In the Kacker and Okapuu model (1982) the factor of 0.914 was applied to obtain Y_p at zero trailing edge thickness condition. In addition, the trailing edge loss is an additional term that is calculated separately.

The profile loss coefficient (Y_p) is expressed as

$$Y_p = 0.914\left(\frac{2}{3} k_p Y_{p(i=0)} + Y_{shock}\right) \tag{2}$$

Where, $Y_{P(i=0)}$ is the profile loss coefficient based on Ainley and Mathieson (1957) and Dunham and Came (1970) models. This coefficient is obtained by interpolation between the results of nozzle and impulse-type sets of cascade tests in conjunction with the equation:

$$Y_{p(i=0)} = \left\{ Y_{p(\alpha in=0)} + \left|\frac{\alpha_{in}}{\alpha_{out}}\right| \left(\frac{\alpha_{in}}{\alpha_{out}}\right) \left[Y_{p(\alpha in=\alpha out)} - Y_{p(\alpha in=0)} \right] \right\} \left(\frac{t_{max}/c}{0.2}\right)^{\frac{\alpha in}{\alpha out}} \tag{3}$$

This equation depends on the inlet (α_{in}) and outlet (α_{out}) flow angles on the stator and rotor rows, the blade thickness (t_{max}) and the blade chord (c). The $Y_{p(\alpha in=0)}$ (nozzle blades) and $Y_{p(\alpha in=\alpha out)}$ (impulse blades) are correlations obtained from cascade data (see Saravanamuttoo, *et al.*, 2001, Fig. 7.24).

The compressibility effect is introduced by a constant k_p

$$k_p = 1 - k_2(1 - k_1) \tag{4}$$

The constant k_p is a combination of two corrections due to the effect of exit Mach number (M_{out}) and the effect of flow acceleration within the blade-to-blade channel. These effects are calculated by Eqs. 5 and 6, respectively.

$$k_1 = 1 - 1.25(M_2 - 0.02) \qquad \text{for } M_2 > 0.2 \tag{5}$$

$$k_2 = (M_{in}/M_{out})^2 \tag{6}$$

Where M_{in} is the inlet Mach number.

The loss due to the shock wave formation (Y_{shock}) is calculated by Eq. (7). It is a function of the inlet Mach number at hub ($M_{1,h}$), hub-to-tip ratio (r_h/r_t), inlet (p_{in}) and outlet (p_{out}) static pressures and inlet and outlet Mach numbers on the blade rows.

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$$Y_{shock} = 0.75\left(M_{1,h} - 0.4\right)^{1.75}\left(\frac{r_h}{r_t}\right)\left(\frac{p_{in}}{p_{out}}\right)\frac{1-\left(1+\frac{\gamma-1}{2}M_{in}^2\right)^{\gamma/\gamma-1}}{1-\left(1+\frac{\gamma-1}{2}M_{out}^2\right)^{\gamma/\gamma-1}} \tag{7}$$

Where, γ is the specific heat ratio.

2.2 Reynolds number correction

Kacker and Okapuu (1982) consider that Reynolds number affect only the profile loss source. The correction is applied for Reynolds number different of $2x10^5$. The Reynolds number correction is calculated by:

$$\chi_{Re} = \begin{cases} \left(\frac{Re}{2x10^5}\right)^{-0.4} & Re \le 2x10^5 \\ 1 & 2x10^5 < Re < 10^6 \\ \left(\frac{Re}{10^6}\right)^{-0.2} & Re > 10^6 \end{cases} \tag{8}$$

2.3 Secondary loss

Kacker and Okapuu (1982) introduced the compressibility effect and the influence of aspect ratio into secondary loss source of Dunham and Came model. The secondary loss coefficient given by Kacker and Okapuu (1982) is determined using:

$$Y_s = 0.04\left(\frac{h}{c}\right)\chi_{AR}\left[4\left(tan\,\alpha_{in} - tan\,\alpha_{out}\right)^2\left[\frac{cos^2\alpha_{out}}{cos\,\alpha_m}\right)\left(\frac{cos\,\alpha_{out}}{cos\,\alpha_{in}}\right)\left[1-\left(\frac{h}{c}\right)^2\left(1-k_p\right)\right] \tag{9}$$

Where h is the blade height and α_m is the average flow angle.

The term χ_{AR} is the influence of aspect ratio and it is determined using the following conditions:

$$\chi_{AR} = \begin{cases} 1-0.25\sqrt{2-h/c} & h/c \le 2 \\ 1 & h/c > 2 \end{cases} \tag{10}$$

The last term in brackets of Eq. (9) accounts the compressibility effects for secondary loss.

2.4 Trailing edge loss

In Kacker and Okapuu (1982) loss model, the trailing edge loss is expressed in terms of an energy loss coefficient $\Delta\varphi^2_{TET}$, as presented bellow:

$$\Delta\varphi^2_{TET} = \Delta\varphi^2_{TET(\alpha E\mp 0)} + \left|\frac{\alpha_{in}}{\alpha_{out}}\right|\left(\frac{\alpha_{in}}{\alpha_{out}}\right)\left[\Delta\varphi^2_{TET(\alpha in=\alpha out)} - \Delta\varphi^2_{TET(\alpha in=0)}\right] \tag{12}$$

The values of $\Delta\varphi^2_{TET(\alpha in=0)}$ and $\Delta\varphi^2_{TET(\alpha in=\alpha out)}$ are found from the relationship between the trailing edge loss coefficient and the ratio of trailing edge thickness (t_e) to throat opening (o) between two consecutive blades (see Kacker and Okapuu, 1982, Fig. 14).

Two main parameters are used to account the loss in the blade row. One based on temperature drop and other based on pressure drop. Therefore, the conversion from energy loss to pressure loss (in this case) is required. The conversion is done using the Eq. (13).

$$Y_{Te} = \frac{\left[1+\frac{\gamma-1}{2}M_{out}^{2}\left(\frac{1}{1-\Delta\phi_{TET}}-1\right)\right]^{-\gamma/\gamma-1}-1}{1-\left(1+\frac{\gamma-1}{2}M_{out}^{2}\right)^{-\gamma/\gamma-1}} \quad (13)$$

2.5 Tip leakage loss

The tip leakage loss (Y_k) is one of the most large loss source mainly for high pressure turbines (HPT). This loss increases abruptly from variations in clearance. There are some techniques used to decrease the impact of the flow that leak over rotor tip surface. For some cases, there are different formulations to account the loss. For unshrouded rotor blade, the leakage loss is calculated using an iterative process given by Eq. (14).

$$\frac{\frac{\Delta\eta}{\eta_0}}{\frac{\Delta k}{h\cos\alpha_{out}} x \frac{r_t}{r_m}} = 0.93 \quad (14)$$

Where, η_o is the turbine efficiency without tip clearance; r_m is the mean radius; r_t is the tip radius; $\Delta\eta$ is the difference between the efficiencies with and without tip clearance; Δk is the gap value.

For the shrouded rotor blades the loss coefficient due to tip leakage is calculated from Eq. (15).

$$Y_k = 0.37\frac{c}{h}\left(\frac{k'}{c}\right)^{0.78} 4(\tan\alpha_{in} - \tan\alpha_{out})^2\left(\frac{\cos^2\alpha_{out}}{\cos^3\alpha_m}\right) \quad (15)$$

Where, *k´* is the effective value of tip clearance *k*. It is a function of the geometrical tip clearance and the number of seals (sn)

$$k' = \frac{k}{sn^{-0.42}} \quad (16)$$

3. AXIAL FLOW TURBINE DESIGN PARAMETERS

The computational program developed is based on the meanline technique. The loss coefficients, flow and blade angles, dimensions and so on are determined in the meanline positions. In Tab. (1) are listed the main turbine design requirements and the main input parameters.

Table 1. Turbine design requirements and the main input data.

Mass flow (kg/s)	22.9036	Stage pressure ratio	1.69
Turbine inlet temperature (K) (1)	1114.15	Stage inlet flow angle (degree)	0
Turbine inlet pressure (kPa)(1)	190	Stage outlet flow angle (degree)	7.24
Shaft speed (rpm)	7500	Stage inlet absolute velocity (m/s)	194.029

(1) total proprieties

Using the meanline technique all fluid properties are calculated at blade leading and trailing edges. The developed computational program was named S&M LOSS. A flowchart of the S&M LOSS is showed in Fig. (1).

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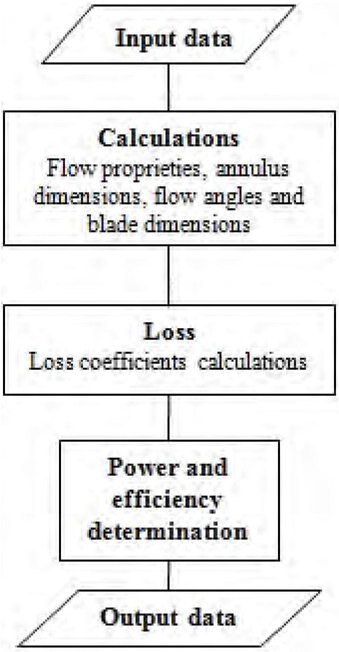

Figure 1. Flowchart of S&M LOSS.

In Fig. (2) is shown the single-stage axial flow turbine sizing calculated using S&M LOSS program.

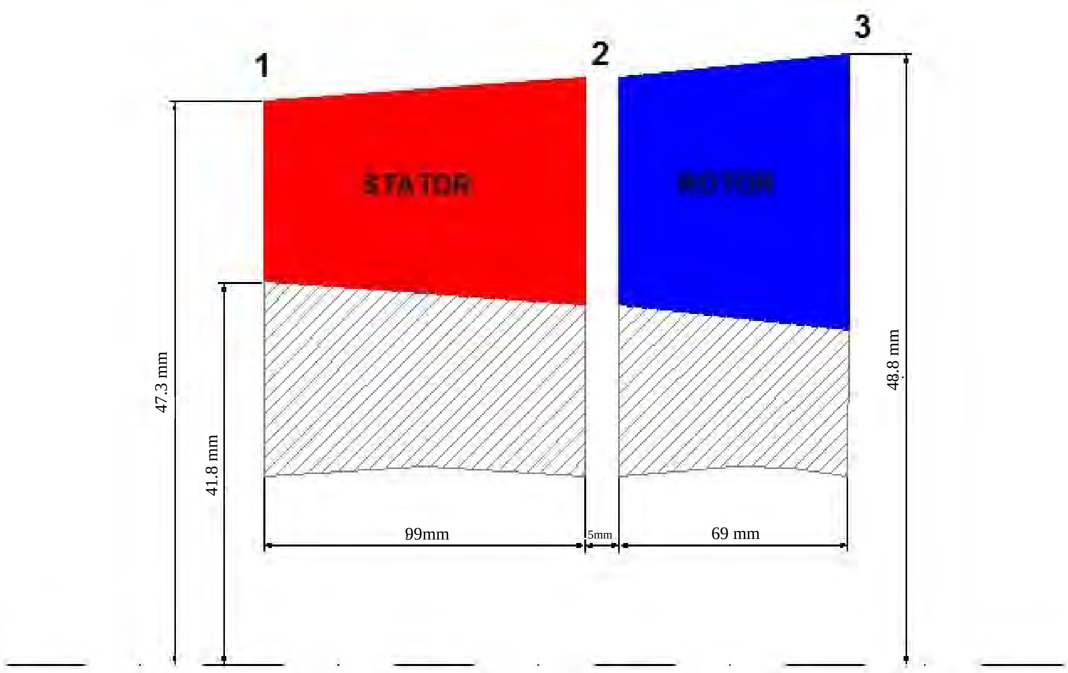

Figure 2. Single stage turbine streamwise view and its main dimensions.

4. RESULTS AND DISCUSSION

The axial turbine design requirements values are input data put the computational program developed in this work. The results are compared with a commercial software dedicated to turbomachines design called AXIAL^{TM} that was developed by CONCEPTS NREC$^{®}$.

Table (2) presents the results of the properties and the radius found after the preliminary design using S&M LOSS and the AXIAL^{TM}, the results of both programs are compared.

Table 2. Fluid properties and channel radius for both programs used at stations 1, 2 and 3.

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Parameters	S&M LOSS			AXIAL™			Error		
	1	2	3	1	2	3	1	2	3
Rt (m)	0.473	0.481	0.488	0.479	0.488	0.492	-1.27%	-1.50%	-0.86%
Rh (m)	0.418	0.411	0.403	0.400	0.400	0.400	4.34%	1.08%	0.84%
Rm (m)	0.446	0.446	0.446	0.441	0.446	0.448	0.96%	-0.09%	-0.63%
P (kPa)	168.28	124.84	98.16	179.65	122.76	100.00	-6.76%	1.67%	-1.87%
T (K)	1113.12	1045.27	994.62	1129.29	1040.14	997,70	-1.45%	0.49%	-0.31%
P_0 (kPa)	190.00	186.06	112.42	190.00	184.00	112.00	0.00%	1.11%	0.37%
T_0 (K)	1144.15	1144.15	1025.65	1144.15	1144.15	1026.85	0.00%	0.00%	-0.12%

Table (2) present the results for both program used. The major percentage error found comparing both programs is 6.76%, this value is considered small, showing that for the preliminary design the S&M program is near to the expectative.

Table (3) shows the results from the S&M LOSS using different loss models as aforementined. This comparison was made to show the differences in the results using these three loss models applied on axial turbine design.

Table 3. Results obtained for different loss models.

	Ainley and Mathieson		Dunham and Came		Kacker and Okapuu	
Total Loss	NGV	Rotor	NGV	Rotor	NGV	Rotor
	0.1208967	0.1339654	0.098442	0.1184296	0.09856	0.101853
Efficiency (%)	85.23		87.07		89.70	

The differences appear mainly because the first method was based on turbine technologies from the 50's (Ainley and Mathieson model). Thus, some important flow characteristics, for turbines that operates with high velocities, is not accounted, as shock wave formation, there is some constraints as blade aspect ratio values. The Dunham and Came model was enhanced doing the correction for the aspect ratio, Reynolds numbers and Mach number, increasing the model capabilities. The Kacker and Okapuu model is an improvement of the Dunham and Came model and account the compressible effects and losses caused by shock waves. Therefore, the most moderns and indicated model is Kacker and Okapuu (1982).

Table (4) shows the comparison of some variables between the results from the S&M LOSS and the commercial software. The S&M LOSS used the loss model developed by Ainley and Mathieson (1957) and Kacker and Okapuu (1982) while the AXIAL ™ used the model developed by Ainley and Mathieson (1957), Kacker and Okapuu (1982) and Moustapha and Kacker.

Table 4 - Comparison between the two numerical tools.

Parameter	NGV		Error	Rotor		Error
	S&M LOSS	AXIAL™		S&M LOSS	AXIAL™	
Total Losses	0.09856	0.098707	-0.15%	0.101853	0.120278	-15.32%
Power (MW)	-----	-----	-----	3.12	3.40	-8,96%
Efficiency (%)	89.70	88.66	1,17%	89.70	88.66	1,17%

Note that the results from both design tools are in good agreement. Each loss source calculated by both programs are presented in Figs. (3) and (4), separately, for Kacker and Okappu loss model.

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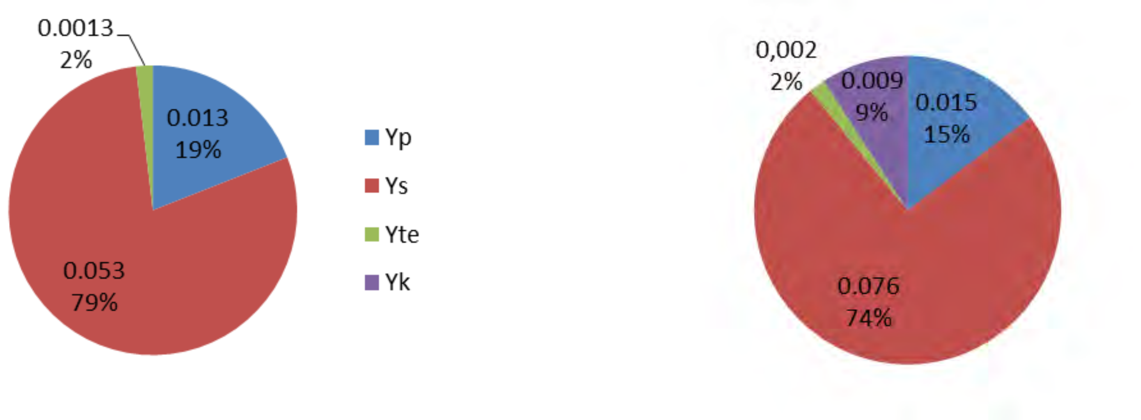
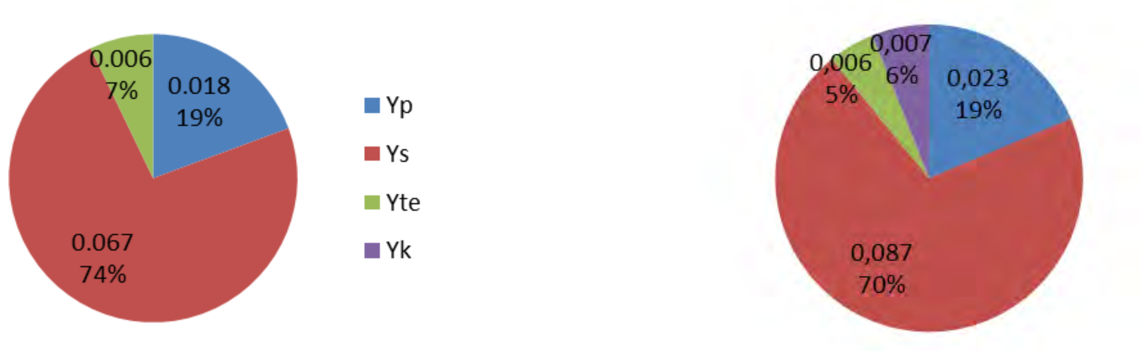


(a) (b)

Figure 3. Loss distributions - program developed. (a) Nozzle losses and (b) Rotor losses.



(a) (b)

Figure 4. Loss distributions - commercial program. (a) Nozzle losses and (b) Rotor losses.

It is possible to observe good agreement between the results. The calibration coefficients used in the commercial program, unfortunately is not the same and it is based on the proprietary expertise. In Figs. (3) and (4) are observed that the secondary losses are the major losses in nozzle and rotor, this occurs because in turbine blades with low values of aspect ratio (Nascimento and Tomita, 2012). With the computational tool developed in this work, it is possible to predict turbine design with good starting point for design refinement before the turbine geometry generation.

5. CONCLUSIONS

The results obtained in this work demonstrates that the computational tool developed is capable to perform the preliminary sizing of axial flow turbines when using an appropriate loss model to improve the turbine design prediction. The differences obtained for efficiencies calculation was 1.17%.

In this work the calculated losses were based on Kacker and Okapuu loss model. However, the program permit that other loss models can be implement because it follows a modular structure to allow an easy way to add new features enhancing the program capabilities.

The differences founded in preliminary design of axial flow turbine and consequently in loss modeling coefficients and its calibration, occur mainly, because there are differences in the design methodology between AXIAL^{TM} and S&M LOSS. But, qualitatively and also quantitatively the results of loss variables of each loss source are in good agreement for both design tool, for the turbine studied in this work. Moreover, optimization techniques as the use of Multi Objective Genetic Algorithm (MOGA), hybrid optimization schemes and others can be couple with the S&M LOSS program aiming better and faster preliminary design process.

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7. REFERENCES

Ainley , D.G. and Mathieson, G.C.R., 1957, “A method of performance estimation for axial-flow turbines”. British ARC. R&M2874.

Craig, H.R.M. and Cox, H.J.A.,1970, "Performance Estimate of Axial Flow Turbines". In *Proceedings of the Institution of Mechanical Engineers*, Vol. 185, No.32, p. 407-424.

Dunham, J. and Came, P.M., 1970, “Improvements to the Ainley/Mathieson method of turbine performance prediction”, ASME Journal Engineering for Power, p. 252-256.

Kacker, S.C. and Okapuu,U., 1982, “A Mean Line Prediction Method for Axial Flow Turbine Efficiency”, ASME Journal of Engineering for Power, Vol. 104, p.111-119.

Moustapha, S. H., Kacker, S. C., and Tremblay, B., 1990, “An Improved Incidence Losses Prediction Method for Turbine Airfoils,” ASME Journal of Turbomachinery., **112**, pp. 267–276.

Nascimento, F. S. and Tomita, J. T., 2012. “Implementation of IAPWS Toll in a 1-D Steam Turbine Design Program”. In *Proceedings of the 7st National Congress of Mechanical Engineering* **-***COBEM2012*. São Luís, Brazil.

Saravanamuttoo, H.I.H; Rogers, G.F.C.; Cohen, H., 2001. *Gas Turbine Theory* , Pearson Education Limited, 5th edition.

Traupel, W.,1966, "Thermische Turbomaschinen". Springer-Verlag, Berlin.

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