

Preliminary Design of Axial Flow Turbine for a Small Jet Engine

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Abstract – A computational program used to calculate the preliminary design of axial turbines which uses the Kacker and Okapuu's loss model was modified to improve the losses predictions implementing Tournier and El-Genk's loss model. The in-house program was written in FORTRAN 90 and is based on the meanline technique to calculate the axial turbine. As the losses interfere in the geometrical calculations applying more accurate loss models, the predictions of the preliminary design are more reliable. The program was applied to design a single-stage turbine and the results are compared with the commercial turbomachine design software AXIAL™. The improvements obtained by applying a more recent loss model have been discussed as the future works to improve the in-house program.

Keywords: Gas turbines, Turbomachines, Preliminary design, Loss model.

Nomenclature

Y_T	Total loss coefficient.
Y_P	Pressure loss coefficient.
Y_S	Secondary loss coefficient.
Y_k	Tip leakage loss coefficient.
Y_{TET}	Trailing edge loss coefficient.
χ_{Re}	Reynolds number correction coefficient.
Z_{TE}	Spanwise penetration depth.
h	Blade height.
k_{in}	Constant factor.
K_P	Mach number correction factor.
Y_{shock}	Shock loss coefficient.
f_{ar}	Aspect ratio factor.
k	The clearance value.
C_L	Blade lift coefficient based on mean vector velocity.
s	Pitch and distance between the blades in cascade.
c	Blade chord.
K_S	Secondary constant.
δ^*	Boundary layer displacement thickness.
C_x	Axial chord.
x	Reference length.
F_t	Tangential loading parameter.
ρ	Density inlet.
W	Relative velocity inlet.
μ_t	Turbine viscosity.
$\Delta\eta$	Efficiency drop.
η_0	Inlet turbo-machine polytropic efficiency.
Δk	Gap between the blade and casing.
r_t	Tip radius.

r_m	Averaged radius of the blade.
τ_{eff}	Effective clearance value.
α_1	Inlet flow angle of the stator row.
α_2	Inlet flow angle of the rotor row.
α_3	Outlet flow angle of the rotor row.
α_m	Mean flow angle between stator and rotor.
$K_G, K_E,$	Constants of Yaras and Slojander loss model.

1. Introduction

Gas turbines engines has several components installed around it to provide thrust or power depending on its application, if aeronautical or industrial. The mechanical main components are compressor, combustion chamber and turbine and to increase the gas turbine design performance, improvements on these components are necessary. This involves a very high amount of investments over several multidisciplinary areas as computing, programming, math, physics, engineering, hardware, labs, tests, instrumentation, certification, high specialized labor of technicians, engineering and researchers. Generally, gas turbines are a subject of Defense Department of each country and there is no technological transfer due to the military aspect and sovereignty. This is the fact that a country, in which, wants to develop this world class technological product have to do the own investments starting with a high-end human resources formation.

Furthermore, manufacturing has the need to take into account the component tests, because it is common that several parts and components of gas turbines are designed and manufactured by the same company. This is due to the requirements of a very specific devices and mechanical pieces as bearings, seals, couplings and so on. Improvements on compressor and turbine designs are highly bound with engine enhancements. New engines depending on an aggressive investment in material technology involving super alloys and high temperature metals. Since the World War II for the actual days the development with these alloys increase the temperature ratio in approximately 14°F per year [1]. Therewith the engines core tends to be smaller and these changes are supported by high velocities, material strain and stresses. For aeronautical purposes, a lightweight and low power-to-thrust ratio are desired including acceptable specific fuel consumption. The new generation of engines must meet the requirements of noise and emissions [2] These topics are today the real challenges of gas turbine industry and involves many research centers, institutes and academic collaborations.

The development of losses models is done by several authors [3-5] in the last years. Ainley e Mathieson [6] made experiments and developed their correlations through experimental data results for axial flow turbines from blade design and measurement equipment for that time. This model has been used for many years and is the basis for many researches. The model has limitations as the flow Mach number, average Reynolds number, aspect ratio, and the ratio between trailing edge thickness and blade chord.

Later, Dunham and Came [7] made improvements in Ainley and Mathieson [6] model based on new tests for high velocity turbines. In this model, some loss sources were changed to validate its use for higher Mach number, influence of Reynolds number, improvements in tip losses and the possibility to apply the old model in turbine blades with low aspect ratios.

More changes in Ainley and Mathieson [6] and Dunham and Came [7] loss models are done by Kacker and Okapuu [8] where are restructured some loss sources, using the experimental data of 33 turbines (modern for the time). This model includes the compressibility effects and the losses generated from shock waves formation. The principal advantages of the Kacker and Okappu loss model [8] is the easy implementation in a current code as the quite approximate of the real axial turbine efficiency having a deviation of 1.5% more or less.

Moustapha et al [9], modified the profile and secondary losses to calculate the off-design conditions to adjust the earlier models to the new airfoils profiles. Looking to correct the predictions of the profile losses, Benner et al. [10] studied experimentally the low-speed cascade wind tunnel obtaining reliable predictions for this loss. More recently, Benner et al. [11,12] proposed new correlations to evaluate the profile and secondary losses in turbines, creating the penetration depth correlation improving the predictions for the newest turbines. Tournier and El-Genk [13] developed a loss model which incorporates the refinements proposed by Benner et al. [11] in the Kacker and Okapuu loss model, also the authors suggested changes in the calculation of the tip clearance losses, showing accurate predictions to the geometrical parameters and losses for a most recent turbine design.

In this work was applied a numerical in-house tool with is used to the preliminary turbine design to determine the geometrical parameters, the engine thermodynamic cycle calculations and the total losses. Basically, the code capabilities obtain reliable results to analyse a new turbine design. The computational code developed uses the Kacker and Okapuu loss model [8] and to improve the predictions of this model is implemented by the Tournier and El-Genk [13] loss model. Modular subroutines and easy handling of variables were structured into the code to facilitate its understanding and enhancements. The technique applied to the preliminary design of the turbomachine is the meanline. The turbine requirements, at the design-point operation, were based on the work of Martins [14]. The simulation results will be compared with the results from a commercial turbomachines design software AXIAL™® analysing the losses and flow properties along the turbine stage.

2. Methodology

The total losses of the Kacker and Okapuu model [8] is expressed by Eq. (1):

$$Y_T = \chi_{Re} Y_P + Y_S + Y_k + Y_{TET}, \quad (1)$$

where, Y_t is the total loss coefficient; Y_p is the profile loss coefficient; Y_s is the secondary loss coefficient; Y_k is the tip leakage loss coefficient; Y_{tet} is the trailing edge thickness coefficient and, χ_{re} is the Reynolds number correction coefficient in [8].

Considering the Eq. (1) the authors incorporate more losses to refine the earlier models [6,7] improving the prediction of the losses to the preliminary design. In the passage of the flow in a row of blades, the profile losses are caused by the blade curvature, then the blade angles had an influence in this loss, additionally, the authors include the channel flow acceleration and the shock loss coefficient.

Secondary loss is caused by the formation of secondary flow along the blade passage due to the rotation of the blade rows generating a non-uniformity of the radial velocity [2], therefore these losses are dependent on the high velocities practised in axial turbines. The coefficient is calculated based on [8] and the function is changed to correct the aspect ratio for values less than two. Also, as the flow is accelerated between the blade endwall it is accounted for the compressibility effects in the loss model, for that is applied a Mach number correction factor.

When it is evaluated losses for high pressure turbines (HPT) the main responsibility for the pressure drop is for the tip clearance loss [8] which could be added to some models in the secondary losses. To improve the earlier models for unshrouded blades, Kacker and Okapuu suggested an iterative process which is dependent on the efficiency drop calculated for the current model, obtaining values more accurate than in the AMDC model [6,7].

All in all, the model also separated the secondary losses in the trailing edge loss coefficient. This loss is defined as the function of the energy coefficient. The trailing edge loss is calculated considering the trailing edge thickness and the exit Mach number because the drag force is dependent on these values.

Tournier and El-Genk [13] proposed refinements in the Kacker and Okappu's model [8], then the total pressure loss coefficient is defined as,

$$Y_T = (Y_P + Y_S)' + Y_{TET} + Y_k' \quad (2)$$

The model suggested to calculate the profile and secondary losses as one, where the spanwise penetration depth corresponds to the separation line between the primary and secondary loss regions [11]. The secondary loss coefficient (Y_s) is changed by the authors to consider the boundary layer displacement thickness to calculate the formation of the secondary flow in the blade row. Also, Tournier and El-Genk [13] modified the tip clearance losses applying the approach of Yaras and Slojander [15] separating into tip and gap losses, for that, is considered the lift coefficient in the calculations.

Table 1 shows the losses changes in the model calculations between the Kacker and Okapuu's and Tournier and El-Genk [13]. Due to the limits established to the article size more details of the Kacker and Okapuu's and Tournier and El-Genk's loss models are encountered in references [8,13].

Table 1: Differences in the loss coefficients calculations for the Kacker and Okapuu [8] and Tournier and El-Genk [13].

Loss Coefficients	Kacker and Okapuu	Tournier and El-Genk
$(Y_P + Y_S)'$	$Y_P + Y_S$	$(Y_P + Y_S)' = (1 - Z_{TE}/h) + Y_P' + Y_S'$
Y_P	$Y_P = 0.914(k_{in}K_P Y_{P(i=0)} + Y_{shock})$	$Y_P = 0.914(k_{in}K_P Y_{P(i=0)} + Y_{shock})$
k_{in}	$2/3$	0.825 for IGV 2/3 for reaction blades
Y_S	$0.04008 f_{ar} \left(\frac{\cos \alpha_2}{\cos \alpha_1} \right) \left(\frac{C_L}{s/c} \right)^2 \frac{\cos \alpha_2^2}{\cos \alpha_m^2} K_S$	$f_{ar} \frac{0.038 + 0.41 \tanh(1.2\delta^*/h)}{\sqrt{\cos \alpha_m (\cos \alpha_1 / \cos \alpha_2) (c \cos \alpha_2 / C_x)^{0.55}}}$
Z_{TE}/h		$\frac{0.1 F_t ^{0.79}}{\sqrt{\cos \alpha_1 / \cos \alpha_2} (h/c)^{0.55}} + 32.7 \left(\frac{\delta^*}{h} \right)^2$
F_t		$2 \frac{s}{c \cos \alpha_m} \cos \alpha_m^2 [\tan(\alpha_1) + \tan(\alpha_2)]$
δ^*		$0.0463x/(\rho_1 W_1 x / \mu_t)^{0.2}$
f_{ar}	$\left(1 - 0.25 \sqrt{2 - \frac{h}{c}} \right) / (h/c)$ for $h/c \leq 2$ (c/h) for $h/c > 2$	$(c/h)^{0.55}$ for $h/c \leq 2$ $1.36604(c/h)$ for $h/c > 2$
Y'_k	$\frac{\Delta\eta}{\eta_0} \frac{\Delta k}{h \cos \alpha_2} \frac{r_t}{r_m} = 0.93$ for unshroud blades $0.37 \frac{c}{h} \left(\frac{k'}{c} \right)^{0.78} \left(\frac{C_L}{s/c} \right)^2 \frac{\cos \alpha_3^2}{\cos \alpha_m^3}$ for shroud blades	$Y_{Tip} + Y_{gap}$
Y_{Tip}		$1.4K_e \left(\frac{c}{s} \right) \frac{k \cos \alpha_2^2}{h \cos \alpha_m^3} C_L^{1.5}$
Y_{Gap}		$0.0049K_G \left(\frac{c}{s} \right) \frac{c \sqrt{C_L}}{h \cos \alpha_m} \frac{\cos \alpha_2^2}{\cos \alpha_m^3} C_L^{1.5}$ for unshroud blades $1.102K_e \left(\frac{c}{s} \right) \frac{\tau_{eff} \cos \alpha_2^2}{h \cos \alpha_m^3} C_L^{1.5}$ for shroud blades

3. Axial Flow Turbine Parameters

Table 1 shows the main data used in design-point turbine operation. Based on these requirements, the computational code will be handled to calculate the main machine characteristics.

Table 1: Axial turbine design data.

Mass Flow (kg/s)	7.952
Inlet Static Pressure (kPa)	476.277
Inlet Static Temperature (K)	1173.0
Drop in stage temperature (K)	172.28
Pressure ratio	2.16
Rotation (rpm)	28150
Stator aspect ratio	0.981
Rotor aspect ratio	1.783
Power (MW)	1.722
Inlet flow angle (grades)	0
Swirl angle (grades)	61.75

This turbine was designed using two different programs (commercial and in-house) and the results are compared. After several design parameters handling and modifications to reach the power, efficiency and pressure ratio the turbine geometry can be estimated. Fig. 1 shows the 3D view of the axial flow turbine calculated.

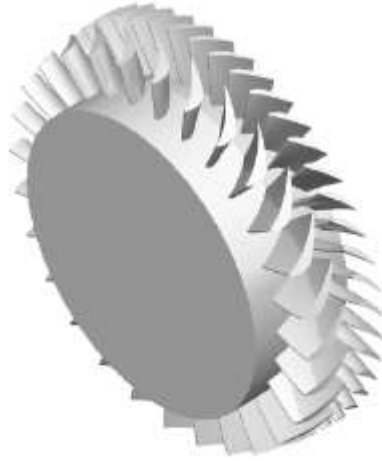


Fig. 1: Meridional section of axial turbine with one stage. Martins (2011).

The flowchart in Fig. 2 shows the procedure utilized in the program to calculate the turbine sizing from design requirements [16]. It was established as a design parameter that the turbine had a constant mean ratio for the stage. The flow, blade inlet and outlet angles are determined to make the use of loss modelling to improve the prediction of power and efficiency due to the losses quantification. Following to compare the preliminary design with the Martins work [14] the temperature drop is set as the same applied in the commercial software of AXIAL™®.

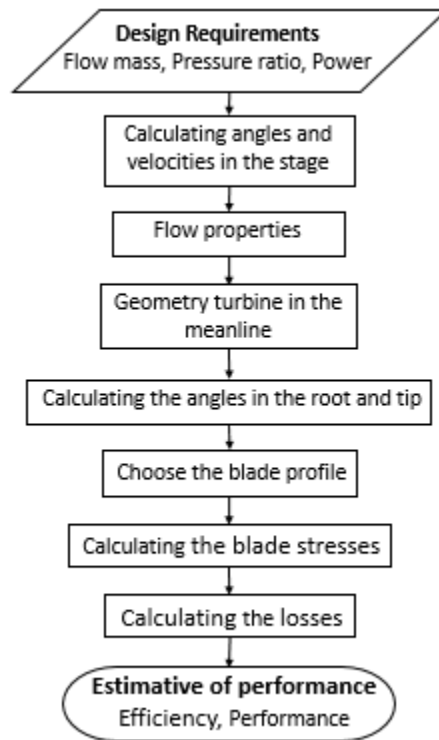


Fig. 2: Procedure for design and loss calculation.

4. Results and Discussion

As explained in the above sections, after the implementation of the loss model of Tournier and El-Genk [13] in the in-house program are set the design requirements shown in Table 1. The in-house program was applied to calculate the losses, properties and geometry of the preliminary design of an axial turbine for the Kacker and Okapuu and Tournier and El-Genk [8,13] loss models. The results obtained with the in-house program were compared with the results obtained in [14] with the AXIAL™ program.

4.1. Losses coefficients

The losses estimative is calculated to the in-house and to the commercial program. Table 2 showed the total losses calculated to the turbine stator and rotor rows.

Table 2: The design dates to the AXIAL™ and the program developed in-home.

		AXIAL™	Kacker and Okapuu	Tournier and El-Genk
Total Losses	Stator	0.0767	0.0754	0.0489
	Rotor	0.1492	0.213	0.117

Analysing Table 1 is presented that the total stator losses demonstrated an underpredict to the in-house program in relation to the commercial program for both loss models, also the closest value is shown to the Kacker and Okapuu [8] loss model. In the rotor the total losses are overpredicted to the Kacker and Okapuu [8] loss model and is underpredicted to the Tournier and El-Genk [13] the last one is the closest value to the commercial program. Therefore, applying the Tournier and El-Genk [13] loss model in the in-house program, the rotor predictions are improved, which is responsible for the higher-pressure losses.

Figs. 3 and 4 show the stator and rotor rows comparing the losses calculate for the Kacker and Okapuu [8] and Tournier and El-Genk [13].

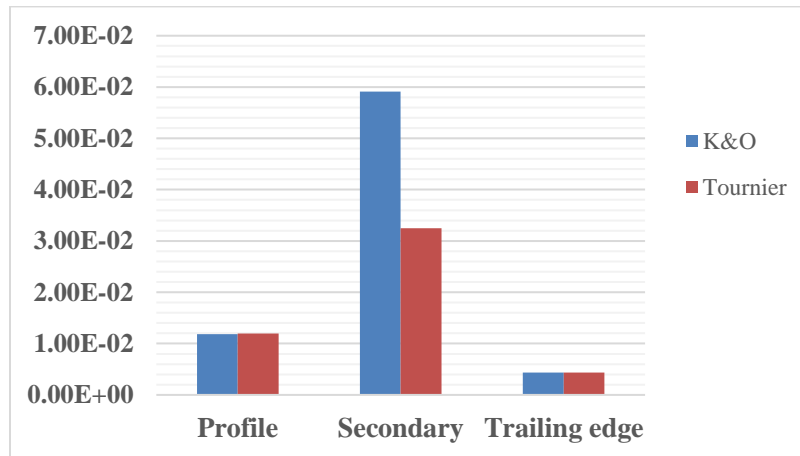


Fig. 3: Losses to the stator row for Kacker and Okapuu [8], Tournier and El-Genk[13] and AXIAL™.

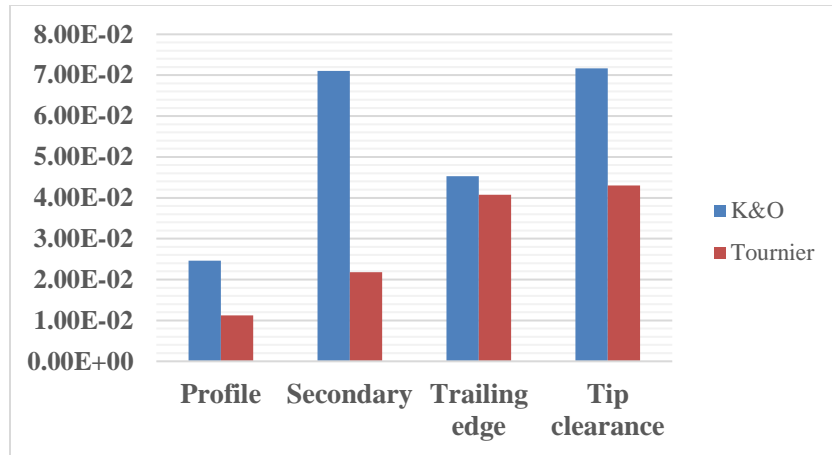


Fig. 4: Losses to the rotor row for Kacker and Okapuu [8], Tournier and El-Genk [13] and AXIAL™®.

Comparing the profile losses obtained to the models of Kacker and Okapuu [8] and Tournier and El-Genk [13] had a difference of 1% in the stator row due to the change in the factor of 2/3 for a higher value to the IGV. Also, other factors are introduced by Tournier and El-Genk [13] as a factor to correct the flow Reynolds number as proposed by [15] and the spanwise penetration depth defined by [12], then the rotor profile loss had higher differences between the models indicating an overpredicted of the Kacker and Okapuu model.

For secondary losses the value of stator and rotor rows made a large difference between the models, indicating an overprediction of the Kacker and Okapuu loss model [8]. It is expected differences due to the introduction of new corrections as the boundary layer displacement thickness and stagger angle, although the main differences occur because the Kacker and Okapuu model is based on the engines test and the Benner et al the corrections are based on the cascade tests. As a result, the Kacker and Okapuu's model overpredict the secondary losses and to improved that Benner et al.[12] suggested an engine-to-cascade correction factor improving the accuracy of the calculated and experimental data.

The trailing edge losses had closest values for both loss models because is not suggested changes to evaluate this loss. The tip clearance losses were estimated to the unshroud blades. The Kacker and Okapuu[8] loss model estimates this loss through an iterative method which related the turbine efficiency without the tip clearance loss and the total efficiency with the tip clearance loss. Next, the iterative process is done until the convergence and then the tip clearance loss is evaluated. Tournier and El-Genk[13] based the tip clearance loss on the work of Yaras and Sjolander[15] which consider the blade lift coefficient in their calculations. It is shown in Fig. 4 an overpredict of the tip clearance loss for the Kacker and Okapuu [8] loss model than to the losses calculated with the Tournier and El-Genk [13] model.

4.2. Properties and Geometric Results

In the Fig. 5 is shown a sketch of the turbine designed applying the in-house program.

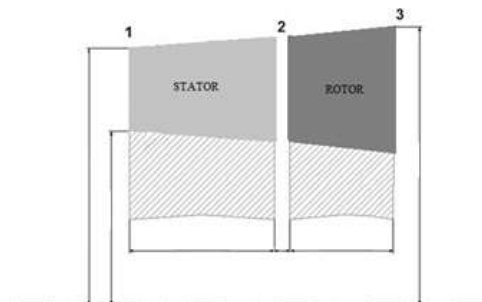


Fig. 5: Sketch of the turbine designed with the program developed in-house.

Table 3 presents the properties calculated for the in-house program and AXIAL program. The stations 1, 2 and 3 are located as shows in the Fig. 3.

Table 3: The gas properties data to the AXIAL™® and the program developed in-house.

		AXIAL™®	Kacker and Okapuu		Tournier and El-Genk	
Properties	Station	Value	Value	Percentage error	Value	Percentage error
<i>Total pressure</i>	1	476227	476227	-	476227	-
	2	467109	468397.55	0.276 %	470329.71	0.689 %
	3	220596	204890.36	-7.12 %	217960.458	-1.195 %
<i>Relative pressure</i>	1	-	-	-	-	-
	2	379479	379424.11	-0.014 %	381173.68	0.447 %
	3	348398	320341.91	-8.05 %	340776.75	-2.188 %
<i>Static pressure</i>	1	438894	438894	-	438894	-
	2	348199	348207.995	0.003 %	349813.62	0.464 %
	3	164215	15294.904	-6.954 %	162541.80	-1.019 %
<i>Density</i>	1	1.288666	1.288666	-	1.288666	-
	2	1.074439	1.067956	-0.603 %	1.072880	0.145 %
	3	0.595291	0.550533	-7.519 %	0.585652	-1.619 %

The properties calculated by the in-house program applying the Kacker and Okapuu[8] and Tournier and El-Genk [13] loss models are compared with the values obtained by the AXIAL™® program in the work developed by [martins] demonstrating differences lower than 9% for the Kacker and Okapuu model and lower than 2.5% for the Tournier model. All in all, applying the Tournier model the properties predictions are improved.

Table 4 presents the geometric parameters calculated to the developed in-house program and to the AXIAL™® program.

Table 4: The geometry data to the AXIAL™® and the program developed in-house.

		AXIAL™®	Kacker and Okapuu		Tournier and El-Genk	
Geometry	Station	Value	Value	Percentage error	Value	Percentage error
<i>Annulus area</i>	1	2.6044E-02	2.60E-02	-0.0001%	2.6044E-02	-0.0001%
	2	3.1530E-02	3.14E-02	-0.4429%	3.1246E-02	-0.8998%
	3	3.7598E-02	3.99E-02	6.1589%	3.7520E-02	-0.207%
<i>Throat area</i>	1	1.6367E-02	1.68E-02	2.4671%	1.6694E-02	1.997%
	2	1.8535E-02	1.01E-02	-45.3332%	1.0086E-02	-45.584%
<i>Blade height</i>	1	3.2844E-02	3.19E-02	-2.8174%	3.1919E-02	-2.817%
	2	3.8744E-02	3.85E-02	-0.7047%	3.8294E-02	-1.160%
	3	4.7963E-02	4.90E-02	2.1664%	4.6004E-02	-4.085%
<i>Stagger angle</i>	1	37.6298	39.0836	3.8635%	39.0840	3.865%
	2	34.6233	36.3292	4.9271%	36.4120	5.166%
<i>Chord</i>	1	3.95E-2	4.30E-2	8.7718%	4.296E-2	8.772%
	2	2.69E-2	2.50E-2	-7.2409%	2.495E-2	-7.241%
<i>Axial chord</i>	1	3.1283E-02	3.34E-02	6.6092%	3.3350E-02	6.609%
	2	2.2136E-02	2.01E-02	-9.1884%	2.0081E-02	-9.285%
<i>Aspect ratio</i>	1	0.9809	0.81915	-16.36%	0.8171	-16.70%
	2	1.7830	1.73277	-2.07%	1.6691	-6.41%

The higher differences in the geometrical calculations are found in the throat area to the rotor row. The discrepancies are due to the differences in the turbine design because the AXIAL™ program considered for each stage the enthalpy constant and to the in-house program is considered the axial velocity constant in each station.

4. Conclusion

The program developed in this work has the capability to help engineers design axial flow turbines without cooling, using the main data from engine thermodynamic cycle calculation. Initially, the in-house program was implemented by the Kacker and Okapuu [8] loss model. Looking to improve the losses predictions is implemented a more recent work by Tournier and El-Genk [13]. The results obtained for the total losses showed improvements in the rotor predictions. The results found that the preliminary turbine design increases the accuracy to predict the flow properties. Although the geometry calculated to the in-house program had some discrepancies due to different design criteria. The algorithm created allows the implementation of other loss models due to the modular structure.

In future works, the implementation of other design criteria and optimization techniques will be improve the predictions to the in-house program.

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