Development of One-dimensional Turbomachinery Meanline Solver with Physics-based Detailed Loss Models

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ARTICLE INFO

Received : 20 June 2019
Revised : 07 October 2019
Accepted : 21 October 2019

Keywords:

Loss model; meanline solver;
entropy generation rate;
turbomachinery, axial turbine

ABSTRACT

Turbomachinery design systems have traditionally been a highly guarded in-house capability of large turbomachinery corporations, though commercial software has started becoming available in the last decade along with the recent release of open source software such as MULTALL. While attempting to use MEANGEN, the meanline solver in MULTALL, for preliminary design optimization of a 2-stage axial turbine for 100 kW micro gas turbine application, it was found that MEANGEN does not provide an estimate of the component losses in a turbomachine such as blade boundary layer loss, trailing edge loss, tip leakage loss, and endwall loss. The present work describes the development of an in-house meanline solver TURBOMEAN with 1D component loss models quantified in terms of entropy generation rate for transonic flow through a multi-stage axial turbine. Special emphasis is put on understanding the physical origins of the loss mechanisms in turbomachinery flows. TURBOMEAN allows the user to run an optimization loop in the first stage of design itself and in turn significantly reduce the time required to optimize a design using 3D-CFD analyses. The 1D component losses being dependent on the flow physics rather than historical correlations may very well be used for modern blade designs which have considerably three-dimensional shape. Major effort has been put in realizing those loss models into codes and integrating them with a meanline solver. Practical assumptions were sort after wherever analytical solution was not possible for developing expressions of terms which depended on flow conditions. These assumptions are then tested by validating the design system. The solver and the models are implemented in MATLAB and validated against data available in literature. It is expected that the outcome of the present work can also be extended for axial compressors as well as radial turbomachines due to considerable similarities in underlying aerodynamic loss mechanisms.

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1. Introduction

Aerodynamic efficiency of a turbomachine is directly related to the losses generated within the flow passages of the turbomachine. Many loss models have been proposed since the 1950s with the most prominent ones being those by Ainley and Mathieson (1951), Soderberg (1949) and Craig and Cox (1971) [(C.A.Hall, 1978)]. These loss models were based on loss correlations obtained experimentally from 2D cascade measurements and were tuned so that they could satisfy the experimental data available back then. Despite giving consistent results within their applicable range, these loss models essentially failed to capture the detailed physics of the flow field and can very well be considered outdated today as turbomachinery blade designs become more and more three dimensional in shape.

Earlier loss models were mainly quantified in terms of stagnation pressure loss due to the ease of measuring stagnation pressure in a cascade test. Later, enthalpy and energy loss coefficients were also introduced, though each of these coefficients depends on the frame of reference. Thus, they are inconvenient to use in real turbomachinery flows where both stationary and rotating components are present. Denton [(J.D.Denton. 1993)] provided a different approach where the losses are quantified in terms of entropy – a thermodynamic property that is independent of the frame of reference. Flow in a turbomachine can be mostly considered adiabatic; hence, the losses generated can be entirely attributed to the

irreversibility in the flow which in turn leads to entropy generation [(J.D.Denton. 1993)]. By accounting for this increase in entropy, aerodynamic loss resulting from any flow behavior characteristic can thus be captured.

The typical aerodynamic design process for a turbomachine consists of 1D meanline design, 2D through-flow analysis, 2D blade design followed by 3D CFD analysis along with multiple iterative loops in-between. The underlying software for such design systems have traditionally been a highly guarded in-house capability of large turbomachinery corporations. Commercial software has started becoming available in the last decade for 2D and 3D analyses but very few are available for performing 1D meanline design. A recently released open-source turbomachinery design system, MULTALL [(J.D.Denton. 2017)], was being explored for designing a 2-stage axial turbine for a 100-kW micro gas turbine (MGT) application. The meanline solver in MULTALL, called MEANGEN, gives a reasonable estimate of the axial turbine corner points for a given input target efficiency. However, it does not provide an estimate of the typical component losses encountered in a turbomachine, such as the blade boundary layer loss, trailing edge loss, tip leakage loss, and endwall loss. Another drawback of MEANGEN is that it has many hard-coded assumptions such as constant axial velocity, repeating stages, fixed Zweifel coefficient, etc. resulting in simplified velocity triangle calculations. Thus, a designer can use MEANGEN only for the creation of initial flowpath geometry but not for preliminary design optimization studies aimed at maximizing efficiency.

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As the turbine blade designs become more complex in shapes, new flow behaviour pattern is observed which is not quite captured by the earlier loss models as mentioned previously. Old conventional designs are suitable for the usage of such models. These models might predict higher losses due to a new design feature owing to their heavy dependence on the correlations which are based on the experimental cascade data of old conventional design and not on the actual flow conditions. The designer might become discouraged to try a new design feature owing to large loss or no improvement predicted by using the old loss models. In order to have a better understanding of the performance of such blades, special experiment rigs can be set up and their experiments can be conducted. The setup of the rig and its experimentation is itself a costly affair and consumes a lot of time. Also, frequent changes in shapes require a special setup which significantly increases the cost to the turbomachinery organization. Computational techniques can come to the rescue as far as the time required for the experiment setup is concerned. But, iterating the blade designs with 3D CFD analyses takes a lot of time and is heavily dependent on the experience of the designer. This time can be considerably reduced by adding an optimization loop at the first stage of design itself. For having an iterative loop at 1-dimensional design stage, it is essential to have a 1-D component loss prediction model in place. For modern turbine blade shapes, it is crucial to have a loss model in place which depends on the flow physics rather than on the geometric parameters. Thus there is a need of having a meanline solver integrated with such a loss model which would enable the designer to have an optimized result for the first stage itself, which in turn will help in speeding up the whole design process and making a better design with lesser reliance on the designer experience.

The present work describes the development of an in-house 1D meanline solver, TURBOMEAN, with integrated physics-based detailed loss models considering entropy generation as a measure of loss. TURBOMEAN is primarily aimed at preliminary design optimization studies of multi-stage axial turbines, however, the implementation has been kept generic enough to allow easy extension and future validation for axial compressors as well as radial turbomachines.

2. Methodology

The subsequent sections are mainly divided into two segments, where the first segment concerns itself with the loss model formulations and the second segment deals with the development of an in-house meanline solver and integrating it with the loss models for predicting turbine efficiency.

2.1 Loss Model Formulation

The loss model equations in terms of entropy loss coefficient developed by Denton [(J.D.Denton. 1993)] are the most fundamental physics-based equations in the literature. They are extended in the present work and are summarized below. Development of loss models requires a thorough understanding of the physical origins of each loss component. Flow in a turbomachine consists of multiple physical phenomena making it challenging to capture the individual loss components, although a breakdown of losses into individual components is always beneficial as it helps to focus the design effort to target the origins of the individual loss mechanisms. Historically, turbomachinery losses have been broken down into profile loss, endwall loss and tip leakage loss [(J.D.Denton. 1993)]. As it was also mentioned previously, we would consider the breakdown of losses when entropy generation is used as a measure of loss. This approach takes into account only those flow phenomena which actually lead to loss i.e. entropy generation. As an example, flow leakage over shrouded blades does not necessarily contribute to the losses, but in turn, increases the net flow over the blade which helps in increasing the work extraction. Thus, having entropy as a measure of loss effectively helps in including only those flow phenomena that contribute to the loss. The losses are calculated in terms of entropy loss coefficients which is explained below

An entropy loss coefficient (ζ_s) is defined by Denton as

$$\zeta_s = \frac{T_2 \Delta s}{h_{02} - h_2}, \quad (1a)$$

$$\zeta_s = \frac{T_2 \Delta s}{h_{01} - h_1}, \quad (1b)$$

In Eqn. (1), subscripts 1 and 2 are blade row or stage inlet and exit stations, subscript 2s is the isentropic exit, T is the static temperature (K), Δs is the change in specific entropy (KJ/K.kg) and h_0 (KJ/kg) and h (KJ/Kg) are the stagnation and static enthalpies, respectively.

The primary mechanism of entropy generation [(J.D.Denton. 1993)] in turbomachinery flows is due to viscous friction in the boundary layers as well as free shear layers wherever mixing of two fluids takes place. Thus, the losses generated due to boundary layer formation on blade surfaces and endwalls as well as the losses occurring due to mixing process downstream of the trailing edge and over tip and/or hub clearances, can all be accounted for by this mechanism [(J.D.Denton. 1993)]. Entropy generation can also occur due to mixing of coolant fluid with the main flow as well as due to non-equilibrium processes such as rapid expansion or shock. As we don't intend to include the cooling arrangements in our blade designs, the losses generated due to coolant mixing are safely ignored. Loss Mechanisms in Turbomachines [(J.D.Denton. 1993)] gives information about the contribution of each individual loss component to the overall loss varying with Mach number. It is from the paper [(J.D.Denton. 1993)] that we conclude that for transonic flow conditions pertaining to our MGT application, shock losses generated are relatively smaller than the other losses and contribute less to the overall loss. Thus, they are considered negligible.

The losses that are considered are due to the viscous friction mechanism of entropy generation consist of blade boundary layer loss, trailing edge loss, endwall loss, and tip leakage loss. Boundary layer formation and shear mixing process are viscous phenomenon and thus can be accounted for by viscous friction mechanism of entropy generation. In this way, the entropy increase contributed by each of loss can be calculated. When expressing each of the loss model in terms of entropy [(J.D.Denton. 1993)], the expressions also depend on one tuning parameter in addition to the flow conditions. Each of the loss models consists of only one empirical term, the so-called tuning parameter, which needs to be obtained through experiments. These are the dissipation coefficient C_d , discharge coefficient C_{dis} and base pressure coefficient C_{pb} . Also all loss models require the blade surface velocity distribution. This data is not expected to be available at the preliminary design stage and hence an idealized velocity distribution must be assumed upfront. Denton [(J.D.Denton. 1993)] assumed a rectangular velocity distribution as shown in Figure 1 Blade Loading Distribution [4], [8] and calculated variation of each loss component as a function of inlet and exit flow angles. Such a velocity distribution is too idealized for use in real design scenarios, however, the data presented in [(J.D.Denton. 1993)] is highly valuable for validation of the present work. A more realistic, though still idealized, piecewise-linear blade loading distribution was presented by Dambach et al. [(Dambach, 2000)] as shown in Figure 1 Blade Loading Distribution [4], [8]. A parametrized version of this loading distribution based on inlet and exit Mach numbers is used in the current implementation.

1a. Rectangular Blade Loading Distribution

1b. Piece-wise Linear Loading Distribution

Figure 1: Blade Loading Distribution [4], [8]

Each of the loss models also requires an estimate of boundary layer thicknesses such as δ^* and δ . In addition to surface velocity distribution, most boundary layer thickness correlations require knowledge of type of boundary layer, laminar or turbulent, as well as an accurate estimate of the point of laminar to turbulent transition.

For the validation purpose, all losses have one loss model which uses the conditions prescribed in Ref. [(J.D.Denton. 1993)] and uses the rectangular blade loading distribution. Subsequent models become more realistic with complex flow conditions and with piece-wise linear loading distribution. Also in later sub-models, the tuning parameter is used as a function of a flow parameter in order to avoid over-estimation of losses when they are taken as a standard constant. Each loss model formulation and their sub-models have been explained below:

2.1.1 Blade Boundary Layer (BBL) loss

The blade boundary layer loss is associated with the formation of boundary layer on the blade surfaces which are well away from the endwalls.

$$\zeta_{BBL} = 2 \sum \frac{C_s}{p \cos \alpha_{ref}} \int_0^1 C_d \left(\frac{V_o}{V_{ref}} \right)^3 d(x/C_s) \quad (2)$$

where the summation is for both blade surfaces, x is surface distance(m), p is the total length of the surface(m), C_d is blade pitch(m), C_s is the dissipation coefficient (dimensionless entropy production rate), V_o is blade surface velocity(m/s) and V_{ref} (m/s) and α_{ref} are reference velocity and flow angles, which would be based on inlet for compressor and exit for turbine.

Sub-models of Blade Boundary Layer Loss Model:

A. BBL Denton Model:

Here the tuning parameter has been given a fixed value and rectangular loading has been used. The numerical efforts in calculating the loss are reduced to a great extent and the loss formulation assumes a very simplified expression:

$$\zeta_{BBL} = C_d * \left(2 * \frac{\bar{V}}{\Delta V} + 6 * \frac{\Delta V}{\bar{V}} \right) * (\tan(\alpha_2) - \tan(\alpha_1)) \quad (3)$$

Here, α (rad) is the flow angle, V is mean velocity(m), ΔV (m) is the magnitude by which the pressure and suction velocity differ from the mean velocity.

B. BBL Dambach Model:

Here, piecewise linear loading distribution is used and other conditions are same as in previous model.

C. BBL Laminar Dambach Model:

Variation in dissipation coefficient (C_d) is considered here with the Reynolds number based on momentum boundary layer thickness (R_{θ}). Piecewise linear loading distribution is used. Laminar flow is assumed over the pressure surface while a realistic flow is considered over the suction surface.

Approximate method due to Von Karmann and K. Pohlhausen [(Schlichting, 1951)] is used for calculating the boundary layer thickness on the blade surface. The point of instability (transition) is assumed at the point of maximum velocity. This assumption is justified in following cases- (Schlichting, 1951) explains the conditions for which flow separation takes place with the help of velocity distribution over the suction surface of an aerofoil. After attaining maximum velocity there are three cases possible

with $\left(\frac{d^2 U}{dx^2} \right)$ value [(Schlichting, 1951)]

(i) $\left(\frac{d^2 U}{dx^2} \right) > 0$ - Flow separation may be avoided

(ii) $\left(\frac{d^2 U}{dx^2} \right) < 0$ - Flow separation will take place

(iii) $\left(\frac{d^2 U}{dx^2} \right) = 0$ - Flow separation will take place

Dambach case is an example of case (iii), thus we come to the conclusion that the flow would start separating once it crosses the point of maximum velocity.

2.1.2 Trailing edge loss

This loss is associated with wake formation downstream of the trailing edge. Blade boundary layer loss and trailing edge loss are often merged together and termed as *profile loss*.

$$\zeta_{TEL} = -\frac{C_{pb} t}{w} + \frac{2\theta}{w} + \left(\frac{\delta^* + t}{w} \right)^2 \quad (4)$$

where $w(m)$ is sum of throat width $d_o(m)$ and trailing edge thickness $t(m)$, $\theta(m)$ is momentum boundary layer thickness and $\delta^*(m)$ is displacement boundary layer thickness. C_{pb} is a dimensionless base pressure coefficient which is a measure of the pressure acting over the base of the trailing edge with respect to a reference pressure.

Sub-models of Trailing Edge Loss Model:

A. TEL Flat Plate Model:

In this model, both the surfaces are assumed to be flat and flow condition is considered to be turbulent on both the sides. This being a very crude assumption, it cannot be relied upon. It has

been included just for the sake of completeness of the loss model classification. Blassius flat plate correlation is used here in order to find the boundary layer thickness at the trailing edge [(White)]

Laminar Boundary Layer Thickness:

$$\frac{\delta^*}{x} = \frac{1.72}{\sqrt{Re_x}} \frac{\theta}{x} = \frac{0.664}{\sqrt{Re_x}}$$

Turbulent Boundary Layer Thickness:

$$\frac{\delta^*}{x} = \frac{0.048}{(Re)^{1/5}} \frac{\theta}{x} = \frac{0.037}{(Re)^{1/5}}$$

B. TEL Laminar Dambach Model:

Surfaces are not assumed to be flat. Laminar flow condition is assumed over the pressure surface. Suction surface is assumed to be flat after the point of transition and Blassius correlation has been used for the flat plate portion.

2.1.3 Endwall loss

Often termed as secondary loss, this loss is associated with secondary flow generation when the endwall boundary layers pass through a blade row and roll up into large secondary flow vortices. It is difficult to separate this loss from other losses because of strong interaction between the endwall boundary layer and leakage flows as well as the flow downstream of the trailing edge

$$\zeta_{EBL} = \frac{0.5 T_2}{\dot{m} V_{ref}^2} \int_0^1 \frac{C_d}{T} \frac{(V_s^4 - V_p^4)}{(V_s - V_p)} \rho w d(x/C_x) \quad (5)$$

where $x(m)$ is axial distance, C_x is axial chord(m), $w(m)$ is local suction to pressure surface gap, C_d is dissipation coefficient, $\rho \left(\frac{kg}{m^3} \right)$ is static density, $T(K)$ is static temperature, $\dot{m} \left(\frac{kg}{s} \right)$ is mass flow rate, $V_s \left(\frac{m}{s} \right)$ and $V_p \left(\frac{m}{s} \right)$ are suction and pressure side surface velocities and $V_{ref} \left(\frac{m}{s} \right)$ is as defined earlier

Sub-models of Endwall Boundary Layer Loss Model:

A. Endwall Denton Model:

Tuning parameter is fixed and rectangular blade loading distribution is used

B. Endwall Dambach Model:

Tuning parameter is kept fixed and piecewise linear loading distribution is used.

2.1.4 Tip leakage loss

This loss is associated with leakage flow over the tip and hub clearances and highly depends on whether the blades are shrouded or unshrouded.

$$\zeta_{TLL} = \frac{2 C_{dis} g C}{h p \cos \alpha_{ref}} \int_0^1 \left[\left(\frac{V_s}{V_{ref}} \right)^3 \left(1 - \frac{V_p}{V_s} \right) \cdot \sqrt{1 - \left(\frac{V_p}{V_s} \right)^2} \right] d(z/C) \quad (6)$$

where $g(m)$ is tip clearance, C_{dis} is discharge coefficient, $h(m)$ is blade span, $C(m)$ is true chord, $z(m)$ is distance along blade chord and p, V_s, V_p, V_{ref} and α_{ref} are as defined earlier. C_{dis} is used to determine the local rate of leakage flow together with the static pressure difference across the blade tip.

Sub-models of Tip Leakage Loss Model:

A. TLL Denton Model:

Tuning parameter is fixed and rectangular blade loading distribution is used

B. TLL Dambach Model:

Tuning parameter is kept fixed and piecewise linear loading distribution is used.

Von Karmann and Pohlhausen method [(Schlichting, 1951)] is applied to a general problem of a two-dimensional boundary layer with pressure gradient. The starting point of the method is to use the momentum-integral equation. By assuming a suitable velocity profile the equation allows us to calculate for the momentum thickness, the displacement thickness and the shearing stress at the wall. The velocity profile that the method assumes takes into account the no-slip condition at the wall surface, the continuity condition, positive and negative pressure gradients, and the flow separation condition. The method assumes a polynomial of the fourth degree for the velocity function which is as follows,

$$\frac{u}{U} = f(\eta) = a\eta + b\eta^2 + c\eta^3 + d\eta^4 \quad (7)$$

With the application of relevant boundary conditions and exhaustive calculations explained in [(Schlichting, 1951)], boundary layer thicknesses can be calculated. These are then used in the loss models to calculate the overall losses. Main limitations of this method are that it can be only used for laminar flows and cannot be used for flows with separation. Thus, certain practical assumptions are needed in order to estimate the boundary layer thickness at the end of the trailing edge.

2.2 Meanline Solver – TURBOMEAN

The meanline solvers help in predicting the performance of an axial-flow turbine. These prediction methods can be broken down into two groups, according to Sieverding [(Earl Logan, 2003)]: the first one requires very limited geometrical information while the other group requires detailed geometrical information to carry out velocity triangle calculations. The first group forms the 'Design Mode' and the second group forms the 'Analysis Mode' of the meanline solver. 'Design mode' consists of preliminary turbine sizing practices where aerodynamic parameters are taken as input and the designer uses Smith's chart to have input target efficiency. Designer has no idea about the geometry of turbine section except for overall space constraints.

'Analysis mode' does rigorous calculations for the loss components using aerodynamic parameters to obtain an accurate estimate of efficiency. It is more likely a designer would know about the geometric constraints of turbine section beforehand or otherwise he would use a meanline solver of the first group type to have an initial guess on geometry. Analysis mode is better suited for more detailed turbine design studies and accurate performance predictions.

MEANGEN falls in first group while the meanline solver in present work, TURBOMEAN, falls in second group. TURBOMEAN requires an initial stick flow-path geometry which is then divided into stations where all thermodynamic calculations take place. The location of these stations depends on the critical points in the geometry such as upstream and downstream of stator blade row, etc. As shown in Figure 6 Integrated TurboMean Working, TURBOMEAN takes geometrical, aerodynamic and thermodynamic parameters as input and performs velocity triangle calculation and thermodynamic property calculation [(P.K.Nag, 2008)] at stage inlet, stage exit, and stator-rotor interface station for each stage. The solver makes no simplifying assumptions like MEANGEN but rather takes an initial guess of total-to-static efficiency as input and iteratively

solves forefficiency using the integrated loss models until convergence. With the optimum efficiency, it calculates for flow angles, velocities, pressure, temperature, density, entropy at intermediate stations. Both stagnation and static properties are calculated by the solver [(J.H.Horlock, 1966)].

Loss model requires flow angles as inputs to calculate the overall loss coefficient for an individual blade row. The meanline solver does take an initial guess on efficiency as an input and calculates the flow angles. Flow angles calculated are then put into the loss model, which give a modified efficiency as an output. This efficiency will be then used as an input for the meanline solver and thus, this loop will continue to operate till it converges on efficiency. This process takes place for each blade row and does not proceed to the successive rows until it converges for the previous blade row.

The code works in such a way that it assumes a blade loading distribution to get started with the iterative process. In the first stage of design, the designer is not aware of the 2-D blade shape and hence he has no idea about the pressure distribution over the blade surfaces. But, as an analysis mode of meanline solver requires it as an input, an assumption on the blade loading distribution gives just a headstart for the solver. This loading distribution is parametrized and changes after every iteration with the help of loss models. After the user gives an input of target efficiency, the system solves for an optimal efficiency using the loss models. Having a knowledge of the magnitude of each component losses occurring in the machine, the designer can take appropriate measures to decrease these losses. Measures can be altering the geometry of the turbine section, or adding another stage, etc.

Advantages of the code have been discussed aplenty in the previous sections. Some downsides of the code is that it does not have any analytical or numerical method to solve for turbulence boundary layer thickness. Thus, it relies on some assumptions so as to estimate the boundary layer thickness after the transition. The loss model is applicable for transonic flow operation and does not include shock losses. Another loss component can be added to take shock losses into consideration.

3. Result and Discussion

An important task to be done before integrating the loss model code and TURBOMEAN was to validate them separately. Loss model code was validated against literature given in [(J.D.Denton, 1993)]. The loading distribution used was same as that of in the research paper. The validation of the codes was done so as to have an idea about the effects of assumptions that were taken for simplified calculation. A proper validation would then imply that the assumptions taken are practical and are not too harsh to neglect the real flow situation.

As an open-source analysis meanline solver was not available, TURBOMEAN was validated against the MEANGEN. As these two meanline solvers belong to different types, validation was carried out by using the geometry calculated by MEANGEN for a given set of inputs, as an input to TURBOMEAN. TURBOMEAN then would solve for the flow angles which were then compared with the MEANGEN. Geometry constructed by MEANGEN has all the simplifying assumptions such as constant axial velocity, etc. So, if the same geometry is used for TURBOMEAN, then the output should confirm this assumption which we see in the validation tables. Sections below explain the details of the validation and discuss their outcomes.

3.1 Validation of Loss Models

The loss models are implemented in MATLAB with flexibility to choose between Denton's and Dambach's blade loading distributions. This allows generating plots of each component loss model as a function of inlet and exit flow angles for a given blade loading distribution, geometrical parameters such as $C_r, C_x, C, p, h, d, \sigma, t$ and g and inlet and exit flow conditions. Validation of loss models are shown in Figure 2 Blade Boundary Layer Loss Model Comparison, Figure 3 Tip Leakage Loss Model Comparison, and Figure 4 Endwall Loss Model Comparison. Reference graph from the literature is used to compare with those generated from the MATLAB code. The trend of the graphs generated through MATLAB code align with those given in the literature. The sub-models with more relevance to the flow-physics predict higher losses than those sub-models with simplified assumptions and rectangular blade loading distribution. As for example, in Figure 2 Blade Boundary Layer Loss Model Comparison, for the same set of flow angles, Figure 2 Blade Boundary Layer Loss Model Comparison predicts higher loss coefficient than Figure 2 Blade Boundary Layer Loss Model Comparison. It is interesting to see that the effect of piece-wise lin loading distribution is quite nominal for constant dissipation coefficient C_d . In some loss models,

2a. BBL Reference Graph

2b. BBL Denton

2c. BBL Dambach

2d. BBL Laminar Dambach

Figure 2: Blade Boundary Layer Loss Model Comparison

3a. TLL Reference Graph

3b. TLL Denton

3c. TLL Dambach

Figure 3: Tip Leakage Loss Model Comparison

the tuning parameter is not assigned a particular constant value unlike others, and is taken as a function of a flow parameter (Re_θ). The real impact of the piece-wise loading distribution can be seen when C_d is calculated as a function of momentum thickness based Reynolds number using Pohlhausen family of velocity profiles [(Schlichting, 1951)]. This avoids overestimation of loss as can be seen from Figure 2 Blade Boundary Layer Loss Model Comparison. For the same set of flow angles, Figure 2 Blade Boundary Layer Loss Model Comparison predicts lesser loss

coefficient than Figure 2 Blade Boundary Layer Loss Model Comparison. In order to avoid the extra effort in performing the mathematics, the tuning parameter is given a suitable value after conducting many experiments, which usually overestimates the loss occurring in the turbomachine. It is evident from comparing Figure 2 Blade Boundary Layer Loss Model Comparison and Figure 2 Blade Boundary Layer Loss Model Comparison.

4a. EBL Denton

4b. EBL Dambach

Figure 4: Endwall Loss Model Comparison

5a. Single Stage Axial Turbine Flowpath

5b. Two-Stage Axial Turbine Flowpath

Figure 5: Axial Turbine Flowpaths

Figure 6: Integrated TurboMean Working

3.2 Validation of Meanline Solver-TURBOMEAN

Validation of TURBOMEAN without the loss model has been done against MEANGEN which is a 'Design Mode' meanline solver. The validation has been done for two cases, a single stage axial flow turbine and a two-stage axial flow turbine. The geometry of flowpath is shown in Figure 5 Axial Turbine Flowpath, for both the cases is generated with the help of MEANGEN. Accordingly, the geometry inputs and other relevant inputs are fed to TURBOMEAN. The same efficiency is used for both the solvers. Properties such as flow angles, density, Mach number, entropy, axial velocity, pressure, and temperature are compared at respective stations. These are tabulated in Table 1 TurboMean Validation for Single Stage Axial Turbine and Table 2 TurboMean Validation for 2-stage Axial Turbine. We observe for both the tables the error percentage is very less and is within the accepted tolerance.

Table 1 TurboMean Validation for Single Stage Axial Turbine

Meanline Solver Validation of single stage axial turbine				
Properties		Meangen	Turbo Mean	Error (%)
Flow Angles	ALPHA2	74.225	73.064	-1.5642
	ALPHA3	-40.695	-41.228	1.3097
	BETA2	57.002	54.840	-3.7928
	BETA3	-70.727	-70.820	0.1304
Density (kg/m3)	RHO1	0.9391	0.9469	0.8306
	RHO3	0.6439	0.6434	-0.0776
Mach Number	MABS1	0.2466	0.2102	-14.7324
	MABS2	0.7146	0.7155	0.1301
	MREL2	0.3567	0.3619	1.4718
	MREL3	0.6000	0.6037	0.6216
Entropy (J/kg.K)	S3	14.7708	14.7844	0.0920
Axial Velocity (m/s)	VM1	126.926	125.900	-0.8083
	VM2	126.926	136.160	7.2751
	VM3	126.926	127.020	0.0740
Pressure (bar)	PS3	1.9473	1.9457	-0.0822
Temperature (K)	TS3	1053.6350	1053.5000	-0.0128

Absolute flow angles are denoted by 'ALPHA' followed by the station number which is appropriately shown in Figure 5 Axial Turbine Flowpath. Relative flow angles are similarly denoted by 'BETA'. Static density is denoted by 'RHO'. Absolute and relative Mach numbers are denoted by 'MABS' and 'MREL' respectively. 'VM' denotes axial velocity in the turbomachine and, 'PS' and 'TS' denote static pressure and static temperature respectively.

TURBOMEAN integrated with the loss model enables design space exploration for maximising the efficiency. The current work considers transonic flow condition where shock losses are neglected. The shock losses can very well be added to increase the scope of implementation.

4. Conclusions

In the present work, the loss models developed by Denton [(J.D.Denton. 1993)] have been extended by introducing a more realistic piece-wise linear blade loading distribution as well as momentum thickness boundary layer based calculation of the dissipation coefficient. The loss models are first validated against data available in [(J.D.Denton. 1993)] and then integrated in an in-house meanline solver TURBOMEAN. TURBOMEAN has been further validated against open source meanline solver MEANGEN [(J.D.Denton,2017)]. With the separate validation of both loss models and meanline solver, the results obtained from incorporating both of them will reduce the reliability on CFD for the preliminary design of the flowpath.

Table 2 TurboMean Validation for 2-stage Axial Turbine

Meanline Solver Validation of 2-stage Axial Turbine				
Properties		Meangen	Turbo Mean	Error (%)
Flow Angles	ALPHA2	74.225	73.064	-1.5642
	ALPHA2	65.678	64.944	-1.1179
	ALPHA3	-28.258	-28.490	0.8210
	ALPHA4	65.678	65.001	-1.0309
	ALPHA5	-28.258	-28.362	0.3680
	BETA2	43.908	42.787	-2.5531
	BETA3	-60.775	-60.742	-0.0543
	BETA4	43.905	42.887	-2.3186
Density (kg/m3)	BETA5	-60.775	-60.661	-0.1876
	RHO1	0.9137	0.9134	-0.0285
	RHO3	0.6243	0.6235	-0.1281
Mach Number	RHO5	0.4054	0.4047	-0.1727
	MABS1	0.3413	0.3423	0.2930
	MABS2	0.7587	0.7602	0.2043
	MABS3	0.3619	0.3651	0.8870
	MABS4	0.8086	0.8105	0.2325
	MREL2	0.4337	0.4387	1.1529
	MREL3	0.6530	0.6565	0.5360
	MREL4	0.4622	0.4674	1.1250
Entropy (J/kg.K)	MREL5	0.6979	0.7017	0.5388
	S3	14.7708	14.7844	0.0921
Axial Velocity (m/s)	S5	31.5867	31.6163	0.0937
	VM1	203.0828	203.6400	0.2744
	VM2	203.0828	209.2100	3.0171
	VM3	203.0828	204.3600	0.6289
	VM4	203.0828	208.8300	2.8300
Pressure (bar)	VM5	203.0828	204.8500	0.8702
	PS3	1.8676	1.8648	-0.1499
Temperature (K)	PS5	1.0618	1.0597	-0.1978
	TS3	1042.1393	1041.8000	-0.0326
	TS5	912.4437	912.0700	-0.0410

Acknowledgments

The authors are greatly indebted to the funding support provided by MHRD and Ministry of Power, Government of India under IMPRINT project number 6042. Further, Srinivas Diwanji acknowledges the financial support received under IIT Kanpur SURGE summer internship program.

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