IOP Conf. Series: Materials Science and Engineering **302** (2018) 012038 doi:10.1088/1757-899X/302/1/012038

Mean-line Modeling of an Axial Turbine

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Abstract. The article describes the approach for axial turbine modeling along the mean line. It bases on the developed model of an axial turbine blade row. This model is suitable for both nozzle vanes and rotor blades simulations. Consequently, it allows the simulation of the single axial turbine stage as well as a multistage turbine. The turbine stage model can take into account the cooling air flow before and after a throat of each blade row, outlet straightener vanes existence and stagger angle controlling of nozzle vanes. The axial turbine estimation method includes the loss estimation and thermogasdynamic analysis. The single stage axial turbine was calculated with the developed model. The obtained results deviation was within 3% when comparing with the results of CFD modeling.

1. Introduction

The turbomachine development is very complex process that includes thermogasdynamic, structural, technological and economic investigations. Thermogasdynamic designing is iterative process, in which the calculations results performed at the conceptual design stage are subsequently improved with application the higher fidelity models [1, 2]. During the conceptual design stage the simple models such as the model of the one-dimensional estimation along the flow path mean line is used for performance prediction and preliminary optimization of the turbomachine design. These models are very important, because they allow fast estimation of velocity triangles, losses and parameters of the blades at mean line with acceptable accuracy.
In most cases stations of the turbine flow path are normal to the axis of the engine [3, 4]. This paper presents the method in which these stations are normal to the mean line (figure 1). Thus the radial velocity component can be excluded from consideration. The main advantage is capability for both axial and radial turbines modeling.

2. One-dimensional estimation of an axial turbine

2.1. Model description

The one-dimensional (1D) mathematical model is reasonable for the turbomachines performance calculation along the flow path mean line. This model (figure 2) describes the relation between thermodynamic and kinematic parameters and geometric parameters of blade row at various operation conditions.

A model of the axial turbine stage (figure 3) can be draw up using the turbine blade row model. The model of the axial turbine stage also includes the calculation elements of nozzle vanes, blades, stage overall performance as well as inlet and outlet data interface elements for an external model. The turbine stage model can take into account the cooling air flow before and after a throat of each blade row, outlet straightener vanes existance and stagger angle controlling of nozzle vanes, etc.

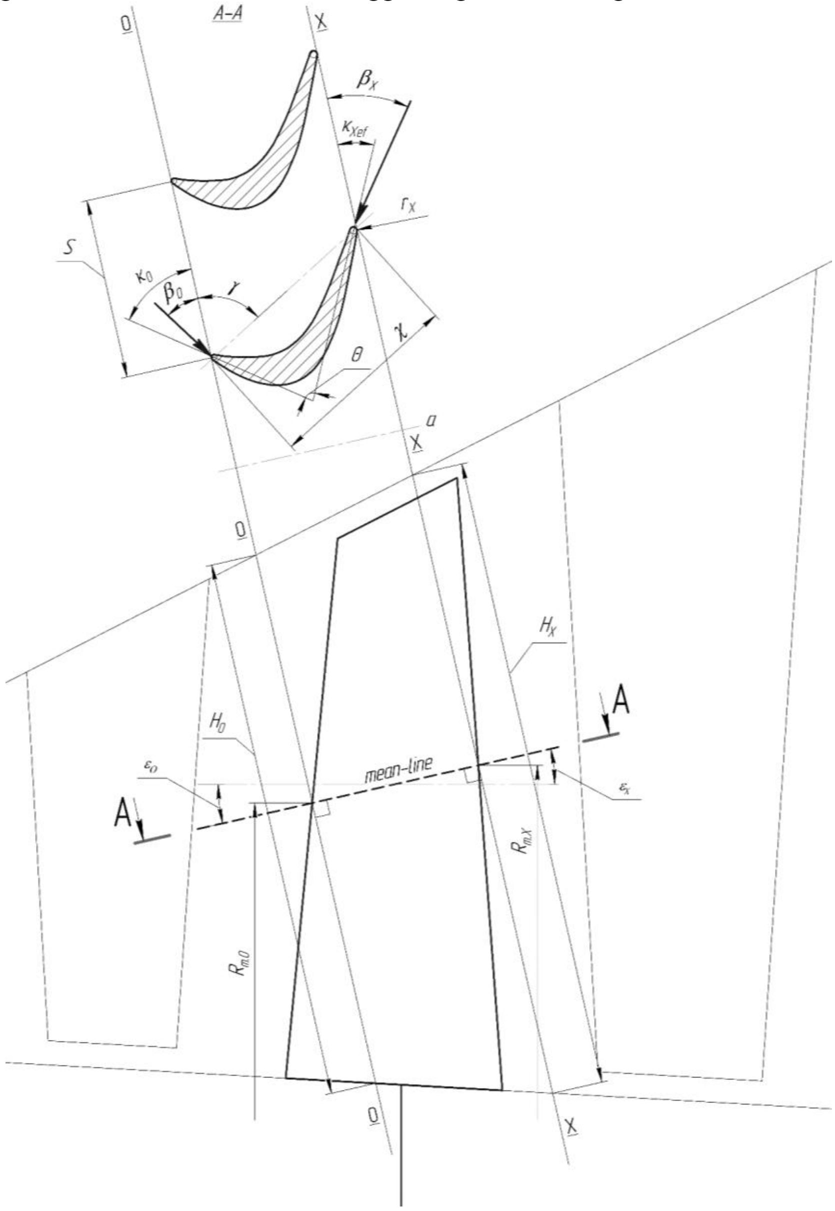

Figure 1. Meridional cross section of a turbine blade row.

2.2. Method for loss estimation in an axial turbine blade row

Loss coefficients at turbine blade row reflect the deviation of real process of the gas expansion from its ideal (isoentropic) instance. Losses in a turbine blade row are usually expressed in a velocity coefficient or a thermodynamic loss coefficient. The velocity coefficient φ represents deviation of the gas flow real relative velocity at blade row outlet from ideal velocity at the same pressure ratio.

The thermodynamic loss coefficient ζ represents loss of enthalpy:

$$\zeta = \frac{i_X - i^*_{Xs}}{i^*_{wX} - i_X} \tag{1}$$

where i_X – static enthalpy at outlet from blade row;
i_{Xs} – total isoentropic process enthalpy at outlet from blade row;
i^*_{wX} – total relative flow enthalpy at outlet from blade row.

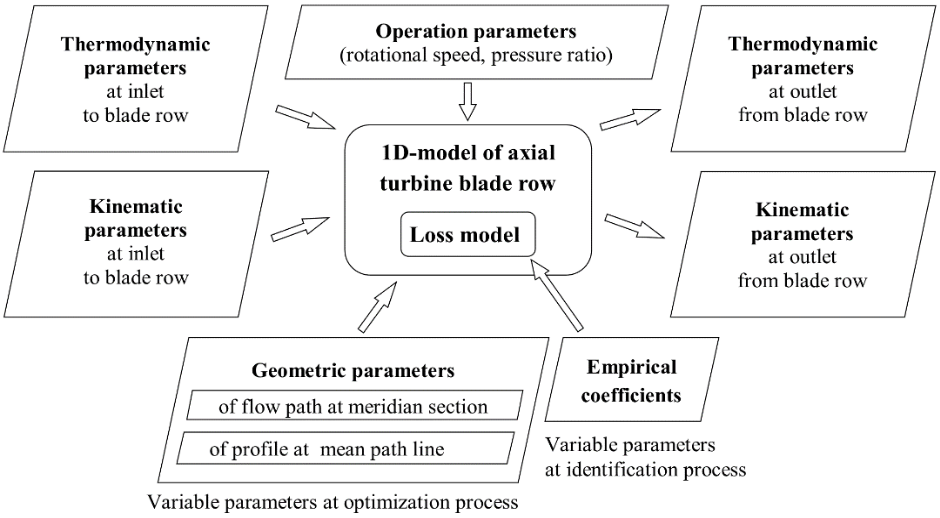

Figure 2. Data streams of the 1D-model of an axial turbine blade row.

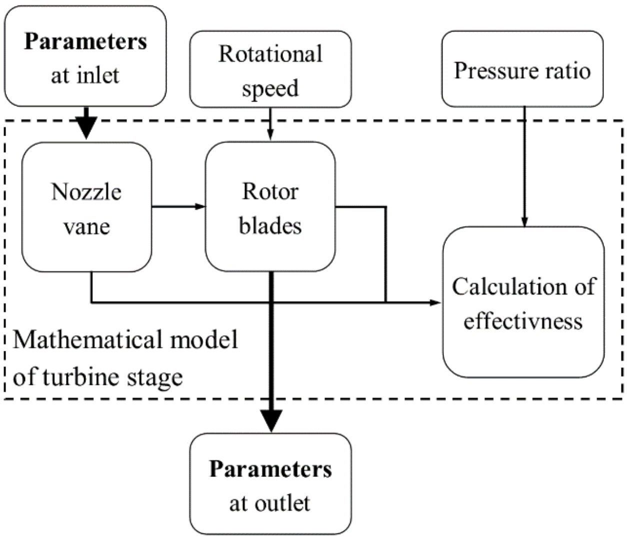

Figure 3. Turbine stage model.

The velocity coefficient is related with the thermodynamic loss coefficient:

$$\varphi = \sqrt{1-\zeta} \tag{2}$$

Total loss has complex structure and includes profile, secondary and cooling losses:

$$\zeta = \zeta_{pr.i.W} + \zeta_s + \zeta_c \tag{3}$$

The structure of the total loss is represented in figure 4, where $\overline{G_a}$ – relative cooling air mass flow rate, K_{KW}, K_V, K_ι – auxiliary coefficients. The cooling losses and other kind of losses are described in detail in [5, 6, 7].

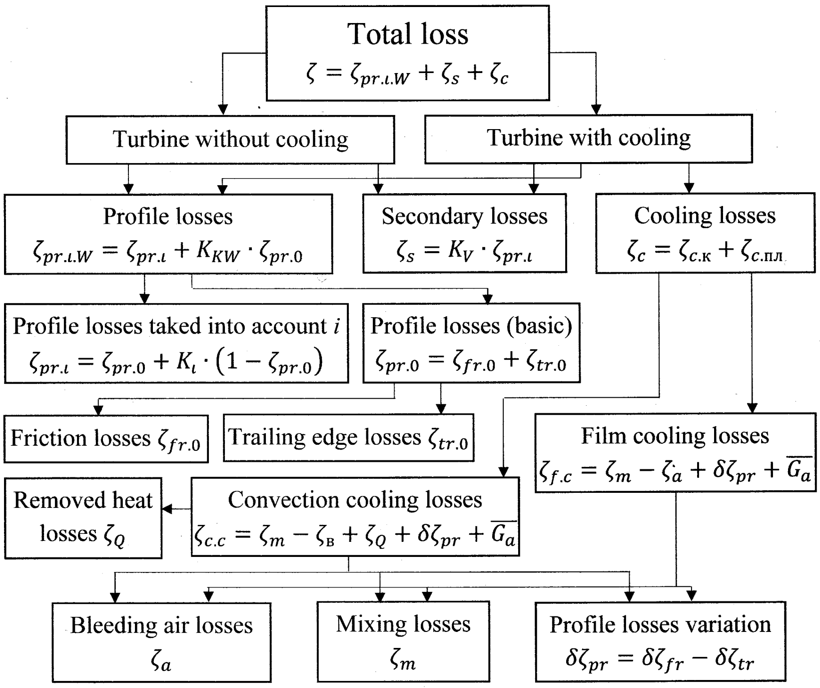

Figure 4. The structure of total loss taked into account in the 1D-model of an axial turbine blade row.

The function for determination of the loss coefficient ζ can be presented in the following general form:

$$\zeta = f(\{G\}, p_O, T_O, T_X, V_X, \lambda_{WX}, \iota, \kappa_O, \kappa_{X_{ef}}, H_X, \chi, s, r_X, \zeta_c), \tag{4}$$

where G – working fluid parameter array;
p_O, T_O – static pressure and static temperature at blade row inlet;
T_X – static temperature at blade row discharge;
V_X – absolute velocity at discharge;
λ_{WX} – corrected relative velocity at discharge;
ι – incidence angle;
κ_O, $\kappa_{X_{ef}}$ – blade inlet angle and efficient blade outlet angle;
H_X – blade height at discharge;
χ – chord profile length;
s – blade spacing;
r_X – trailing edge radius;
ζ_c – cooling losses coefficient.

2.3. Method for the thermogasdynamic analysis of an axial turbine blade row

The method for the thermogasdynamic analysis of an axial turbine blade row (nozzle vane as well as rotor blade) along the flow path mean line allows to calculate the gas flow parameters at inlet and discharge stations and the cycle parameters depending on the gas flow parameters at inlet, rotational speed and M_{CX} parameter. The M_{CX} parameter describes the flow state at outlet.

Since the methods of the loss coefficients estimation and thermogasdynamic analysis needs the detail description, they will be published in future papers.

3. Model validation

The software modules for computer-aided environment for thermogasdynamic calculations and analysis ASTRA [8] were implemented to perform validation of proposed models. The testing was performed on high pressure turbine of the small-scale engine. The results of calculation are shown in the table 1.

Table 1. Results of the test calculation

G_O	kg/s	2,527	Gas flow rate
p^*_O	kPa	332,526	Total pressure at inlet to nozzle vane
T^*_O	K	1175	Total temperature at inlet to nozzle vane
n	rpm	40000	Rotational speed
$\varepsilon_{O.nv}$	°	0	Taper angle at inlet to nozzle vane
$R_{m.O.nv}$	m	0,094	Mean line radius at inlet to nozzle vane
$\alpha_{O.nv}$	°	0	Angle of entrance at inlet to nozzle vane
$\varepsilon_{X.nv}$	°	0	Taper angle at outlet from nozzle vane
$R_{m.X.nv}$	m	0,094	Mean line radius at outlet from nozzle vane
$r_{X.nx}$	m	0,001	Trailing edge radius of nozzle vane
$\kappa_{O.nv}$	°	0	Blade inlet angle of nozzle vane
$\kappa_{X.ef.nv}$	°	-66	Efficient blade outlet angle of nozzle vane
$\varepsilon_{X.i}$	°	0	Taper angle at outlet from rotor blade
$R_{m.X.i}$	м	0,09375	Mean line radius at outlet from rotor blade
$r_{X.i}$	м	0,00035	Trailing edge radius of rotor blade
$\kappa_{O.i}$	°	23	Blade inlet angle of rotor blade
η	—	0,8828	—
N	kW	431,1	—
Nozzle vane			
ζ_c	—	0	Cooling losses coefficient
ζ_{fr}	—	0,028	Friction losses coefficient
$\zeta_{pr.0}$	—	0,063	Profile losses coefficient (basic)
$\zeta_{pr.i}$	—	0,063	Profile losses coefficient taked into account the angle of incidence
$\zeta_{pr.i.W}$	—	0,0634	Profile losses coefficient taked into account the angle of incidence and flow velocity
ζ_{tr}	—	0,035	Trailing edge losses coefficient
ζ_s	—	0,048	Secondary losses coefficient
$\zeta_{pr.i.th}$	—	0,028	Coefficient of the profile losses before the throat area taked into account the angle of incidence
$\zeta_{pr.i.W.th}$	—	0,0282	Coefficient of the profile losses before the throat area taked into account the angle of incidence and flow velocity
$\zeta_{s.th}$	—	0,0214	Coefficient of the secondary losses before the throat area
ζ_{th}	—	0,0496	Losses coefficient at the throat area
ζ	—	0,1114	Total losses coefficient
φ	—	0,9426	The velocity coefficient
π	—	1,5343	Pressure ratio p^*_O/p_X
π*	—	1,046	Pressure ratio p^*_O/p^*_X
Rotor blade			
ζ_c	—	0	Cooling losses coefficient
ζ_{fr}	—	0,042	Friction losses coefficient
$\zeta_{pr.0}$	—	0,059	Profile losses coefficient (basic)
$\zeta_{pr.i}$	—	0,0603	Profile losses coefficient taked into account the angle of incidence
$\zeta_{pr.i.W}$	—	0,063	Profile losses coefficient taked into account the angle of incidence and flow velocity
ζ_{tr}	—	0,017	Trailing edge losses coefficient
ζ_s	—	0,0315	Secondary losses coefficient

$\zeta_{pr.i.th}$	—	0,0433	Coefficient of the profile losses before the throat area taked into account the angle of incidence
$\zeta_{pr.i.W.th}$	—	0,0452	Coefficient of the profile losses before the throat area taked into account the angle of incidence and flow velocity
$\zeta_{s.th}$	—	0,0226	Coefficient of the secondary losses before the throat area
ζ_{th}	—	0,0678	Losses coefficient at the throat area
ζ	—	0,0945	Total loss coefficient

Basing on presented results it can be concluded that model provides the adequate solution and stable convergence in the numerical solution process. The obtained results deviation was within 3% when comparing with the results of CFD modeling.

4. Conclusions

The blade row model of the one-dimensional estimation along the mean line was developed for more detailed analysis and preliminary optimization of an axial turbine. The blade row model allows to make up a single axial turbine stage as well as a multistage turbine. The turbine stage model can take into account the cooling air flow before and after a throat of each blade row, outlet straightener vanes existence and stagger angle controlling of nozzle vanes.

In the presented method stations are normal to the mean line. It simplifies the computation algorithm because the radial velocity component may be excluded from consideration. The main advantage is capability for both axial and radial turbines modeling.

It can be concluded that proposed model provides the adequate solution and stable convergence in the numerical solution process. The detailed comparing of the 1D and 3D CFD modelling results will be possible after models integration and creation of an identification unit.

Acknowledgments

This work was supported by the Ministry of education and science of the Russian Federation in the framework of the implementation of the Program of increasing the competitiveness of Samara University among the world's leading scientific and educational centers for 2013-2020 years.

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