COMPARATIVE ANALYSIS OF ONE AND TWO-STAGE AXIAL IMPULSE TURBINES FOR LIQUID PROPELLANT ROCKET ENGINE

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Abstract

Brazilian Space Agency (AEB) intends to develop a vehicle for launching communication and/or meteorological satellites, using liquid propellant rocket engine technology. To numerically investigate the turbine design that best equip that engine, it is developed a computer program, using the software "Mathcad", to analyze and compare one and two-stage axial impulse turbines for defining the option that maximize the turbine efficiency. The algorithms for one and two-stage turbines and the loss model were written based on the works of Russian researches as Ovsyannikov, that used their experience in turbine design and manufacture at Soviet aerospace industries for presenting many empirical relations and data for safety coefficients and loss and efficiency factors. In this context, the algorithm for one-stage axial impulse turbine was used to validate the methodology and the computer program by applying actual design data of the one-stage turbine of the Russian engine RD-0109 from Engine Atlas of the Moscow Aviation Institute.

Keywords: Turbine, Turbopump, Liquid rocket engine, Loss model, Partial admission.

1. INTRODUCTION

Brazilian Space Agency (AEB) intends to develop a vehicle for launching communication and/or meteorological satellites to low orbits, in a first phase of the development program and geosyncronous transfer orbits, in a second phase of the development program.

In this first phase, the vehicle, called Satellite Launch Vehicle 2 (VLS-2), is proposed considering a thrust-chamber assembly, a turbopump unit (TPU) as propellant-feed system, a turbine drive system (heated pressurizing gas from gas accumulator at starting mode and main propellant's combustion products from gas generator at operational mode) and a control system with its elements (valves, regulators, reductors, etc).

In light of the need to numerically investigate the turbine design that best equip the engine, this work is concerned with the implementation of a computer program, using the software "Mathcad", to define the turbine geometry and calculate the consequent gasdynamic performance at design point to justify design decisions as number of stages.

The initial algorithm was written based on the works of Ovsyannikov *et al* (1973), that explain the theory and design of TPU turbines. Starting from basic principles of gas dynamics, it is discussed the basic features of the turbines, the gas flow through the cascades, the main losses, efficiency, energy characteristics, selection of parameters for calculation and jointing operation with TPU pumps. It is derived equations for designing the flow parts of one and two-stage turbines with radial or axial flux. Moreover, the authors used their experience in turbine design and manufacture at Soviet aerospace industries for presenting many empirical relations and data for safety coefficients and loss and efficiency factors.

Tchervakov (1997) presented a method of calculation for two-stage axial impulse turbine in the algorithm shown in Fig. 1. During the calculations, the dimensions of the turbine elements and the flow characteristics (temperature, pressure, speed, etc) are defined according to initial data and gas dynamics parameters at preceding stage for design point.

The combustion product parameters from two-stage reducing gas generator at design point conditions were modelated from empirical test data with kerosene $CH_{1,9423}$ and liquid oxygen O_2, according to Kessaev (1997), allowing the definition of the working fluid parameters in initial data analysis.

2. THEORETICAL FORMULATION – THE LOSS MODEL

According to Ovsienikov *et al*. (1973), the working fluid flowing through the rotor vanes causes the rotor resistant moment M_u. In this context, the power N_u is resulting from the moment M_u produced when the wheel is rotating at shaft rotational speed ω_T. By this way, the turbine power N_T is defined as the power N_u decreased by the power expended in disk and shroud friction losses and partial admission losses. For axial impulse turbines, Ovsienikov *et al*. (1973) suggest the empirical formulation described bellow for definition of the loss factors.

2.1. Disk friction losses.

With the wheel rotation, the fluid (gas) also begins to rotate inside the clearances as a result of friction forces. Thus, power is expended according to the following expression:

$$N_{fr\,disk} = 2\,C_{fr\,disk}\,\rho_1 \left(\frac{D_{av} - h_{bl\,1}}{2}\right)^5 \omega_T^{\,3}. \tag{1}$$

Here, D_{av} is the average diameter of the turbine wheel. Considering that the index 1 and 2 refer to inlet and outlet conditions, respectively, ρ_1 is the fluid density at the rotor inlet and $h_{bl\,1}$ is the inlet blade height. Ovsienikov *et al*. (1973) defined the disk friction loss coefficient $C_{fr\,disk}$ as function of the Reynolds number of the fluid near the disk Re_{disk} taking into account experimental data and considering the following conditions: smooth disk surfaces; fluid angular speed inside the clearance equal to half of ω_T and $Re_{disk} > 10^5$.

2.2. Shroud friction losses.

In axial turbines with large ratio $D_{av}/h_{bl\,1}$, the friction energy expended at shroud external surface is significant. This is the resistance of a rotating cylinder inside another cylinder. Then, the power expended on shroud friction losses is:

$$N_{fr\,shr} = C_{fr\,shr}\,\rho_1\,b_{shr}\left(D_{av} + h_{bl\,1}\right)^4\,\omega_T^3; \qquad (2)$$

where b_{shr} is the shroud width and $C_{fr\,shr}$ is the shroud friction loss coefficient defined as function of the Reynolds number of the working fluid near the shroud Re_{shr}.

2.3. Partial admission losses

In LRE turbines, the tip leakage would be very high if the working fluid was fed over the entire row, because of the short blades and working fluid low flow rate. Therefore, partial admission is frequently used, in spite of additional energy losses.

Partial admission losses are divided in four groups, according to flow pattern: losses due to sudden expansion on nozzle arc boundaries; losses due to "ejection" of the stagnation gas from the channels, which came from non-operative zone; losses due to blade edge friction against residual gas, backflow moment and residual gas mixing and turbulence along the non-operative zone and losses due to gas leakage in circular direction.

It is difficult to evaluate the effect of each kind of loss experimentally, because they are interconnected, but Ovsienikov *et al*. (1973) proposed the following equation for evaluating the first three groups:

$$N_{\varepsilon} = 0.015\,\rho_1\,\frac{h_{bl\,1}}{D_{av}}\left(1 + 10\,\frac{b_{bl\,1}}{D_{av}}\right)\left(1 - \varepsilon\right)D_{av}^5\,\omega_T^3; \qquad (3)$$

where $b_{bl\,1}$ is the blade width and ε is the admission degree at the rotor inlet.

When the number of group of nozzles i_c increases, losses due to "ejection" and due to sudden expansion increases approximately in the same proportion to i_c and losses due to flowing effects along non-operative zone do not change.

Since these three kinds of losses due to partial admission are commensurable, it is possible to consider that N_{ε} increases $0{,}5\,(i_c + 1)$ times, when the groups of nozzles increase i_c times.

In axial two-stage partial admission turbines, the losses due to partial admission are defined as the sum of losses in each stage.

2.4. Tip losses

The tip losses are mainly determined by the flow and the surface conditions of the casing and the blade tip, because they determine friction losses and, consequently, stagnation conditions, which implies velocity gradient and developed boundary layer over these surfaces. Then, friction losses on blade limiting surfaces causes parasitic vortex flows called pair vortices, which sharply increase when pair vortices are linked to each other at blades with relative height $h_{bl\,1}/b_{bl\,1}$ lower than 1-1.5.

Ovsienikov *et al*. (1973) proposed that tip losses should be considered as a correcting factor η_k of the efficiency η_u, according to the expression:

$$\eta_k = 1 - \frac{0.003}{0.003 + \dfrac{h_{bl1}}{D_{av}}} . \qquad (4)$$

The correct account for the N_T and the turbine efficiency η_T must also consider the leakage $\dot{m}_y$ through clearances between casing and rotor, since the fluid that passes through the cascades is significantly lower than the fluid from the turbine drive system $\dot{m}_T$. The expressions of $\dot{m}_y$ for different types of rotors and labyrinth seals can be found at Ovsienikov *et al*. (1973). Thus, η_T is defined as:

$$\eta_T = \eta_u \eta_k \left(1 - \frac{\dot{m}_y}{\dot{m}_T}\right) - \xi_{fr\,disk} - \xi_{fr\,shr} . \qquad (5)$$

The general expression (5) is used for one-stage axial impulse turbines when η_u is: defined as:

$$\eta_u = 2\varphi^2 \left(\cos(\alpha_1) - \frac{u}{\varphi c_{ad1}}\right)\left(1 + \psi_{av}\frac{\cos(\beta_2)}{\cos(\beta_1)}\right)\frac{u}{\varphi c_{ad1}} \qquad (6)$$

and for two-stage axial impulse turbines when η_u is described by:

$$\eta_u = 2\varphi^2 \left[(1+\psi_{av})\left(1+\psi_{av}^2\right)\left(\cos\alpha_1 - \frac{u}{\varphi c_{ad1}}\right) - (1+\psi_{av})\psi_{av}\frac{u}{\varphi c_{ad1}}\right]\frac{u}{\varphi c_{ad1}} \qquad (7)$$

where u is the tangential speed at turbine wheel mean diameter; α_1 is the absolute flow angle at the rotor inlet; β_1 and β_2 are relative flow angle at the rotor inlet and outlet, respectively; c_{ad1} is the adiabatic absolute speed at the first rotor inlet; φ is the velocity coefficient, which express the hydraulic losses in nozzles and ψ_{av} is the average value of the first rotor velocity coefficient ψ_I, stator velocity coefficient φ_{st} and second rotor velocity coefficient ψ_{II} that express the hydraulic losses in their cascades.

The disk friction loss factor $\xi_{fr\,disk}$, shroud friction loss factor $\xi_{fr\,shr}$ and partial admission loss factor ξ_{ε} are obtained by:

$$\xi_{fr\,disk} = \frac{N_{fr\,disk}}{\dot{m}_T \frac{c_{ad1}^2}{2}} = 0.32\,\frac{C_{fr\,disk}\left(1 - \frac{h_{bl1}}{D_{av}}\right)^5}{\varepsilon \frac{h_{bl1}}{D_{av}} \sin\alpha_1}\left(\frac{u}{c_{ad1}}\right)^3; \qquad (8)$$

$$\xi_{fr\,shr} = \frac{N_{fr\,shr}}{\dot{m}_T \frac{c_{ad1}^2}{2}} = 5.08\,\frac{C_{fr\,shr}\frac{b_{shr}}{D_{av}}\left(1 + \frac{h_{bl1}}{D_{av}}\right)^4}{\varepsilon \frac{h_{bl1}}{D_{av}} \sin\alpha_1}\left(\frac{u}{c_{ad1}}\right)^3; \qquad (9)$$

$$\xi_{\varepsilon} = \frac{N_{\varepsilon}}{\dot{m}_T \frac{c_{ad1}^2}{2}} = 0.076\,\frac{\left(1 + 10\frac{b_{bl1}}{D_{av}}\right)}{\varepsilon \sin(\alpha_1)}\left(\frac{1-\varepsilon}{\varepsilon}\right)\left(\frac{u}{c_{ad1}}\right)^3. \qquad (10)$$

3. NUMERICAL IMPLEMENTATION

In Pereira (1998), the algorithm suggested by Tchervakov (1997), shown in Fig. 1, was used for implementation of a computer program, using the software "Mathcad", release 6.0, to calculate two-stage axial impulse turbine geometries for the designed engine requirements.

Starting from the algorithm for two-stage turbine, Pereira (1998) proposed another algorithm for one-stage turbine by: removing the blocks for calculating the guide cascade and the second rotor parameters and making adjustments for not taking into account the guide cascade and the second rotor parameter contributions in turbine general characteristics as turbine adiabatic specific energy $L_{0\,ad}$; N_T and η_T.

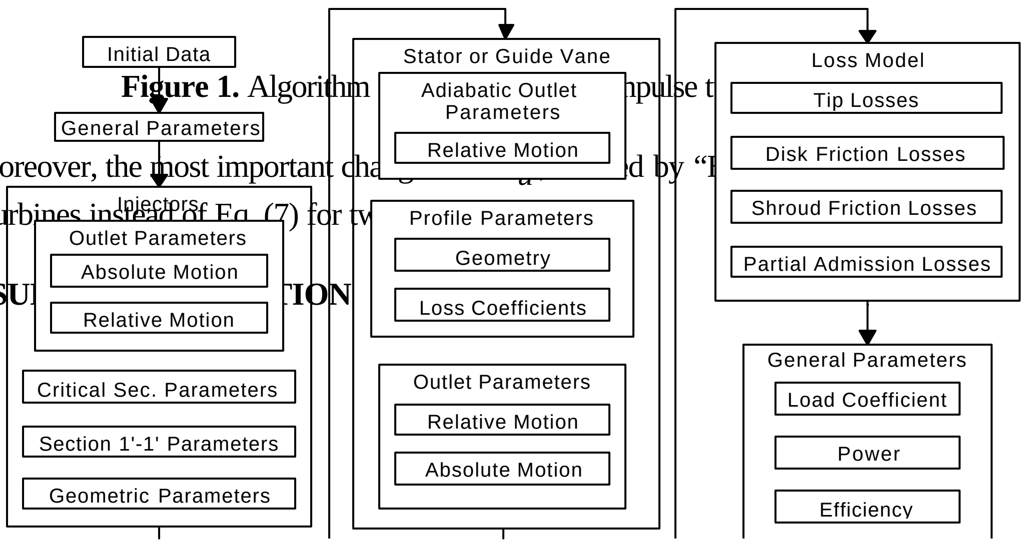

Figure 1. Algorithm ... impulse t...

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4. RESU... TION

The algorithm for one-stage turbine was used to validate the methodology and the computer program by applying actual design data of the one-stage turbine of the Russian engine RD-0109 from Garun *et al*. (1973). Then, the RD-0109 turbine geometry, $\_\ _T$, the design total inlet pressure P□, and total inlet temperature T□ were set as algorithm input data and $\dot{m}_T$, N_T, $\hbar_T$ and the static outlet pressure P_2 from algorithm (output data) were compared with RD-0109 data from Garun *et al*. (1973). The comparative analysis results are described in Pereira (1999) and it is concluded that the methodology is adequate for the design and analysis of a turbine wheel for LRE, because discrepancies are lower than 6.5 %.

For TPU design, it was necessary to do a comparative analysis of one and two-stage turbine characteristics to define the best number of turbine stages. To this aim, both algorithm were used to determine the variation of the main parameters through turbine sections and the dependence of the load coefficient L_u/u^2, $\hbar_T$, $\hbar_u$ and the sum of the loss coefficients □□ on the ratio $u\ /\ c_{ad1}$.

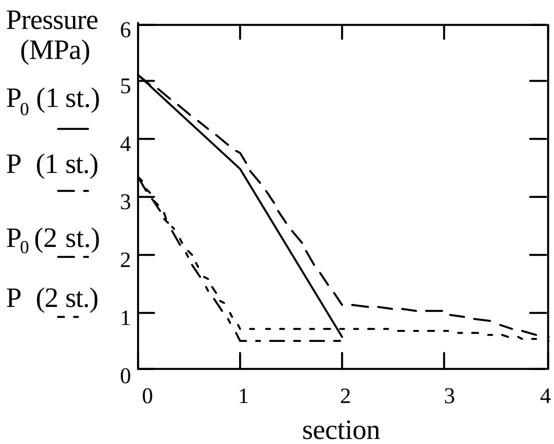

Figure 2. Variation of the total P_0 and the static P pressures through turbine sections.

Figure 2 presents the total P_0 and the static P pressure at the nozzle critical section (section 0), first rotor inlet (section 1), first rotor outlet (section 2), second rotor inlet (section 3) and second rotor outlet (section 4) for one and two-stage turbines. It can be seen that one-stage turbine has higher pressure gradients, which implies more efforts acting on blade root.

Figure 3 shows the dependence of the load coefficient L_u/u^2 on the ratio $u\ /\ c_{ad1}$. One can observe two-stage turbines presents higher L_u/u^2 than one-stage turbine for the same u, which means more specific work L_u available for the same tangential speed u and, consequently, more N_u for the same $\dot{m}_T$, according to Eq. 11:

$$L_u \,□\, N_u/\dot{m}_T\ . \tag{11}$$

Figure 4 depicts the dependence of the $\hbar_T$, $\hbar_u$ and □□ on the ratio $u\ /\ c_{ad1}$. First of all, one can easily observe two-stage turbines have lower □□ than one-stage turbines for the same u /

c_{ad1}. Additionally, two-stage turbines have higher efficiency than one-stage turbines when u / c_{ad1} is lower than 0.25 for the same working fluid (the same gas constant R, adiabatic coefficient k and total inlet temperature T_0). As a result, two-stage turbines reaches the required efficiency at lower u, for the same c_{ad1}, which corresponds to lower n_T and less severe work conditions for bearings, seals, etc.

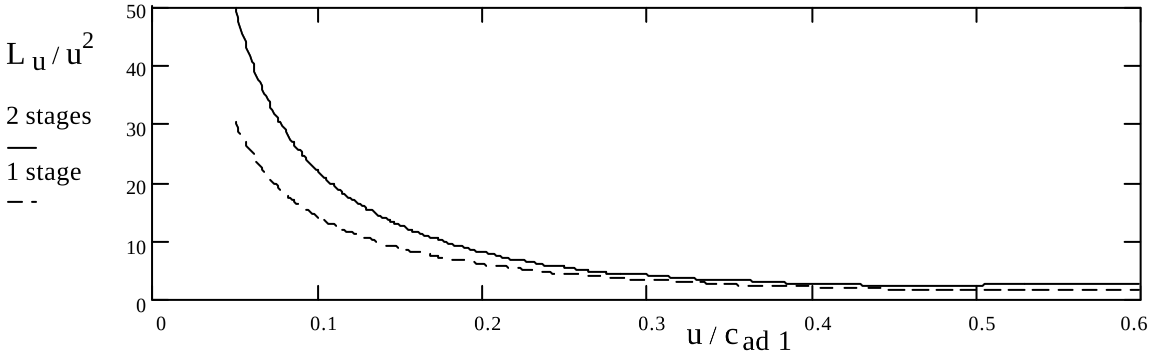

Figure 3. Dependence of the load coefficient L_u/u^2 on the ratio u / c_{ad1}.

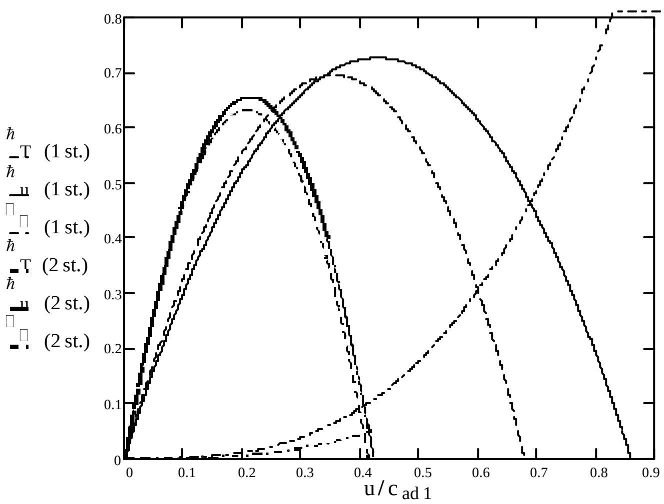

Figure 4. Dependence of the η_T, η_u and [illegible] on the ratio u / c_{ad1}.

Considering the required power to drive oxidizer and fuel pumps is defined as:

$$\sum N_P = N_T = \dot{m}_T L_{0ad} \eta_T = \dot{m}_T \left[\frac{k}{k-1} R T_0 \left(1 - \left(\frac{P_2}{P_0} \right)^{\frac{k-1}{k}} \right) \right] \eta_T , \qquad (12)$$

one can realize higher $\hbar_T$ at the same working fluid conditions (R, k and $T_{\square}$) implies more N_T available to the pumps or to a control device, considering $\square\ N_P$ constant. Otherwise, if it is not necessary to increase N_T, higher $\hbar_T$ means lower $T_{\square}$ or pressure gradient.

5. CONCLUDING REMARKS

As it was pointed before, test data for fully validation the methodology and its results are scarce, the reason clearly understood. Access to actual design and test data was not possible, except for the information contained in the Garun *et al*. (1973) [11]. Nevertheless, personal discussions with Russian researchers that used, and in some extent contributed to the construction of that large amount of information, of the RD-0109 Russian LRE were significant.

As a by-product, a good design tool was also developed, since all the turbine design was carried to produce the turbine geometry that maximize the efficiency. As an example, it was evaluated the ideal number of the stages.

Even considering that the second rotor is responsible for lower part of the total turbine power N_T, it is able to decrease strongly the kinetics energy losses and improve $\hbar_T$.

Therefore, if no satisfactory $\hbar_T$ is reached using one-stage turbine, the two-stage turbine as a possible solution may be investigated using the algorithms as a readily applicable method to compare both performances. Consequently, the two-stage turbine may be an interesting solution to improve performance if its higher height and more complex construction are not relevant.

6. REFERENCES

- GARUN, G. G.; ALEKSEIEV, I. G.; BAULIN, V. I., 1973, “Atlas de construção de motor foguete a propelente líquido (Descrição) Parte II”, Instituto de Aviação Sergo Ordijonikdze / Cátedra 203, Moscou, Rússia, pp. 108-109.
- KESSAEV, J., 1997, “Theory and calculation of liquid propellant engines”, Apostila do Fundamental Course in Engine Design do Moscow Aviation Institute ministrado na ASA-P/IAE/CTA, São José dos Campos, São Paulo, Brasil, pp 127.
- OVSYANNIKOV, B. V.; BOROVSKY, B. Y., 1973, “Theory and calculation of feed units of liquid propellant rocket engines”, (FTD-MT-24-1524-72), manual of the Foreign Technology Division of Wright-Patterson Air Force Base, Ohio, Unites States of America, 764p.
- PEREIRA, F. C. V., 1998, “Diploma project: II Stage liquid propellant rocket engine”, Trabalho de conclusão do Fundamental Course in Engine Design do Moscow Aviation Institute ministrado na ASA-P/IAE/CTA, São José dos Campos, São Paulo, Brasil, 431p.
- PEREIRA, F. C. V., 1999, “Numerical Simulation of a two-stage axial impulse turbines for turbopumps of liquid propellant rocket engine”, Trabalho de conclusão do Mestrado em Ciências do Instituto Tecnológico de Aeronáutica ITA, São José dos Campos, São Paulo, Brasil, pp. 180.

• TCHERVAKOV, V. V., 1997, “Theory and calculation of turbopumps”, Apostila do Fundamental Course in Engine Design do Moscow Aviation Institute ministrado na ASA-P/IAE/CTA, São José dos Campos, São Paulo, Brasil, 259p.