Algorithm of Forming the Appearance of the Flow Path of Turbomachinery of Two-Shaft Aircraft Engine Core

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Abstract: Description of formation process of two-dimensional scheme of flow path of a two-shaft core engine turbomachinery of aviation gas turbine engine is presented. There were three steps in the design. The first step includes rational distribution of specific work and pressure ratio in the core engine between intermediate and high-pressure compressors, as well as the pressure ratio between high and intermediate-pressure turbines. At the second step we select the rotational speed of the high-pressure cascade and determine the main structural-geometrical cascade parameters in the meridional plane. At the third step the rotation frequency of the intermediate-pressure cascade and the main structural-geometric parameters of the intermediate pressure cascade with a transition duct between compressor compartments are determined. The excess of the middle diameter of the intermediate-pressure compressor over the middle diameter of the high-pressure compressor and introduction of the transition duct between them into the scheme of the compressor flow path of the core engine is justified. Applying of a diagonal turbine in an intermediate-pressure cascade is proposed. The axial length of the flow path channels between compressors and turbines was chosen considering the influence of the duct opening angle on hydraulic losses and mass-dimensional characteristics of the core engine.

1 INTRODUCTION

Approaches to the flow path (FP) shape selection for the turbomachinery (TM) of aviation gas turbine engines (GTE), as well as their main structural unit - the core engine (CE) have already been proposed in a number of works (Kholshevnikov, K. (1965), Bakulev, V. (2003), Bochkaryov, S., Kuzmichev, V. (2005)). However, as the GTEs develop, their schemes become more complex and new generations of engines appear, so there is a need to adjust the algorithms of core engine flow path formation and constraints of mode, gas-dynamic and structural-geometric character. This is related both to new approaches and information capabilities of GTE design, and to new materials, production technologies and design innovations.

This paper examines the issue of designing a twin-shaft gas generator of a gas turbine engine. This type of gas generator is installed on three-shaft engines such as the RR Trent. They have a cascade of low (LP), medium (IP) and high (HP) pressure. The gas generator includes IP and HP cascades. The LP cascade changes depending on the engine modification. The use of such gas operators makes it possible to increase the efficiency of the engine and the stability of its operation, and reduce weight and size. However, three-shaft engines are significantly more complex.

2 ALGORITHM FOR DESIGNING A TWIN-SHAFT GAS GENERATOR

The flow path formation of CE turbomachines in the meridional plane is carried out after determination of

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its main parameters (pressure ratio in the CE $\pi^*_{c\,CE}$, gas temperature T^*_g at the turbine inlet, pressures p^*_i and temperatures T^*_i, as well as flow rates G_i of the working fluid in characteristic cross-sections of the FP) at the step of thermodynamic calculation of the whole engine.

The purpose of formation of the FP appearance of CE turbomachines is, at least, to ensure the values of parameters accepted and obtained in the thermodynamic calculation of GTE, which allow:
- to reach the thrust values at cruise and take-off modes established by the technical specification (TS);
- to ensure that the specific fuel rate at cruise mode does not exceed the value specified in the TS;
- to ensure the compressor cascades operation conditions in the modes without stall;
- not to exceed the specified limit of the total mass of the turbomachinery;
- to fulfil the CE turbomachines with a minimum number of stages.

It is reasonable to divide the formation of the TM flow path of a two-shaft core engine in the meridional plane into the following steps.

Step 1. Determination of rational distribution of specific work and pressure ratio in the CE between the intermediate pressure compressor (IPC) and the high pressure compressor (HPC).

Step 2. Selection of the high-pressure (HP) cascade speed and determination of the main structural and geometrical parameters of the high-pressure turbine (HPT) and HPC in the meridional plane.

Step 3. Selection of rotation frequency of the intermediate pressure (IP) cascade and determination of the main structural and geometrical parameters of the intermediate pressure turbine (IPT) with a transition duct from the HPT to the IPT and the IPC with a transition duct from the IPC to the HPC in the meridional plane.

3 RATIONAL DISTRIBUTION OF ENERGY BETWEEN IPC AND HPC

The distinctive feature of core engine of perspective GTE schemes of the fifth and sixth generations is further increase of gas temperature at the turbine inlet T^*_G up to 1900 and 2100 K, total pressure ratio in CE compressor $\pi^*_{c\,CE}$ up to 25-30 and 30-40, reduction of stage number of compressor cascades and application of only single-stage cooled HPT and IPT.

The FP scheme of a two-shaft CE in the meridional plane with the designation of characteristic sections is shown in Figure 1.

The distribution of pressure ratio $\pi^*_{C\,CE}$ and specific work $L_{C\,CE}$ of the entire core engine by compressor cascades is proposed to be carried out focusing on the rational distribution of specific work and the degree of pressure decrease in the CE between the HPT and IPT, Grigor'ev, V. (2009). It should be taken into account that these turbines at cruise mode operate at the loading parameter $Y^*_{st} \approx Y^*_{T\,opt} \approx 0.55$ and reactivity degree ρ_{st}= 0.40-0.45.

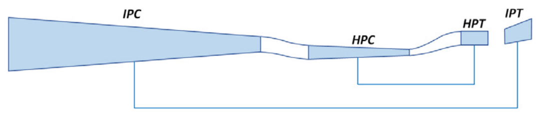

Figure 1: Flow path scheme of a two-shaft core engine.

Under these conditions, it is necessary to achieve such a loading of the HPT and IPT that no zones with increased supersonic flow velocities and, consequently, wave losses would appear in their flow path at the middle diameter. In this case, it is desirable that the values of the reduced isentropic flow velocities both at the outlet of the stator blade (SB) in absolute motion λ_{1s}, and at the outlet of the rotor wheel (RW) in relative motion λ_{W2s} would be approximately the same in the HPT and IPT.

In contrast to the previously proposed variants (Grigor'ev, V. (2009)), it is recommended to determine the rational distribution of specific work between intermediate and high-pressure cascades in accordance with the algorithm shown in Figure 2.

As initial data for the first approximation, the values of parameters from thermodynamic calculation of the whole engine at cruise mode and the same values of pressure ratio in IPC π^*_{IPC} and HPC π^*_{HPC} are taken: $\pi^*_{IPC} = \pi^*_{HPC} = \sqrt{\pi^*_{c\,CE}}$.

According to the standard methods, using the initial data for both the HPT and IPT, the reduced isentropic flow velocities at the SB outlet in absolute motion λ_{1s} and at the RW outlet in relative motion λ_{w2s} of the HPT and IPT at the middle diameter are determined (Figure 2).

After that, the values of the corresponding velocities in the HPT and IPT are compared with each other. If at practically identical values of ρ_{st} of HPT and IPT the value of relative difference $(\lambda_{1s\,HPT} - \lambda_{1s\,IPT})/\lambda_{1s\,HPT}$ or $(\lambda_{w2s\,HPT} - \lambda_{w2s\,IPT})/\lambda_{w2s\,HPT}$ will be more than 3%, then the value of π^*_{HPC} at the next iteration of calculation decreases. And if these values are less than minus 3%, then π^*_{HPC} increases.

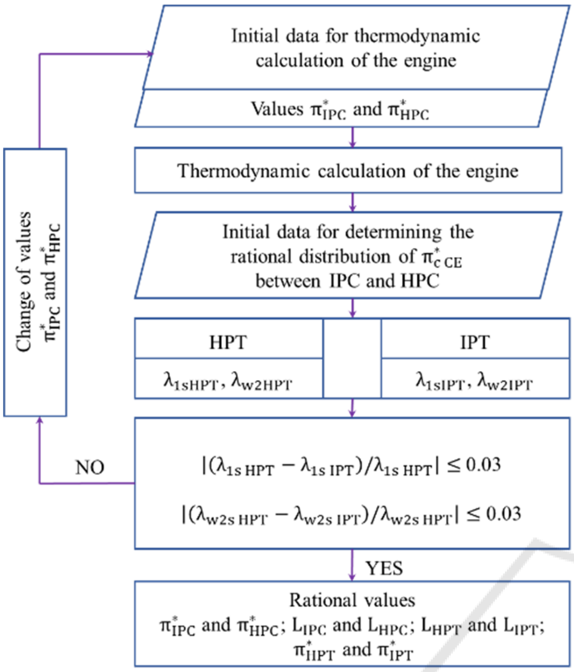

Figure 2: Algorithm for determining the rational distribution of specific work between intermediate and high-pressure cascades.

At the expense of some adjustment of the values ρ_{st} of HPT and IPT in the above ranges, it is reasonable to ensure approximate equality of the reduced velocities $\lambda_{1s\,HPT}$ and $\lambda_{W2s\,HPT}$, as well as $\lambda\lambda_{1s\,IPT}$ and $\lambda_{W2s\,IPT}$ in order to reduce the wave losses in the turbine flow path.

When the conditions $|(\lambda_{1s\,HPT} - \lambda_{1s\,IPT})/\lambda_{1s\,HPT}| \le 0.03$ and $|(\lambda_{w2s\,HPT} - \lambda_{w2s\,IPT})/\lambda_{w2s\,HPT}| \le 0.03$ are reached, the final values of pressure ratios π^*_{IPC} and π^*_{HPC}, specific works L_{IPC}, L_{HPC}, L_{HPT} and L_{IPT}, as well as the pressure decrease ratios in the HPT π^*_{HPT} and IPT π^*_{IPT} are recorded.

The results of determining the reduced isentropic flow velocities at the outlet from the SB and RW of high and intermediate pressure turbines at successive iterations of the calculation of a promising aviation GTE at different values of $\overline{\pi^*}_{IPC} = \pi^*_{IPC}/\pi^*_{K\,CE}$ are presented in Figure 3.

As can be seen from the presented dependences, the $\lambda_{1s\,HPT} \approx \lambda_{1s\,IPT}$ and $\lambda_{W2s\,HPT} \approx \lambda_{W2s\,IPT}$ equalities occur approximately at $\overline{\pi^*}_{IPC} = 0.225$ and $\overline{\pi^*}_{HPC} = 0.145$. At these values of relative pressure ratios in IPC and HPC, the values of relative specific work are $\overline{L}_{IPC} = L_{IPC}/L_{C\,CE} = 0.43$ and $\overline{L}_{HPC} = L_{HPC}/L_{C\,CE}$=0.57. Relative pressure decrease ratios in HPT are $\overline{\pi^*}_{HPT} = \pi^*_{HPT}/\pi^*_{T\,CE} = 0.399$ and in IPT $\overline{\pi^*}_{IPT} = \pi^*_{IPT}/\pi^*_{T\,CE}$ =0.363 ($\pi^*_{T\,CE}$ is the total pressure decrease ratio in CE turbines).

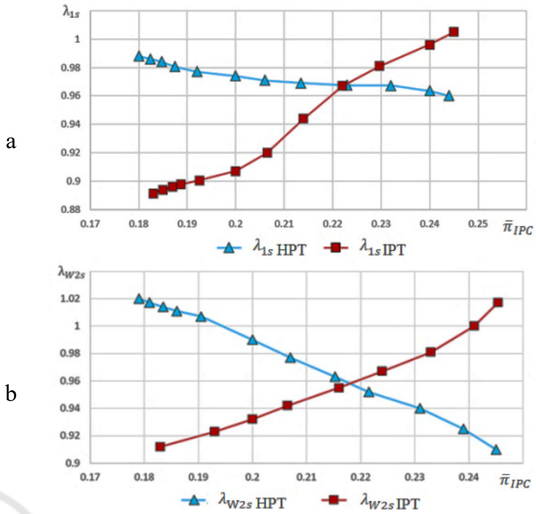

Figure 3: Dependences of the reduced isentropic flow velocities on $\overline{\pi^*}_{IPC}$: a - at the SB outlet; b - at the RW outlet.

4 SELECTION OF THE HP CASCADE SPEED AND FLOWPATH GEOMETRY

Assignment of the rotational speed *n* of the cascade rotor at the design cruise mode of engine operation and determination of the main design-geometric parameters of the TM of the cascade has a determining influence on the efficiency of its operation and the FP dimensions. Basically, as cited in Krupenich, I. (2006), the following system of equations is used to determine the circumferential velocities at the TM middle diameter, the rotational speed of the cascade, as well as its main design and geometrical parameters (stage number of the cascade compressor and TM middle diameters) at the design mode:

$$U_{c\,mid} = \pi D_{c\,mid} n_c/60;$$
$$U_{t\,mid} = \pi D_{t\,mid} n_t/60;$$
$$L_{st} = U^2_{t\,mid}\eta^*_{st}/[2(Y^*_{st})^2];$$
$$L_c = z_c \overline{H}_{c\,mid} U^2_{c\,mid}$$

$U_{c\,mid}$ - circumferential velocity at the middle diameter of the compressor cascade;
$D_{c\,mid}$ - middle diameter of the compressor cascade;

n_c- rotor speed of the compressor cascade;
$U_{t\,mid}$ - circumferential velocity at the middle diameter of the turbine cascade;
$D_{t\,mid}$ - middle diameter of the turbine cascade;
n_t- rotor speed of the turbine cascade;
L_{st} - specific work of the turbine stage;
η^*_{st} - efficiency of the turbine stage;
Y^*_{st} - turbine stage loading parameter;
L_c - specific work of the compressor cascade;
z_c - stage number of compressor cascade;
$\overline{H}_{c\,mid}$ - average coefficient of expended head at the middle diameter.

This system has four equations and eight unknowns ($U_{c\,mid}$, $D_{c\,mid}$, $U_{t\,mid}$, $D_{t\,mid}$, Y^*_{st} , $\overline{H}_{c\,mid}$, z_c and $n_c = n_t = n$). The values of the parameters L_c , L_t and η^*_{st} are known from the thermodynamic calculation of the whole engine and having performed the previous step of the calculation.

Thus, in the above system of equations, the four parameters of the equations must be either given or otherwise determined.

According to Bakulev, V. (2003) the value of the loading parameter Y^*_{st} is usually set close to the optimum in the range of 0.50-0.60, the value of $\overline{H}_{c\,mid}$ is in the range of 0.30-0.40.

The smallest value of the middle diameter of the turbine stage is limited for RW strength considerations by the following ratio

$$(D_{t\,mid}/h_{2t}) = U^2_{t\,mid\,take-off}/\varepsilon_{t\,take-off},$$

$U_{t\,mid\,take-off}$ - circumferential velocity of turbine RW at the middle diameter at take-off mode;

h_{2t} - blades height at the outlet of the turbine RW;

$\varepsilon_{t\,take-off}$ - the largest permissible stress parameter at take-off mode, the value of which is roughly in the range of (18…20)·10^3 m^2/s^2 for HPT, and for IPT is in the range of (23…25)·10^3 m^2/s^2, Grigor'ev, V. (2009).

The middle diameter of the compressor cascade is determined either from the condition of achieving the highest permissible reduced circumferential velocity at the periphery of the first stage RW $U_{in\,shrud\,red}$ (430-450 m/s for IPC and 330-350 m/s for HPC), or from the condition of limiting the minimum permissible blades height at the compressor cascade outlet $h_{c\,min}$ = 15-20 mm, Bakulev, V. (2003).

Taking into account the mentioned restrictions and FP areas in characteristic sections of turbomachinery calculated using the equation of continuity, the algorithm of finding the main geometrical parameters of turbomachinery FP in the meridional plane and determination of their rotation frequencies is compiled.

The choice of rotor speed n_{HP} and determination of the main structural and geometrical parameters of the HP cascade in the meridional plane is carried out using the algorithm, which has two parallel branches (Figure 4):

- a branch defining the highest permissible rotational speed of the HPT $n_{HPT\,add}$ in terms of strength (left column of the algorithm);
- a branch defining the highest permissible rotational speed $n_{HPC\,add}$ in terms of the efficiency of the working process of the last HPC stage (right column of the algorithm).

As initial data, we take the parameters whose values were found in the thermodynamic calculation of the engine at cruise and take-off modes, as well as in determining the rational distribution of specific work between the cascades of core engine at the first step of the calculation. The initial data includes also the equivalent engine operating time at take-off mode $\tau_{take-off}$, as well as the density of the rotor blade material ρ_{bl} and the Larson-Miller diagram corresponding to this material.

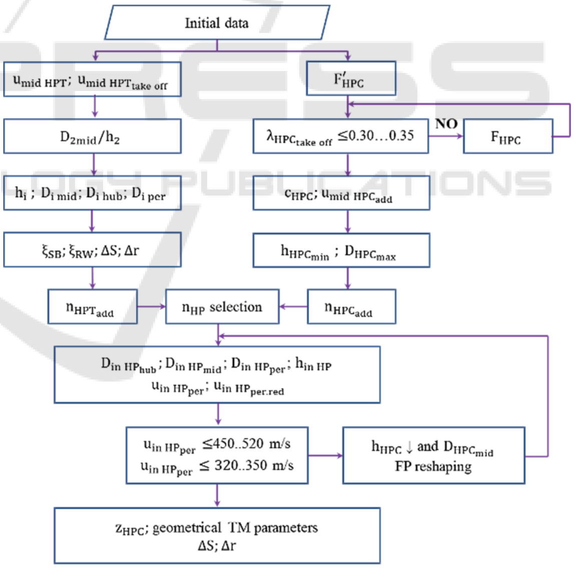

Figure 4: Algorithm for determining n_{HP} and structural and geometrical parameters of HPC and HPT.

Based on the initial data, when determining the value of $n_{HPT\,add}$, the circumferential velocities at the RW middle diameter at cruise and take-off modes and the ratio of the middle diameter to the height of the blades at the turbine outlet $D_{2\,mid}/h_2$ are initially determined. Then the blade heights h_i and diameters $D_{i\,mid}$, $D_{i\,per}$, $D_{i\,hub}$ of the flow path in characteristic sections are determined (index i denotes the number of the characteristic section), as well as, in accordance with the known recommendations cited in Belousov, A. (2006), the values of the width of the blade row of the stator blade S_{SB} and rotor wheel S_{RW}, axial ΔS and radial Δr clearance.

As a result, the permissible RW rotational speed of the HPT at cruise mode is calculated as $n_{HPT\,add} = 60U_{mid\,HPT}/(\pi D_{2\,mid})$ in terms of the strength of the rotor blades.

When determining the permissible highest rotational speed $n_{HPT\,add}$ in terms of the working process efficiency of the last HPC stage, we find in the first approximation the cross-sectional area at the HPC outlet $F'_{c\,HP}$ and the corresponding reduced flow velocity at take-off mode $\lambda_{c\,HP\,take-off}$, which value should not exceed 0.30-0.35, Bakulev, V. (2003). If this condition is not fulfilled, the value of the area at the outlet of the HPC is corrected (Figure 4).

After that, the flow velocity at the outlet of the HPC $C_{c\,HP}$ is found at cruise mode. Then, the largest permissible circumferential velocity at the middle diameter of the last RW stage is determined as $U_{mid\,c\,HP\,add} = C_{c\,HP}/\overline{C}_{a\,min}$. Where $\overline{C}_{a\,min}$= 0.39-0.41 is the range of selection of the smallest permissible value of the flow rate coefficient at the outlet of the last compressor stage, recommended in terms of an acceptable degree of flow diffusivity in the blade passage of the RW and guided vane of the last stage.

The circumferential velocity $U_{mid\,c\,HP\,add}$ is related to the middle diameter at the outlet of the last stage of the HPC $D_{c\,HP\,mid}$ and the rotor speed of the HPC $n_{HPC\,add}$ by the expression $U_{mid\,c\,HP\,add} = \pi D_{c\,HP\,mid} n_{HPC\,add}/60$. By specifying the diameter $D_{c\,HP\,mid}$, from the last expression, the rotational speed $n_{HPC\,add}$, that provides the highest circumferential velocity $U_{mid\,c\,HP\,add}$, can be obtained.

In this case two variants are possible:

- If $n_{HPC\,add}$ is larger than $n_{HPT\,add}$, then $n_{HPT\,add}$ is taken as the shaft rotational speed of the HP cascade n_{HP};

- if $n_{HPC\,add}$ is less than $n_{HPT\,add}$, then $n_{HPC\,add}$ is taken as the shaft rotational speed of the HP cascade n_{HP}.

As the GTE design experience of the latest engine generations shows, Bakulev, V. (2003), the second variant of events turns out to be practically impossible. The point is that even at the largest middle diameter at the HPC outlet $D_{c\,HP\,mid\,largest}$, which is calculated by the formula: $D_{c\,HP\,mid\,max} = F_{c\,HP}/\pi h_{c\,HP\,min}$, where the smallest blade height of the last HPC guided vane $h_{c\,HP\,min}$ is taken not less than 15-20 mm, Bakulev, V. (2003), the value $n_{HPT\,add} = 60U_{mid\,c\,HP\,add}/\pi D_{c\,HP\,mid\,max}$ appears to be higher than the rotational speed $n_{HPT\,add}$.

The performed example calculation confirmed this trend. The value of $n_{HPT\,add}$ obtained at $h_{c\,HP\,min}$= 21 mm and $D_{c\,HP\,mid\,max}$ was found to be greater than $n_{HPT\,add}$.

After that, the shape of the HPC flow path in the meridional plane is selected and such parameters at its inlet as the diameters $D_{in\,HP\,hub}$, $D_{in\,HP\,mid}$, $D_{in\,HP\,per}$ and blade height $h_{in\,HP}$, the circumferential velocity at the RW periphery of first stage $U_{in\,HP\,per}$ and the corresponding reduced circumferential velocity $U_{in\,HP\,per\,red}$ are determined.

At the same time, there is a check of compliance with the restrictions on the values of these last velocities. Namely, the velocity $U_{in\,HP\,oer}$, based on strength conditions, should be less than 450-520 m/s, and the velocity $U_{in\,HP\,per\,red}$ for gas dynamic reasons should not exceed 320-350 m/s.

Otherwise, it is necessary to increase $h_{c\,HP}$ and decrease $D_{c\,HP\,mid}$ or change the FP shape of the HPC in the meridional plane and correct the calculation, starting with determination of geometrical parameters at the HPC inlet (Figure 4).

After that, according to the standard methodology, the HPC stage number z_{HPC} is determined and geometrical parameters of each blade row (diameters, height and width) characterising it in the meridional plane are calculated, and axial ΔS and radial Δr clearances are selected.

5 DESIGN OF A INTRMIDEATE PRESSURE CASCADE

It is reasonable to estimate the main parameters of the flow path of the intermediate-pressure cascade in the same sequence as the parameters of the high-pressure cascade, but taking into account a number of peculiarities.

Firstly, since the middle diameter at the outlet of the IPT rotor wheel $D_{2\,IPT\,mid}$ is significantly larger than the same diameter at the HPT outlet $D_{2\,HPT\,mid}$, the intermediate pressure turbine should be made diagonal. This makes it possible to exclude the transition duct between the HPT and IPT.

Secondly, for design reasons related to the necessity to place the rotor of the low-pressure cascade inside the shaft of the intermediate-pressure cascade, it is required to provide the hub diameter of the IPC larger than the minimum permissible value.

Thirdly, it should be taken into account that the circumferential velocity at the RW periphery of the first stage of IPC $U_{in\,IP\,per}$ in terms of strength is limited by the value of 450-500 m/s, Belousov, A. (2006), and the reduced circumferential velocity $U_{in\,IP\,per\,red}$ in the same section is limited by the values of 430-450 m/s for gas-dynamic considerations.

Fourthly, as well as in the case of high-pressure cascade, the permissible IPC rotational speed $n_{IPC\,add}$ is usually higher than the permissible IPT rotational speed $n_{IPT\,add}$. Therefore, $n_{IPT\,add}$ is taken as the rotational speed of the IP shaft n_{IP}, and in order to maintain the accepted value of the circumferential velocity $U_{in\,IP\,per\,red}$ and, possibly, not to increase the IPC stage number, it is reasonable to increase the middle diameter of the IPC by a factor of $n_{IPC\,required}/n_{IPT\,add}$.

The increase of the middle diameter $D_{IPC\,mid}$ leads to the necessity of the transition duct between IPC and HPC.

For the considered example, the flow path scheme of the turbomachinery of a two-shaft core engine with observance of proportions in axial and radial directions is shown in Figure 1.

6 CONCLUSIONS

The developed algorithm of the flow path design formation of the turbomachinery of a two-shaft core engine makes it possible to find rational proportions of pressure ratios of IPC and HPC, as well as pressure ratios of HPT and IPT, using the reduced flow velocities at the outlet of the turbine blade row as rationality criteria.

It makes it possible to select the rotational speed of intermediate and high-pressure cascades taking into account strength and gas dynamic limitations, as well as to determine the main structural and geometric parameters of the core engine turbomachinery in the meridional plane.

To reduce the axial dimensions of the turbine part of the core engine a diagonal-type IPT can be used.

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