AERODYNAMICS AND STRENGTH OF A TWO-TIER STAGE BASED ON A FORK-SHAPE BLADE

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ABSTRACT

The specific metal consumption is a metal amount spent per an installed power unit. This criterion is important for turbine quality assessment. This criteria drops rapidly when a turbine power increase remarkably leaves behind its total metal consumption growth. This event comes to reality in design of a low pressure turbine with maximal steam flow capacity. This report shows that this solution inevitably hurts the LP turbine efficiency. The two-tier blade is an alternative solution of this problem. The lower tier airfoil carries a plate with two upper tier airfoils that have relatively small chords. This presents a complete solution of the fan factor aerodynamic problem. On the other side this 1200 mm span blade may cause operational reliability problems. This report discloses strength analysis for this blade and the structure improvements that combine fulfilment of the reliability requirements and the high stage efficiency.

KEYWORDS

LOW-PRESSURE TURBINE, STEAM TURBINE, TWO-TIER TURBINE

NOMENCLATURE

\( l \) blade length, [mm]
\( D_{\text{mean}} \) mean diameter, [mm]
\( D_r \) root diameter, [mm]
\( u \) mean velocity, [m/s]
\( u/c_f \) kinematic parameter
\( \varphi \) velocity coefficients at vane assembly, [-]
\( \varphi \) velocity coefficients at blade assembly, [-]
\( \eta \) blade efficiency, [%]
\( H_{0z} \) accessible enthalpy drop in the last stage, [J/kgK]
\( D_{\text{mean}}/l \) relative mean diameter, [-]
\( \Delta l \) absolute overlap of the vane and blade airfoil grids, [mm]
\( \Delta l/l \) relative overlap, [-]

INTRODUCTION

Current tendency of steam turbine manufacturing shows an interest to development of single-shaft condensing turbines with one Low Pressure Turbine (LPT) of maximal power.

At given steam parameters the maximal power is determined by the maximal LPT steam flow to condenser.

The traditional solution of this problem is development of longer blades for the condensing turbine last stage based on available possibility for reliable stage operation.

In Russia there is a successful experience in a few decades running high speed condensing turbines with 1200 mm long last stage blades.
Mitsubishi Hitachi Power Systems company has developed a 1500 mm long blade for a high speed turbine that allows a remarkable increase of the condensing turbine maximal power which drastically improves such an important economy performance as specific metal consumption (Tanuma, 2017; Fukuda, et. al, 2009; Shigeki, et. al, 2014).

Nevertheless the available published research materials don’t disclose the blade length influence upon the LPT efficiency because the LPT flowpath configuration generally differs from the HPC and IPC ones.

In a LPT the steam travel from the first to the last stages is followed by 25-30 times increase of its specific volume. Thus the flowpath has a very large meridional plane expansion degree. The transition overlay between the last and previous stages is up to 40-50% and the outer housing inclination is 40-45°.

At the given inclination angles of the Low Pressure Turbine (LPT) outside contour not only the stages overlap are aerodynamically not acceptable but also is the overlap between vanes and blades within the stage.

In this stage it causes a rapid increase of the energy peripheral losses and the overlap between stages actually does not allow acceptable local efficiency values in the LPT last stages peripheral parts that are filled with the steam flow due to the inevitable radial flow in inter-vane channels of the last stages.

The both mentioned effects drastically reduce the last stage efficiency when the last stages blade length is increased.

The paper (Scheglyaev, 1947) data on detailed efficiency investigations of the K-100-90 turbine LPT show that the test data values are 15-20% below the analysis ones. This shows serious problems concerned with the LPT efficiency when the steam flow capacity to these turbines condensers is increased by direct increase of the last stages blade length. Unfortunately we could not find later data on direct efficiency evaluation of last stages in powerful condensing turbines.

The test data (Scheglyaev, 1947) show that the power increase of single flow condensing turbines by increase of the last stages blade length causes a remarkable efficiency drop.

An alternative solution of this problem is a transition to the two-tier LPT. It allows application of successful 1200 mm long blade combined with the maximal power increase similar to application of longer 1500 mm blades (Zaryankin, et. al, 2015; Zaryankin, et. al, 2017; Zaryankin, et. al, 2018).

In this version the LPT contains two independent turbines, the existing flowpath is located in the lower tier of two-tier LPT and the aerodynamically independent turbine with its smaller number of stages occupies the upper tier. This turbine is designed for the enthalpy drop equal to the lower tier turbine one. The upper tier flowpath design is based on the Bauman two-tier stage. Length of its last stage blade is equal to the lower tier blade length. This blade design is based on the original “fork-shape” blade (Russia patent #139602) that allows reduction of the negative effects that are specific for exit stage long blades of powerful condensing turbines. This results in 1.5 time increase of the steam flow into condenser with retained length of the last stages blade. The two-tier LPT efficiency parameters are not lower than those in the LPT with longer blades.

The problems of two-tier stages efficiency and reasonable design are sufficiently described in published papers (Zaryankin, et. al, 2015; Zaryankin, et. al, 2016; Zaryankin, et. al, 2017) but the practical introduction and the strength and vibration reliability are presented for wide discussions for the first time.

**TWIN-TIER TURBINE STAGE WITH “FORK-SHAPE” BLADE AERODYNAMIC PERFORMANCE.**

The 1500mm long blade by “Mitsubishi Hitachi Power Systems” is surely a remarkable success of modern turbine manufacturing industry. This allows development of super powerful single shaft steam turbines above 2000 MW with extremely low specific metal consumption (Tanuma, 2017; Fukuda, et. al, 2009, Shigeki, et. al, 2014).
Application of this blade to condensing turbines of over 500 MW power remarkably reduces the specific metal consumption by reduction of the metal consuming LPTs number.

This highlights the problem of the LPT efficiency improvement. For a more distinct understanding of this problem consider the flowpath of a single-flow four stage LPT with the last blade length 1400mm (figure 1). At the specific mean diameter $D_{mean}/l = 2.5$ the root diameter $D_r$ of this stage is $D_r=2100$ mm.

Respectively, at the constant enthalpy drop along the blade span the kinematic parameter $u/c_f$ design value at the blade root and tip will be 40% lower and 40% higher than the mean diameter $D_m=3500$ mm value. Here $u$ – circular velocity, $c_f = \sqrt{2H_0z}$ velocity equivalent to the enthalpy drop $H_0z$ in the LPT last stage.

If the assumed in the efficiency analysis the velocity coefficients $\phi$ and $\psi$ stay constant along the height at the given $u/c_f$ deviation from its optimal value $(u/c_f)_{opt}$ at the mean diameter the root and tip efficiency reductions will be 10% and 4% respectively. Here the stage efficiency drop includes the reaction degree reduction down to 10% at the root and the reaction increase at the tip up to 80%.

Long blades tip losses are an order higher than the shorter blades with $D_{mean}/l > 10$ ones with lower coefficients $\phi$ and $\psi$. In tip sections the velocity coefficient $\phi$ drops to 0.78 from 0.965 in the mean section and the coefficient $\psi$ drops from 0.958 to 0.82 (Troyanovsky, et. al, 1985).

In root sections these coefficients reduction at the design point operation is smaller and our calculations give $\phi=0.89$ and $\psi=0.87$.

![Figure 1: LPT with 1400 mm long last stage blade](image)

This event causes a large reduction of the local stage efficiency values in root and tip parts. This is confirmed by full-scale tests of the efficiency distribution along the blade span in the fourth stage of the five stage BK-100-5 LMZ turbine LPT (ref. picture 2 in the paper (Lagun and Simoyu, 1967)). The compartment was tested at the overheated steam operating mode with $Re = (1.2\div1.4)10^5$.

In this rig the fourth stage mean diameter efficiency is 79%, in tip direction it drops rapidly to 60% and in blade root direction the rig efficiency drops down to 70%.
This old experimental work says that “at \( u/c_f = 0.6-0.8 \) the last stage internal specific efficiency (besides the humidity losses) has 40-45\% values and 10-25\% differs from the design values”. Based on this investigation the report (Lagun and Simoyu, 1967) includes the following important conclusion: “The high efficiency of an LPT stage cannot be reached only by the spin design based on the radial equilibrium equation. In the blade analysis and aerodynamic development it is necessary to take into consideration the meridional side flow, but to avoid the rapid flowpath expansion and large overlays”.

A review of the current steam turbine LPT flowpath configurations shows that the recommendations above are not followed, and the negative influence of mentioned factors grows together with the last stages blade length increase. This is confirmed by the LPT flowpath configuration with the last stage blade length \( l = 1400 \text{ mm} \) shown in figure 1. Here the flowpath expansion is 45\° and the overlay between the last stage is up to 50\%. The test data (Deitch, 1996) help to analyze the described factors influence upon the energy losses increase in the considered stages.

Figure 3 shows the energy loss coefficients along an annular vane grid with a cylindrical housing and steam supply overlay 34.4\% and 51.5\% (curves 1 and 2). As expected that maximal losses occupy the grid peripheral part.

![Figure 2: Efficiency distribution along the third stage blade span in the BK-100-5 ЛМЗ turbine LPT three stage rig (Lagun and Simoyu, 1967).](image)

It is worth mentioning that the large overlay principally changes the vane grid exit velocity distribution, it increases the loss coefficient from 5\% at zero overlay \( \Delta l/l_0 = 0 \) to 16\% at \( \Delta l = 0.344 \) and to 27\% at \( \Delta l = 0.515 \). These loss coefficients correspond to very low velocity coefficients \( \phi \) equal to 0.905 and 0.851 respectively. Note that in the last stage mean diameter vane analysis the mentioned velocity coefficient is assumed as 0.965 (Troyanovsky, et. al, 1985).

The actual stage problem is more complicated because the vane flowpath outer counter has a cone angle \( \gamma = 40-45^\circ \).

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Experimental evaluation of the fan factor $D_{\text{mean}}/l$ influence upon the energy losses shown in figure 4 (Deitch, 1996) is also interesting. Here the fan factor $D_{\text{mean}}/l$ reduction from 10 to 2.5 at linear $\alpha_1$ angle increase from root to tip is followed by the fan losses increase of 5.3%.

The test data above obtained in 1960-s are highly appreciated because taking them into account results in the condensing turbine last stages actual efficiency remarkably below the design values.

In this case the financially reasonable alternative solution is the transition to two-tier LPT. The two-tier LPT flowpath shown in figure 5 consists of lower tier five stages and upper tier three stages.

Here the second and third stages are usual Bauman stages and the fourth stage is based on a new fork-shape blade. The two-tier fork shape blade (figure 6) is the key element of the shown LPT flowpath.

The described blade based on the Russia patent #139602 has the $l = 1200$ mm total length and differs from the Bauman blade with its upper tier two blades located on the lower tier carrying blade and the upper tier airfoils have chords remarkably smaller than the lower tier ones.

The twice larger upper tier blades number allows optimal pitch values and smaller tip loss coefficients at smaller chords. Besides this the last stage upper tier specific mean diameter $D_{\text{mean}}/l_z$ is increased to 9.2. At this parameter value the fan factor losses are near to zero. In other words in this case the upper tier blade efficiency including the lower exit velocity losses is higher than the common configuration lower tier efficiency. In the presented flowpath the lower tier blade efficiency $\eta=0.881$ is almost equal to the upper tier efficiency $\eta=0.884$.

![Figure 3: Distribution of energy loss coefficient along the height of annular grid with cylindrical housing at different overlays (Deitch, M.E., (1996))](image1)

![Figure 4: Influence of vane grid fan factor upon the additional losses, 1 – linear $\alpha_1$ increase from root to tip, 2 – grid with constant airfoil along height (Deitch, M.E., (1996))](image2)
As mentioned in the introduction the two LPT tiers are aerodynamically equal. The practical introduction of this solution requires detailed strength studies of the new blade shown in figure 5.

**Figure 5:** Two-tier LPT

**Figure 6:** Two-tier fork shape blade
LONG L = 1200MM TWO-TIER FORK SHAPE BLADE STRENGTH ANALYSIS RESULTS

The FEM stress analysis of the fork-shape blade is done with the Ansys Mechanical code at design point conditions. The flow simulation meshes are hybrid unstructured. The main flow zone is formed of tetrahedrons, the wall zone of prisms. The main flow element maximal linear dimension is 0.2 mm, the total elements number 6.7 to 7.0 million.

The used elements type is SOLID185 (TET4), the tetrahedron first order elements with four nods in endpoints. The mean specific Jacobian value is 0.577, high quality elements have this value from 0.3 to 1.0. This quality criteria shows the element difference from the ideal one, the equilateral tetrahedron in this case.

The stress analysis is done for the root stiff fixture and the inter-tier shrouds interaction conditions. The operation factors like the upstream vane trailing edge traces are not considered. The analysis conditions are the 3000 rpm rotation speed and the all directions zero displacements in the root coupling mounting planes. The analysis involves the aerodynamic loads according to the blade operation conditions. The pressure side steam pressure in upper and lower tiers is assumed from the power Nu re-calculation in each tier mean diameter and each airfoil pressure side area. Each tier power production is taken from the two-tier LPT flowpath thermal calculation (Zaryankin, et. al, 2015; Zaryankin, et. al, 2017; Zaryankin, et. al, 2018). The upper tier and disk temperature is assumed constant and equal to the lower tier inlet temperature. The inter-tier split shoulders of the neighboring blades interaction was simulated in local coordinate systems on the both shoulders faces. The coordinates were directed along the face surface axis and in radial direction.

Also are assumed the equations for the shoulder face rotation angle around the local coordinate radial axes in the proportional ratios 1:1. The sliding freedom degrees stayed free.

The blade material is the domestic titanium alloy BT-6 similar to the foreign alloy Ti-6Al-4V used for the 1500 mm long blade by Mitsubishi. Advanced heat treatment and hardening methods for titanium slags provide the yield stress of 1100 MPa (Il’in et. al, 2009)

The fork shape blade stress for upper and lower tiers is presented in figures 7 and 8 respectively. The titanium alloy BT-6 yield stress $\sigma_{02} = 1103$ MPa makes a base for the strength margin calculation which is evaluated for the each tier blade specific zones and the inter-tier shoulder. The upper tier blade analysis shows four main zones, the trailing edge root fillet with the $\sigma_{eq} = 500$-518 MPa stress, root zone with $\sigma_{eq} = 324$ MPa stress, middle zone with $\sigma_{eq} = 270$-280 MPa stress and the tip zone with $\sigma_{eq} = 124$ MPa. The root fillet with the maximal stress level is the upper tier blade critical zone where the stress margin is $n = 1103/517=2,13$. According to the regulations (RD. 24.033.02-88) the twisted blade root section stress margin must be above $[n]=1,25$.

The lower tier airfoil (figure 8) has larger stress variation of 145-679 MPa than the upper tier one. Like in the upper tier the lower tier maximal stress zones are located near the root $\sigma_{eq} = 665$ MPa and in the root fillet $\sigma_{eq} = 679$ MPa, the stress margins are $n = 1103/665=1,65$ in the root zone and $n = 1103/679=1,62$ in the root fillet.

The mostly loaded element of the fork shape blade is the inter-tier split shoulder. The critical zones are the area between blades with $\sigma_{eq} = 886$ MPa and the shoulder overhang zone in front of the upper tier blade with $\sigma_{eq} = 719$ MPa. The stress margin values are $n = 1103/886=1,25$ in the area between the upper tier blades and $n = 1103/719=1,53$ in the shoulder overhang zone.

Stress analysis of the developed two-tier fork shape blade shows that its critical zones with maximal equivalent stresses correspond to the strength criteria stipulated in the industry requirements for twisted blades (RD. 24.033.02-88).

The blade reliability is also determined by its vibration state so the further analysis was the blade natural frequencies and oscillation modes, the calculation results are summarized in table 1.
Dynamic frequencies of single blades, blade clusters and the blades with annular and staggered links at the 501/sec must comply with the steam turbine blade vibration tuning regulations (Zaryankin, 2015) and be within one of the following frequency intervals:

- $f_{d1} < 91$;
- $107 < f_{d2} < 140$;
- $158 < f_{d3} < 188$;
- $209 < f_{d4} < 237$;
- $260 < f_{d5} < 285$;
- $310 < f_{d6} < 47z$,

where $z$ – number of upstream vanes. Thus, according to the regulations (RTM 108.021.03-77) the blade requires additional tuning of the first axial coupled, or second in order mode.

**Figure 7: Upper tier blade equivalent stress, MPa**

**Figure 8: Lower tier fork shape blade equivalent stress, MPa**
Table 1: Blade natural frequencies and mode shapes

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency, Hz</th>
<th>Vibration mode</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>90.523</td>
<td>First tangential coupled</td>
</tr>
<tr>
<td>2</td>
<td>156.48</td>
<td>First axial coupled</td>
</tr>
<tr>
<td>3</td>
<td>166.75</td>
<td>First tangential upper tier</td>
</tr>
<tr>
<td>4</td>
<td>217.71</td>
<td>Second tangential coupled</td>
</tr>
<tr>
<td>5</td>
<td>264.82</td>
<td>First torsional coupled</td>
</tr>
<tr>
<td>6</td>
<td>325.59</td>
<td>Second tangential upper tier</td>
</tr>
</tbody>
</table>

The question of the blade manufacturing technology arises due to its complicated configuration. So after completion of the blade detailed design its possible manufacturing technology was developed. The natural scale blade demonstrator mockup (figure 9) confirms the manufacturing possibility.

Figure 9: Fork shape blade demonstrator mockup

CONCLUSIONS

Available test data on operating fluid flow in large condensing turbine last stages at very large stages overlay and very large flowpath expansion show that in this case intensive radial flow in the last stages vanes drastically hurts the stage efficiency in blade root and tip regions.

Increase of the LPT flow capacity by increase of the last stages blades length inevitably causes a remarkable efficiency reduction caused by flow separation in the last stages root zones.

An alternative method of the flow into condenser capacity increase is proposed. The alternative structure is a two-tier LPT with the new fork-shape last stage blade which allows a 50% steam flow capacity increase without the last blade length increase.

Strength and vibration analysis of the new “fork shape” blade shows prospects for its practical application in LPT with 1.5 times increase of the steam flow into turbine condenser. Possibility of the fork shape blade manufacturing is approved by manufacturing its full scale demonstrator mockup.
REFERENCES
*Regulations for steam turbine blades vibration tuning*, RTM 108.021.03—77