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Development of last stage blade of 13K215 turbine intermediate pressure module

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Abstract Paper is considering the purpose and the process of development of last stage blade for intermediate pressure module of 13K215 steam turbine. In the last 20–30 years most of the steam turbine manufacturers were focused on improving such a turbine mainly by upgrading low pressure module. In a result of such a modernization technology were changed from impulse to reaction. The best results of upgrading were given by developing low pressure last stage blade. With some uncertainty and based on state of art knowledge, it can be stand that improving of this part of steam turbine is close to the end. These above indicators show an element on which future research should be focused on – in the next step it should be intermediate pressure module. In the primary design the height of intermediate pressure last stage blade was 500 mm but because of change of technology this value was decreased to 400 mm. When to focus on reaction technology, the height of the last stage blade is related to output power and efficiency. Considered here is the checking the possibility of implementing blades, in a reaction technology, higher than 400 mm and potentially highest. Article shows a whole chosen methodology of topic described above. It leads through the reasons of research, limitations of 13K215 steam turbine, creation of three-dimensional models, fluid flow calculations, mechanical integrity calculations and proposed solutions of design.

Keywords: Efficiency; Strength calculation; Steam turbine; Last stage blade; Fluid flow calculation

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Nomenclature

13K215	–	turbine model
C_{Z2}	–	outlet velocity
CFD	–	computational fluid dynamic
G-0 Blade	–	stationary blade of the last stage
GE	–	General Electric
HP	–	high pressure
IP	–	intermediate pressure
KΩ	–	leaving losses
L-0 Blade	–	rotating blade of the last stage
LP	–	low pressure
LSB	–	last stage blade
OEM	–	original equipment manufacturer
SAFE	–	Singh's advanced frequency evaluation

1 Introduction

Taking into account the fast-growing demand for green energy production which results in decreasing energy production from fossils [1], steam turbine development might be considered irrelevant. However, the fact is that modern zero-emission technologies are not capable of entirely covering the needs, mostly due to the lack of efficient energy storage capabilities. Therefore, there is still room for the development of conventional units to minimize operational costs and maximize their performance. The first goal is steam turbines' efficiency improvement and the second is an extension of their service life. The benefit for the environment is obvious. Higher efficiency guarantees lower fossil fuels consumption, which reduces carbon dioxide production.

This paper presents a technical solution for the performance development of the 13K215 steam turbine (Fig. 1) intermediate module last stage blade, which has been neglected so far by the original equipment manufacturer due to technical difficulties described further in this article. Moreover, the article considers various aspects of development direction.

Steam path performance is a key factor for the overall steam turbine efficiency [2]. Especially critical for steam path performance is the last stage blade (LSB) due to non-homogenous steam conditions along with the last stage blade height. This results in changes of blade profiles and stager angles along with the blade height. The last stage blade design must also accom-

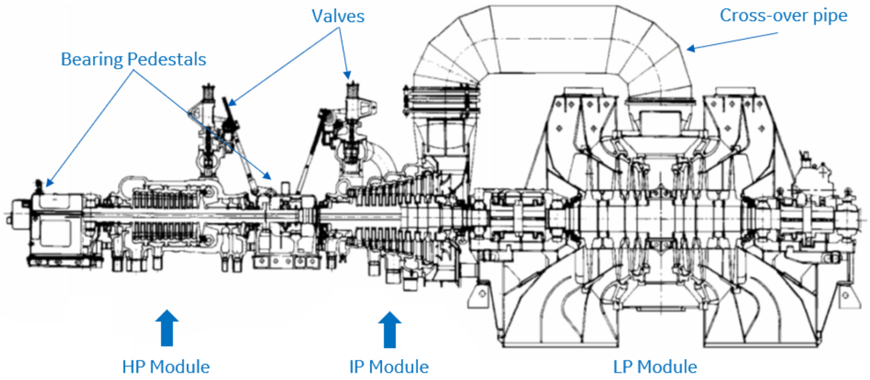

Figure 1: Longitudinal section of a 13K215 turbine.

modate strength and dynamic aspects. Moreover, for the considered case there are design limitations related to the axial and radial space, which can be utilized to fit the last stage blade. These limitations in the considered intermediate module last stage blade development follow from the constraints placed on the 13K215 steam turbine retrofit. The steam turbine retrofit design shall cope with available space left in the original steam turbine design (i.e. before retrofit – see Section 2). The existing constraints prevent the turbine manufacturer from using its standard LSB technology. The paper describes a modern LSB solution developed to meet the demanding efficiency requirements of the 13K215 steam turbine retrofit, taking into account the existing constraints, which have been a development obstacle for the original equipment manufacturer for years. The developed solution of the intermediate module last stage blade includes an integrated shroud, which is a new feature for this family of blades. This design is expected to reduce extensive tip loses to improve blade performance. Moreover this design bonds blades into one row which result in moving away blades' natural frequencies from operation speed.

2 Modernizations of 13K215 steam turbine

The 13K215 turbine was designed and manufactured by Mechanical Works Zamech in Elblag, Poland, whose heritage is part of General Electric – GE Power now. Originally, it was an impulse type turbine, which consisted of three double-shell modules. The high pressure (HP) and intermediate pressure (IP) parts are single-flow, whereas low pressure (LP) is doubleflow. The live steam (at the HP inlet) parameters are 127.5 bar and 535°C. The reheated steam parameters (at the IP inlet) are 19.5 bar and 535°C.

The 13K215 turbine retrofit changed the energy conversion technology from impulse to reaction and was done in a few phases. The first phase was focused on the maximum performance gain, which entailed the low-pressure module retrofit. The main feature of the LP module retrofit was a replacement of the Baumann stage by highly efficient blades able to cope with the full LP steam flow. The next modernization phase was the HP and IP retrofit. More details can be found in [3].

3 Literature review

The blades are subjected to significant centrifugal force, making them critical parts of the whole steam turbine [4]. Blade design methods are described in numerous papers and books [5–8]. The blade design process includes various aspects like performance, fluid dynamics and mechanical integrity (i.e. losses reduction, stresses, vibrations, etc.). Moreover, geometrical and topological variables (i.e. shape of blade airfoil, type of blade root, etc.) have to be considered [9].

Successful operation of last stage blades is determined also by their dynamic behaviour. The free-standing blade has low natural frequencies and might be excited by the operational rotational speed. Therefore, in order to increase natural frequencies, design features like snubbers are implemented [10, 11]. Unfortunately, these features result in a blade efficiency decrease. However, the design solution presented in this paper, not only takes into consideration geometry constraints relevant for its scope of application but also ensures reliable operation due to the high blades row stiffness and high efficiency due to elimination of tip leakage losses. To prove the feasibility of the design, computed fluid dynamic simulations were performed to optimize the design and provide boundary conditions to blade mechanical integrity assessment.

4 Justification for 13K215 intermediate pressure last stage blade development

13K215 intermediate pressure last stage blade development was justified by a relevant analysis taking into account economical aspects. The analysis determined an efficiency increase in function of blade length, which allows covering outlet sizes in the assortment. The blade length of 400 mm was considered as an upper threshold.

4.1 Challenges related to elongated 13K215 last stage blade

In a starting point, outlet velocity and exhaust losses were analyzed and results were compared to values obtained for the blade with the threshold height of 400 mm. Table 1 shows results of the analysis depending on the blade heights.

Table 1: Comparison of outlet velocity (C_{Z2}) and leaving loses (KΩ).

Property	Length of last stage blade		
	322 mm	358 mm	400 mm
Outlet velocity, C_{Z2}	125.3%	109.5%	100%
Leaving losses, KΩ	149.3%	117.5%	100%

The analysis results were used to estimate the financial benefits of design implementation. A decrease of outlet velocity caused by the enlarged outlet area and lower exhaust losses resulted in a higher power. There is in Table 2 a results summary for considered design cases. Design case with 400 mm long last stage blade was considered as baseline. For other considered designs, relative differences of output electric power (ΔP_{el}) and heat rate (ΔHR) were calculated. An increase in electric power is observed with a simultaneous decrease in heat rate. The presented values refer to the turbine output power and heat rate, not only to the single stage.

Table 2: Output electric power (P_{el}) and heat rate (HR) for considered variants.

Length of last stage blade (mm)	P_{el} (%)	ΔP_{el} (%)	HR (%)	ΔHR (%)
322	99.87	–0.13	100.14	0.14
358	99.95	–0.05	100.05	0.05
400	100	0	100	0

A substantial difficulty related to long blades development is high blade loading caused by centrifugal forces [10]. Therefore, mechanical integrity aspects were thoroughly considered and results are presented in this article.

4.2 Estimated efficiency benefit

The main drivers of any modernization are power output and efficiency increases. In the case of 13K215 last stage blade modernization, the main enabling factor was not different from other cases from the industry. To support the decision making process, an analysis has been made to estimate IP module power increase related to LSB modernization. The results of this study are illustrated in Fig. 2. The presented values for LSB lengths of 325 mm, 358 mm, and 400 mm were obtained directly from the relevant analysis, whereas the power increase value for LSB of 500 mm length was linearly extrapolated. It can be summarized that each 1mm of additional LSB height above 400 mm results in an approximately IP module power increase of 0.03%. It is obvious that linear extrapolation is an optimistic assumption. However, with a conservative correction factor assumed for the purpose of economic justification, an IP module power increase is substantial enough to proceed with the development and implementation.

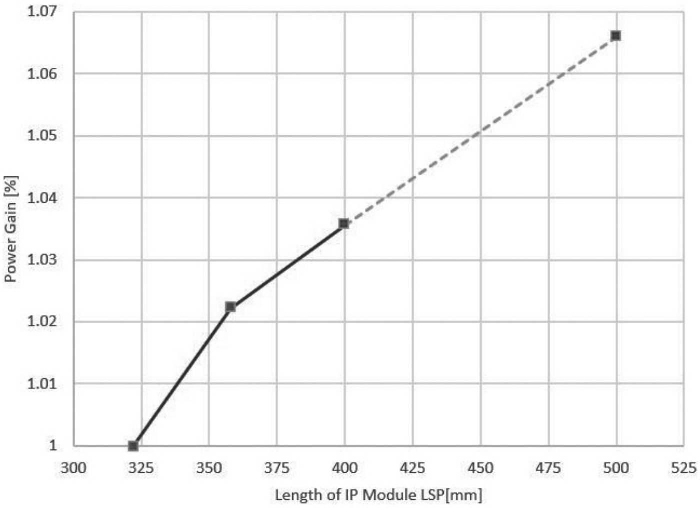

Figure 2: Estimated increase in power versus last stage blade height.

4.3 Geometrical restrains

As a forementioned above, there are certain restrains related to the existing design solution, i.e. welded outlet of the IP casing (its location is shown in Fig. 3). This entailed a feasibility study to determine the maximum viable LSB length fit for the purpose.

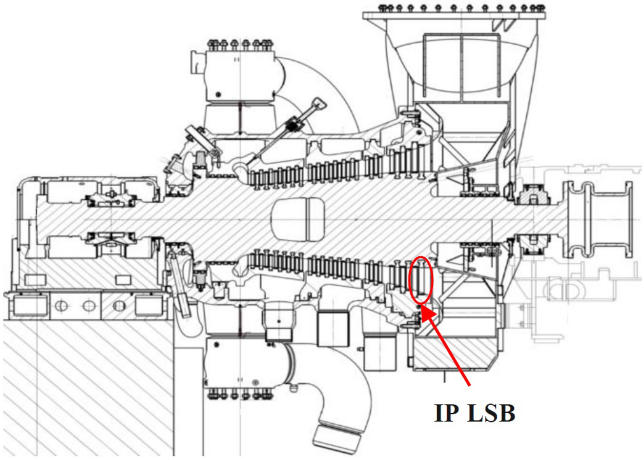

Figure 3: Longitudinal section of intermediate pressure module of 13K215 turbine after modernization. The section includes the current intermediate pressure last stage blade design, i.e. of last stage blade length equal to 400 mm.

The following geometrical restrains have been taken into account:

- position of blade carrier fixation,
- minimum required thickness of blade carrier shell above the G-0 blade (some additional space can be managed by change of blade carrier design),
- minimum required blade carrier shell thickness,
- outlet diffuser optimum shape,
- maximum relative thermal expansion of IP steam path.

As an outcome of the feasibility study, a statement can be given that the maximum viable LSB height, which can be installed in the IP Module of the 13K215 turbine is equal to 502 mm. For further proceedings, LSB of 500 mm length has been selected.

5 Research and analysis methodology

As already mentioned, the main challenges related to the design of a 500 mm long LSB for the 13K215 IP module are coordination of flow, strength and natural frequency analyses. Due to the relatively long LSB, it was necessary to develop an LSB row damping and stiffening solution at the LSB shroud. Standard solutions are not feasible in the case of the considered LSB design, therefore it can be considered as first of a kind and a novelty in the field area.

5.1 Designs components of last stage blade

A starting point of LSB development was a design of blade profiles along the LSB height and LSB root design, which entailed the LSB assembly method. As an initial design subjected to a further optimization process, a 500 mm long LSB with a twisted profile with a reaction ratio of approximately 0.42 was developed. It needs to be noted that the LSB design was free of dumping wires used in the original design, inefficient from a performance perspective, which resulted in a loss and, thus an efficiency reduction. A key factor in the research was avoidance of efficiency degradation features. For the purpose of developing the BladeGen [12], an Ansys [13] build-in extension was used instead of well-known but not state-of-the-art profile generation methods (e.g. according to Rusanov [14] or Lampart and Ershov [15]).

The next considered design feature was the blade root. In the Zamech OEM solution, it was a pinned root. This solution is very labour-intensive and thus expensive. Hook root design solutions have been the most used in the existing GE Power solutions for the IP LSB. The fleet experience has shown that hook root design can be used for a limited blade height due to strength constraints. Therefore, fir-tree root has been chosen as a feasible solution although due to shorter LSB in comparison to standard applications, smaller acting forces and a high LSB number in a row (90 pieces), the root has been also developed. The final development and research effect related to root development is shown in Fig. 4.

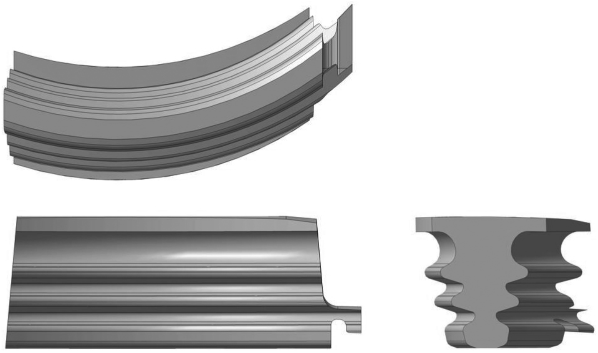

Figure 4: Root of developed last stage blade.

The last feature considered in the research was the LSB shroud. Initially, a shroud with a diagonal cut to the direction of steam flow was considered as a baseline for further elaborations. It has to be pointed out here that the initial design does not meet the design requirement. The LSB shroud has a few purposes and one of them is the damping of the blade's natural vibrations [5]. In the Zamech OEM solution, this task was carried out by two damping wires located along the height of the blade but to avoid the decreased efficiency in the developed designed blade, this alternative was omitted. Some of the considered shroud design variants are shown in Fig. 5. These design variants imply certain difficulties related to the LSB manufacturing process and are caused among others by tight machining tolerances. However, mitigation measures have already been taken to overcome manufacturing issues.

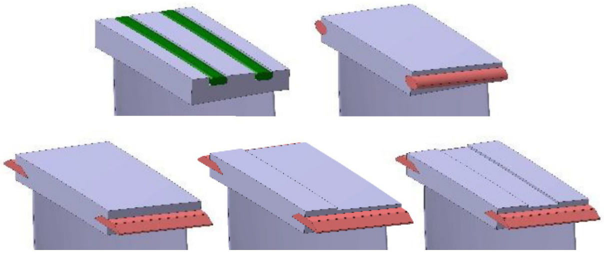

Figure 5: Some of considered solutions of shroud design.

5.2 Research methodology

The research methodology was based on steps performed in the following order:

1) creation of a three-dimensional (3D) model of blade airfoil;
2) elaboration of boundary conditions;
3) LSB profile thermodynamic assessment and optimization in Asnsys CFX;
4) elaboration of LSB 3D model including all design features like root, airfoil blade, shroud and part of rotor to properly model LSB-rotor interaction during mechanical integrity analysis;
5) materials selection;

6) strength analysis and design optimization from a mechanical integrity perspective;
7) modal analysis to determine an interference and Campbell diagrams for the design.

Please note that only results for the optimized design are shown in this paper.

6 Fluid dynamics analysis

The fluid flow analysis was performed to provide refined results of IP module power increase due to the optimized LSB design. The main target of the fluid dynamic analysis, performed in the commercial computational fluid dynamics (CFD) softwar for turbomachinery applications Ansys CFX [16], was LSB airfoil optimization.

6.1 Assumptions of the computational model

LSB has a twisted airfoil to cope with inhomogeneous steam flow conditions along the channel height. These boundary conditions have been included in the analysis model by adding a stationary blade to account for inconstant steam inflow at the LSB (Fig. 6). There are 60 stationary and 90 rotating blades in the stage. For the purpose of optimization to reduce computation

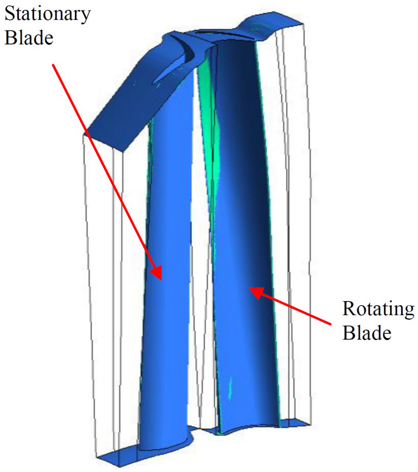

Figure 6: Stage configuration with stream separation indication.

power demand which resulted in border analysis coverage for optimization purpose, one rotating and one stationary blade were used in the model. To achieve this, the minimum distance of the interface ('pitch') was increased to 1.5 (6° sections were used in the stationary blades row and 4° sections were used in the rotating blades row).

This efficient approach from computational power demand caused also the necessity of scaling of the inflow to the stationary blade. To achieve the optimal angle of inflow on the stationary blade, a cylindrical system was applied and the following formulas were introduced for each of its components:

- axial component $= \sin \alpha_{\text{hub}}$,
- radial component $= \cos \alpha_{\text{hub}}$,
- circumferential component $= (r_i - r_{\text{hub}}) \dfrac{\Delta\alpha}{h_{\text{inlet}}}$,

where: α_{hub} – an optimal angle of the steam flow at the hub diameter, r_i – various radius in the range from the hub to the shroud, r_{hub} – radius of the hub, $\Delta\alpha$ – maximum angle change; h_{inlet} – height of the inlet.

The discrete model was generated in TurboGrid [17], an Ansys integrated tool dedicated to turbines, compressors etc. This modeling strategy enabled the use of a variable inflow angle on the stationary blades along the steam channel height. It also allowed to model more reliable steam conditions in the steam channel by avoiding artificial losses related to detachment of the stream caused by modeling insufficiencies.

The thermodynamic inlet steam conditions assumed for the optimization have been taken from the current GE design, valid for the whole unit's frame. This assumption was made to compare the current GE LSB design to the presented in this article solution researched and developed by using state of the art tools and approach. However, there is a development potential for steam path channel of 13K215 IP module to fully account for benefits from the shown optimized LSB design.

6.2 Optimization of the rotating blade airfoil

First optimization targets were to identify locations where there is a risk of flow separation and to verify the stage degree of reaction, stage efficiency and power. Figure 6 shows the baseline model used to identify flow separation areas, whereas Fig. 7 presents these areas for five selected LSB radiuses. Separation areas are strongly visible at spans 0.7 and 0.9. This analysis spotted LSB optimization areas and directions. The optimization strategy included LSB profile and LSB twist angle optimization. Once optimized, the LSB design is free from flow separation.

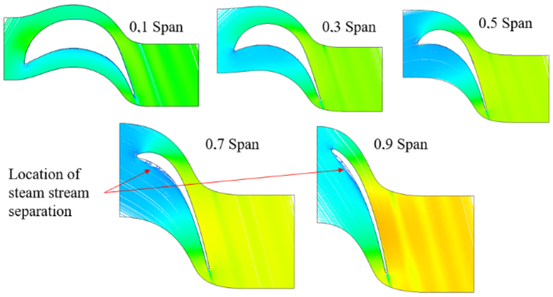

Figure 7: Velocity streams at various span of blade airfoil.

As an outcome of the analysis, a convenience was reached that the development of LSB and related efforts are fully justified. The power increase generated by the presented state-of-the-art LSB design in comparison to the GE solutions currently available on the market, including all sources of losses [18], varies between 1.75–2.15%, whereas the profile efficiency is 94.5%. These refined values are lower than initially assessed but still provide justification for implementation in the industry.

6.3 Boundary conditions for mechanical integrity assessment

LSB performance is one of the design aspects. The computational fluid dynamics results have been used as boundary conditions in LSB mechanical integrity analysis. Only after mechanical integrity analysis, a robust LSB design can be claimed.

7 Last stage blades strength and dynamic assessment

Mechanical integrity calculations have been made to verify and optimize LSB mechanical design. Moreover, mechanical integrity assessment results have been confronted with GE Power internal criteria. Therefore, design criteria for the state-of-the-art shroud and root design had to be developed in parallel to the LSB design.

The mechanical integrity analysis has been made in Ansys and the 3D models for calculations came from 3D computer-aided design software CATIA V5 [19].

7.1 3D models for mechanical integrity analysis

The LSB geometry model was developed in the first step of strength analysis. Then, relevant thermal boundary conditions and mechanical contacts were defined. For the sake of reliable analysis and comparison of results, the models of the considered design variants have been fully aligned.

Figure 8 shows the model prepared for the purpose of mechanical integrity analysis. The model consists of one LSB with the relevant part of the rotor. The upper part of the shroud was cut by a bent surface. This approach|was applied to achieve more precise results for the interaction between adjacent blades and the pin. The rotor groove where LSB is installed was modelled including all adherence surfaces and clearances relevant for LSB fastening.

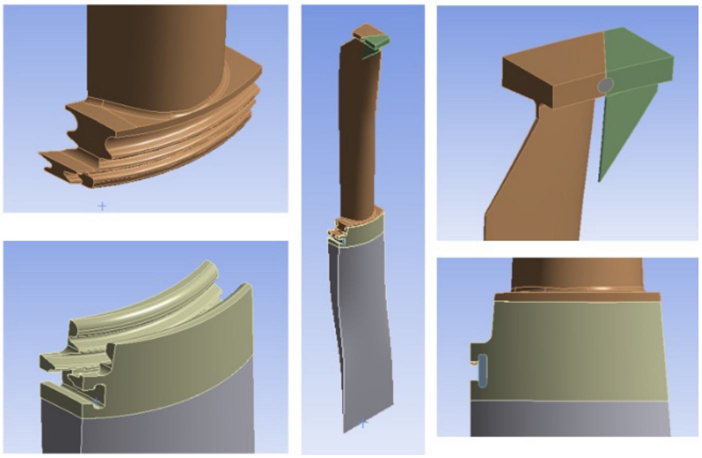

Figure 8: 3D geometry model.

7.2 Strength analysis

The newly developed state-of-the-art LSB concept has been assessed according to GE Power requirements. LSB is subjected to two types of stresses:

static and dynamic and they consist of centrifugal tensile and bending stresses, steam bending loads and synchronous resonance stresses of the blade during the nominal speed of the turbine [20]. Stresses indicated in the blade components have been compared to permissible values for relevant LSB design features: shroud, airfoil, root and transitional radiuses. For the purpose of LSB mechanical integrity analysis, the state-of-the-art approach to material properties modelling has been used.

As it has already been explained, the reliable LSB operation throughout its design life has been ensured by binding LSB at their shrouds, which is a modern design eliminating performance losses caused by design features at the LSB airfoils, e.g. dumping wires or snubbers.

The pin selection and their shape optimization was broadly considered in the research. However, only two cases are presented in this paper: with a trapezoidal and round pin. The maximum stress was observed in the transitional radius between the root and airfoil and at the leading edge in the upper location (Fig. 9). For both solutions, most stressed areas are in the same location but for the round pin results are slightly lower.

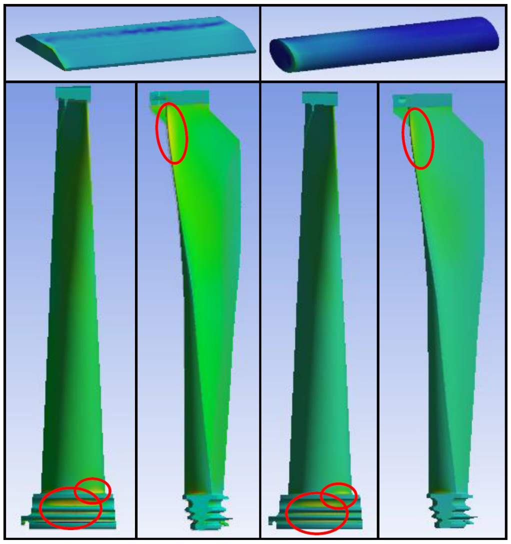

Figure 9: Results of blade airfoil strength analysis: a) concept with a trapezoidal pin (left), b) concept with a round pin (right).

Other relevant LSB locations and their design features have been verified against acceptance criteria. In all cases, these acceptance criteria were met.

7.3 Dynamic assessment

The analysis of natural vibration frequencies was conducted to verify the LSB design against vibration damping requirements. This part of research was the most time consuming to achieve results that satisfy requirements and prevent LSB from working in resonance. The dynamic assessment has considered all relevant loads related to the last stage blade operation regime [21–23].

In general, there are a few measures that can be taken to avoid natural frequencies interference [21, 24]:

- change the number of blades,
- modify blade flexibility,
- change shroud flexibility,
- move the operating speed range.

In the case of the LSB design presented in this paper, optimization of shroud geometry has been used to influence the LSB flexibility. The optimization process has been based on shroud thickness, undercuts, various pin positions, etc. Figure 10 shows the interference diagrams (SAFE diagrams) [25] for the two presented design concepts. For the trapezoidal pin at the 9th nodal diameter, there is a risk of interference. Due to the fact that this point has a higher nodal diameter than the 8th, this does not entail the design variant disqualification. In this case, dynamic stresses at specific conditions

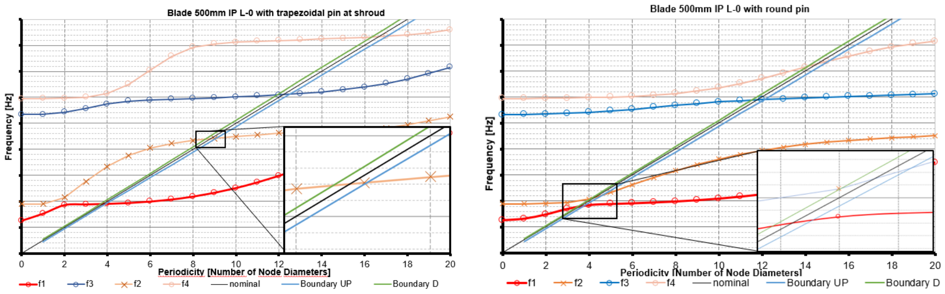

Figure 10: SAFE diagrams for 3000 rpm a) concept with a trapezoidal pin, b) concept with a round pin: f1–f4 – vibrations modes, nominal – first nozzle passing frequency for nominal speed, Boundary UP – first nozzle passing frequency for max. permissible deviation of speed, Boundary D – first nozzle passing frequency for min. permissible deviation of speed.

have to be verified. Therefore this design variant is considered to be feasible after proper justification.

The solution with a round pin has also been considered. The Singh's advanced frequency evaluation (SAFE) diagram was prepared. A conclusion can be drawn that the shroud connection flexibility has a significant impact on LSB natural frequencies and the shape of the mode plots (Fig. 10). As for the previously demonstrated design concept with a trapezoid pin, the design concept with a round pin fulfils all criteria.

The Campbell chart has been a final step of design verification. As it can be seen in Fig. 11 for the round pin design variant, natural frequencies of the 1st and 2nd modes plots do not cross the forbidden zones as per GE Power rules. It means that there are no concerns regarding the LSB dynamic behavior, i.e. the operation outside resonance regions is anticipated.

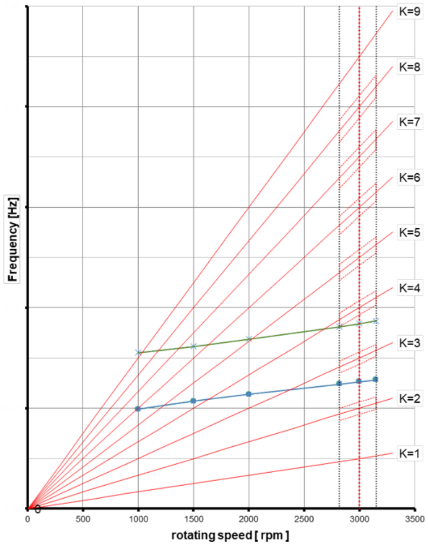

Figure 11: Campbell diagram for two first modes of vibrations (lines with markers) at node diameter No. 0, ($n = 0$), K1–K9 – engine order lines.

7.4 Summary of stress analysis

It can be concluded that one of the optimized LSB concepts fulfilled all internal GE design criteria.. Moreover, from strength and dynamic points of view, it is feasible to design a blade higher than 400 mm (which is the current threshold for GE design), up to 500 mm without any performance deteriorating features like dumping wires or snubbers. Further development steps are related to fatigue analysis.

8 Conclusions

In this paper, the process of optimization and implementation of different concepts in blade designs was presented. The topic and related design and optimization process were considered from several perspectives: strength analysis, fluid flow and dynamic optimization. The state-of-the-art solution presented in this article shows a robust LSB design free from the risk of failure due to dynamic behavior.

All of the optimization steps described are just highlights of all analyses done for the purpose of research and development. Concepts, which were shown have a potential to be further developed in other fields, i.e. locking of pins, manufacturing accuracy difficulties, laboriousness of execution, application costs, etc. Certainly, industry implementation must follow a profit and loss assessment. However, the current stage of development clearly justified economic and environmental benefits.

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References

[1] Choudhary A., Prasad E.: *Steam Turbine Market by Design (Reaction and Impulse) and Application (Power generation, Petrochemical, Oil & Gas and Others): Global Opportunity Analysis and Industry Forecast, 2020–2027.* Market Research Report 2019. Allied Market Research, 2020.

[2] Termuehlen H., Emsperger W.: *Steam turbine technology.* In: Clean and Efficient Coal-Fired Power Plants. ASME Press, New York 2003, 60–62.

[3] Dominiczak K., Radulski W., Banaszkiewicz M., Mróz K., Bondyra R.: *Thermal stress limiter for 13K215 steam turbine retrofit in Połaniec Power Plant, Poland.* J. Power Technol. **96**(2016), 4, 285–294.

[4] Singh M.P., Lucas G.M.: *Blade design and analysis for steam turbines.* McGraw-Hill, New York 2011.

[5] Reddy A.S., Ahmed M.D.I., Kumar T.S., Reddy A.V.K., Bharathi W.P.: *Analysis of steam turbines.* Int. Refer. J. Eng. Sci. **3**(2014), 2, 32–48.

[6] Bloch H.P., Singh M.P.: *Steam Turbines Design, Application, and Re-Rating.* McGraw Hill Prof., 2008, 109–124, 188–218.

[7] Jansen M., Ulm W.: *Modern blade design for improving steam turbine efficiency.* In: Proc. 1st Eur. Conf.: Turbomachinery – Fluid Dynamic and Thermodynamic Aspects, Erlangen, 1–3 Mar 1995.

[8] Segawa K., Shikano Y., Tsubouchi K., Shibashita N.: *Development of a highly loaded rotor blade for steam turbines.* JSME Int. J. B-Fuid. Therm. Eng. **45**(2002), 4, 881–890.

[9] Shimoyama K., Yoshimzu S, Jeong S., Obayashi S, Yokono Y.: *Multi-objective design optimization for a steam turbine stator blade using LES and GA.* J. Comput. Sci. Technol. **5**(2011), 3, 134–47.

[10] Mohan R.S., Sarkar A., Sekhar A.S.: *Vibration analysis of a steam turbine blade.* In: Proc. 43rd Int. Cong. Expo. on Noise Control Engineering: Improving the World Through Noise Control., Internoise 2014 (J. Davy, T. McMinn, N. Broner, Ch. Don, L. Dowsett, M. Burgess, Eds.), Melbourne, 16–19 Nov. 2014.

[11] Heidari M., Amini K.: *Structural modification of a steam turbine blade.* In: Proc. IOP Conf. Ser.: Mater. Sci. Eng. **203** (2017), 012007.

[12] https://www.ansys.com/products/fluids/ansys-blademodeler (accessed 16 April 2021).

[13] https://www.ansys.com (accessed 16 April 2021).

[14] Rusanov A.V., Paschenko N.W., Kosianova A.I.: *Method for analytical profiling of axial turbine blade rows.* East Eur. J. Power Technol. **2**(2009), 7, 32–37 (in Russian).

[15] Lampart P., Ershov S.: *3D shape optimisation of turbomachinery blading.* TASK Quart. **6**(2002), 1, 113–125.

[16] https://www.ansys.com/products/fluids/ansys-cfx (accessed 16 Apr. 2021).

[17] https://www.ansys.com/products/fluids/ansys-turbogrid (accessed 16 Apr. 2021).

[18] Yoon S., Vandeputte T., Mistry H., Ong J., Stein A.: *Loss audit of a turbine stage.* J. Turbomach. **138**(2016), 5, 051004.

[19] https://www.3ds.com/products-services/catia/ (accessed 10 March 2021).

[20] Stuck Z., Schurdak S.: *Steam turbine blade design.* In: Proc. 20th Ann. Freshman Conf., 14 April 2012, 2214.

[21] Rzadkowski R., Kubitz L., Maziarz M., Troka P., Dominiczak K., Szczepanik R.: *Tip-timing measurements and numerical analysis of last-stage steam turbine mistuned bladed disc during run-down.* J. Vib. Eng. Technol. **8**(2020), 3, 409–415.

[22] Rzadkowski R., Lampart P., Kwapisz L., Szymaniak M., Drewczyński M.: *Transient thermodynamic, thermal and structure analysis of a steam turbine during its start-up.* In: Proc. ASME Turbo Expo 2010: Power for Land, Sea and Air, Vol. 4: Heat Transfer, Glasgow, June 14–18, 2010, GT 2010-22813, 2010, 1103–1112.

[23] Dominiczak K., Rzadkowski R., Radulski W., Szczepanik R.: *Online prediction of temperature and stress in steam turbine components using neural network.* J. Eng. Gas Turbines Powers **138**(2016), 5: 052606.

[24] Singh M.P., Vargo J.J., Schiffer D.M., Dello J.D.: *SAFE diagram – A design and reliability tool for turbine blading.* In: Proc. 17th Turbomachinery Symp. Houston, 1988, 93–102, Texas A&M Univ. Turbomach. Lab.

[25] Singh M.P.: *History of evolution, progress and application of SAFE diagram for tuned and mistuned systems.* In: Proc. 42nd Turbomachinery Symp., Houston, Oct. 1-3, 2013, Texas A&M Univ. Turbomach. Lab.