ACGT2014 - 0007

Analysis and Optimization of Partial Admission Stages

Dr. Leonid Moroz, Dr. Boris Frolov, Oleg Guriev

Company: SoftInWay Inc.
ADDRESS: 15 New England Executive Park, Burlington, MA, USA
Phone: 1-781-685-4942, FAX: 1-781-685-4601, E-mail: info@softinway.com

ABSTRACT

Small and midsize turbines are widely used in various sectors of economics for electricity generation or as mechanical drivers. The small volume flow rate of these machines at the inlet requires a small minimal passage area for the first or some upstream stages. This leads to short blade heights and small gauging angles that together compose the passage area and result in significant efficiency reduction. A well-known approach to avoid this is to admit steam to the part of the arc. The partial admission is widely used in the first (control) stage and sometimes even for the group of stages in small turbines. In general, the efficiency of the first partial stage and downstream located stages depends on many parameters. Some major parameters are as follows: total level of admission partiality, number and tangential location of admission arcs, and geometry of mixing chambers between first and downstream located stages. Proper combination of the above mentioned parameters may lead to improvements in turbine performance.

In this article, the interaction of the control stage with partial admission and one downstream stage without partial admission for a high-speed mechanical drive turbine is considered. The comparison of 1D, 2D and CFD calculation results will be performed to reveal full 3D flow structure influence on the performance of both stages.

NOMENCLATURE

F_1 – total nozzle passage area;
D - Outlet mean diameter, D, mm
b_2 – blade chord;
l_2 – blade height;
N_{segm} – number of admission segments;
η_{bl} - blade efficiency;
k_w – windage losses factor
U – tangential velocity
C_0 – theoretical spouting velocity
e – partial admission ratio
G – mass flow
k – correction factor;
μ - seal mass flow coefficient (depends on type and size of seal teeth);
F – clearance circumferential area;
P_0, v_0 – pressure and volume before clearance;
$\varepsilon=P_1/P_0$ – steam pressure relation before and after clearance;
z – number of seal teeth;

ITRODUCTION

This article represents the results of a numerical study of a two stage compartment of a 10 MW industrial turbine for a compressor drive. The turbine was originally designed as a single cylinder reaction machine with partial admission control stage.

The general objective of the study is to investigate the potential for turbine efficiency improvement for a turbine’s future upgrade and in this article we are focusing on the partial admission control stage redesign and downstream stage performance improvement options.

To be more specific:

- for control stage redesign options, we are considering efficiency improvements by applying an algorithm of optimization for the main stage parameters, strongly influencing stage performance, like level of partiality, blade height and angles. The calculation study was performed with 1D and 2D design algorithms, integrated into the Preliminary Design module of the AxSTREAM™ design suite
- for downstream located stages, to mitigate the negative effects of partial admission, the application of multiple admission arcs with uniform circumferential distribution, has been studied. Considering complex effects of partial admission on flow structure and parameters distribution, the study was done with 3D CFD simulation of a two stages compartment with mixing a chamber between the first and second stages.

3D CFD results have also been used to verify and correct tip leakage loss models for the partial admission stage, which is routinely used in 1D and 2D codes for performance calculation.

PARTIAL ADMISSION STAGE OPTIMIZATION

The analysis of the partial admission was performed on the first two stages of a high-speed mechanical drive reaction steam turbine with the live steam pressure of 42 bar (609.2 psi) and enthalpy of 3279.7 kJ/kg (1410 btu/lbm).

The first stage of this turbine (the control stage) has partial admission of 50% and the second one -the full admission arc. Due to the fact that non-axisymmetric flow, which comes out from the first stage, doesn’t dissipate instantly, the second stage, in spite of being full admission, also suffers from the partial admission. Not to such an extent as the first one, though. A special chamber (plenum) is present after the first stage to reduce the partial admission influence.

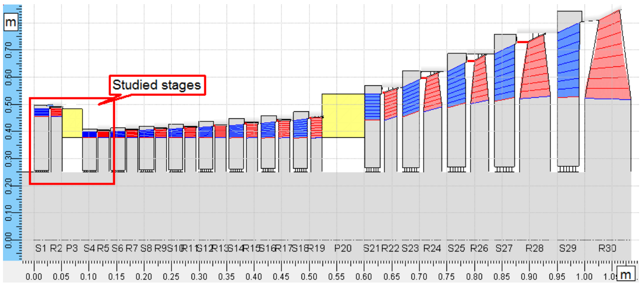

Fig.1. Turbine flow path meridional view

Table. 1 Main geometrical parameters of studied part

Parameter	Stage 1 (control stage)	Stage 2
Inlet mean diameter, mm	469.75	387.28
Outlet mean diameter, D, mm	469.75	387.7
Number of nozzles, Zn	40	46
Number of blades, Zb	96	80
Nozzle height, Hn, mm	17.75	10.5
Blade height, Hb, mm	18.75	11
Nozzle gauging angle at mean diameter, Alfa1, deg	16	21.35
Blade gauging angle at mean diameter, Beta2, deg	20	29.1
Nozzle chord, Bn, mm	50	37.5
Blade chord, Bb, mm	21.9	21.9

The crown minimum passage area (3D throat) is formed by aerofoil height and throat area (gauging angle), and is one of the fundamental parameters that influences the flow in turbines. Thus, proper consideration of this parameter has to be performed during the design or redesign. For turbines with the relatively small volume flow rate at the inlet, the crown minimum passage area may require blade heights and gauging angles that are too small from a manufacturing point of view.

For the core parameters influence study, the Preliminary Design module uses the Meanline Inverse task solver that studies the numerous combinations of design parameters within a small period of time.

Analysis approach

Preliminary design is a procedure of flow path geometry obtained using a predefined set of boundary conditions and design variable ranges. For purposes of machine geometry generation and evaluation in preliminary design, the inverse task was selected because of the possibility to quickly review a number of designs, without certain defined geometry, that matches the selected criteria. A uniform approach can be applied to all types of turbomachines, but different design parameter combinations should be used for each particular one. The sequence of the preliminary design steps is next:

1. Combination of design parameters for selected machine type
2. Variation of design parameters in selected ranges
3. Geometry generation from selected set of parameters and constraints verification
4. Inverse task calculation
5. Selection of best design based on selected criteria

As it is stated above, the meanline inverse task was selected as the most satisfactory for rapid and flexible preliminary design generation. Depending on the machine type, the selection of combined design variables is different, and for axial machines it can be described by this data:

- Number of stages
- Specific diameter (hub, mean, tip)
- Geometry constraints (blade height, gauging angle, diameter/blade height ratio)
- Velocity ratio or work coefficient
- Axial velocity ratio distribution between stages
- Additional constraints on flow path diameters and axial sizes, which can be applied to restrict search to more specific ranges

Design generation should be performed based on some initial set of parameters. A set of multiple design variables should be studied and all possible design combinations considered, so different approaches could be used for this operation to cover the whole range. The proposed method is the LPτ search, that generates quasi-random sequences. These are uniformly distributed sets of $L=M^N$ points in N-dimensional unit cube I^N. The advantage of the algorithm selected is that unlike pseudo-random sequences, quasi-random sequences provide uniform distribution in spaces. This also makes it possible to increase the search points density by setting additional parameters.

From the selected set of parameters, geometry is generated and evaluated in the inverse task solver. Additional parameters, such as clearances, chords, and number of blades are selected based on the prescription given for the specific machine type and sizing ranges, as specified above.

A general theoretical overview of the presented approach methodology for an axial turbine and compressor is given in [5].

The essence of the inverse task calculation is solving an equation set to find unknown angles α_1 and β_2, when velocity coefficients φ and ψ (i.e. losses) are supposed to be predefined based on some known correlations or prescribed efficiency charts (such as Spencer-Cotton, Smith, and Baines diagrams). The task of determining α_1 and β_2 angles is to find the maximum search criterion as a function from set of design variables:

$$\max \eta_i(\vec{u}) \qquad (1)$$

To improve the design selection flexibility, the best possible solution can be chosen based on predefined search criteria such as: internal or polytropic efficiency maximum and maximal power. The process chain described below is representing a machine synthesis procedure.

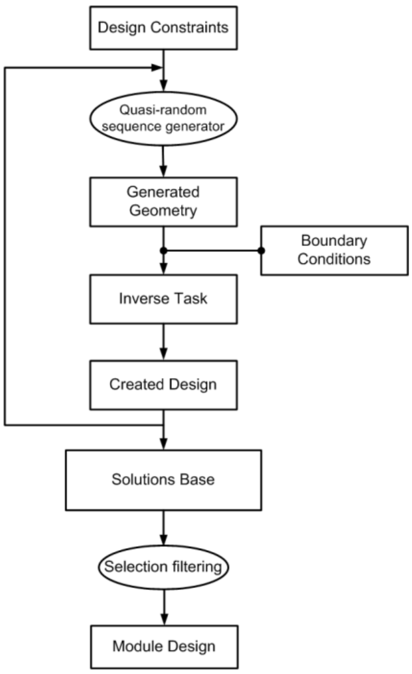

Fig.2. Design procedure scheme

The Craig&Cox loss model [1] was used as a primary (profile) loss model and the Stepanov loss model, [2] as a secondary one. The combination of these two models proved the agreement with the experimental data and performance tests for various turbine types [3].The partial admission losses are considered as a sum of windage (pumping, ventilation) losses and sector end losses. The windage loss refers to the effect of pumping in the inactive blade channels rotating in a steam filled casing. The sector end loss originated from the filling and emptying of the passages as the blades pass through the active sector. Stodola [5], Traupel [6, 7] and Roelke [8] independently found the windage power loss to be proportional to the cube of blade speed.

The equation (1) represents windage loss:

$$X_{wind} = \frac{k_w}{\sin A1gaug} \cdot \frac{(1-e)}{e} \cdot \left(\frac{U}{C_0}\right)^3 z \qquad (1)$$

And equation (2) -sector end loss:

$$X_{segm} = 0,25 \frac{b_2 l_2}{F_1} \left(\frac{U}{C_0}\right) \eta_{bl} N_{segm} \qquad (2)$$

The application of the approach mentioned above allowed the generation of thousands of designs within a short period of time, thus making it convenient to analyze the influence of gauging angle, blade height and partial admission on performance. For the studied case, all three parameters were changed simultaneously so that it was possible to find the best combination. Each point on the charts below represents a different design with its own combination of geometrical parameters.

Nozzle height influence

As it was mentioned before, the nozzle height value has its minimum that is defined by not only manufacturing complexities but also the fact that for very short blade the endwall regions with relatively high secondary losses will occupy almost the whole channel.

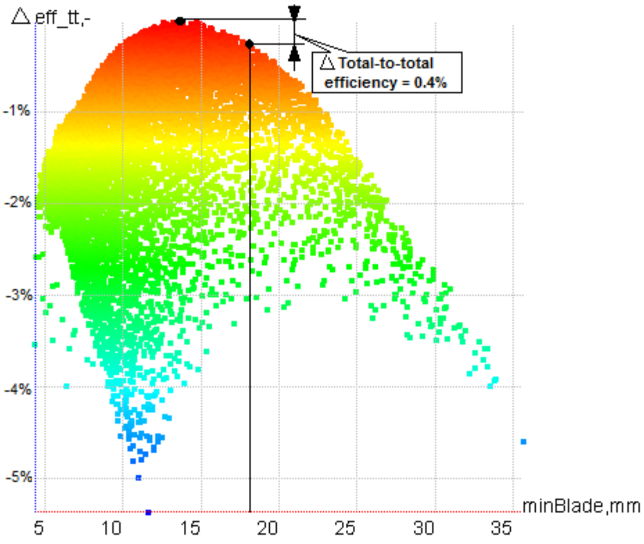

Fig.3. Total-to-static efficiency vs. blade height

For the current design, the optimal nozzle height is around 14 mm, which is far from the original design (17.75mm).

Gauging angle influence

The gauging angle defines how greatly the flow is swirled in the tangential direction. Thus, if it is too small of a gauging angle, the centrifugal force that appears due to its rotation (radial equilibrium) can force the flow to separate at some point.

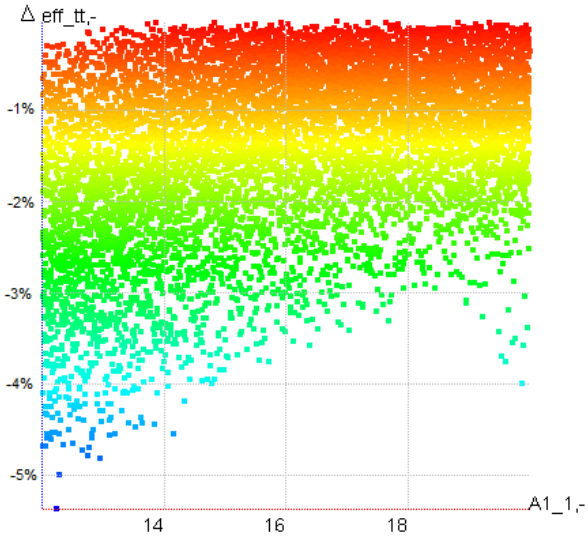

Fig.4. Total-to-static efficiency vs. control stage nozzle gauging angle

From the picture below, it’s obvious that the decrease of the gauging angle for the current case from 20 deg to 14 deg almost has no influence on the efficiency. Now, efficiency starts to decrease due to the reasons mentioned before.

Partial admission influence

Partial admission can cope with a small flow rate, though it decreases the performance because of additional losses present. In particular

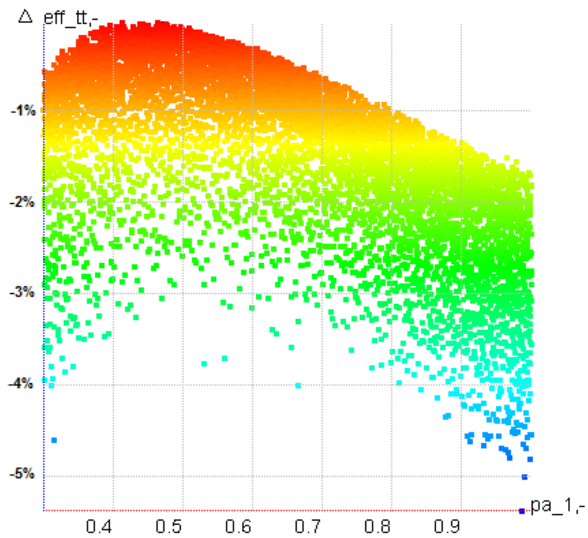

Fig.5. Total-to-static efficiency vs. partial admission

The partial admission for the particular combination of boundary conditions has an optimum at around 0.4-0.5, which is fairly close to the original variant and does not need to be corrected.

Recommended upgrade:

Blade height change from current 17.75 mm to 14 mm, which gives two stages efficiency improvement Delta Eff. = 0.4%, which resulted into 7 kW power gain.

VERIFIYING CALCULATION

Among many different types of losses in turbine stages, the leakages provide the most significant effect on performance, especially in small turbines with short blades. The traditional approach for leakage losses calculation [5, 6] does not take into account the effect of complex 3D flow structure in the partial admission stages, which leads to leakage losses over estimation with meanline and streamline design algorithms. This was observed several times by authors in their design experience by comparing calculations and test data for partial admission stages. Tip leakages loss model verification and correction was performed based on the two stages compartment CFD study.

<u>3D CFD model</u>

The 3D CFD model represents a detailed simulation of the two stages compartments with a mixing chamber between them in a "goose neck-like" shape, disk cavities and sealing above blades, shroud and under the diaphragms.

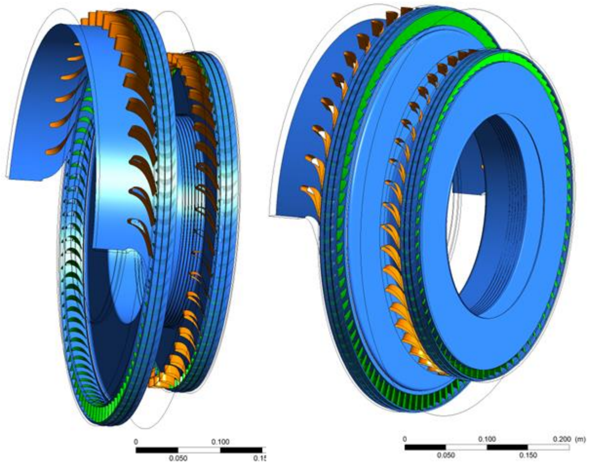

Fig.6. Two stages CFD model views with 50% admission first stage.

Total CFD mesh size has over 20 million elements with high quality nozzles and blades passages meshing (over 50k elements per passage) with boundary layers. Examples of meshes for blade passage, tip, shroud and diaphragm seals are presented at Figures 7 and 8.

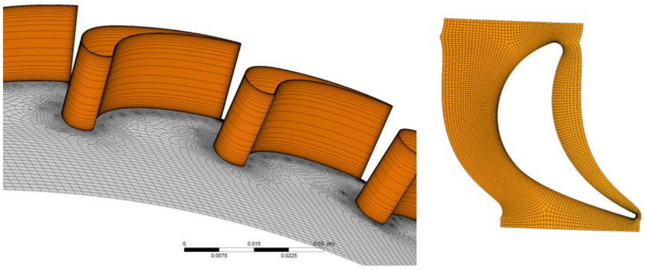

Fig.7. Mesh examples for nozzle and blade passages

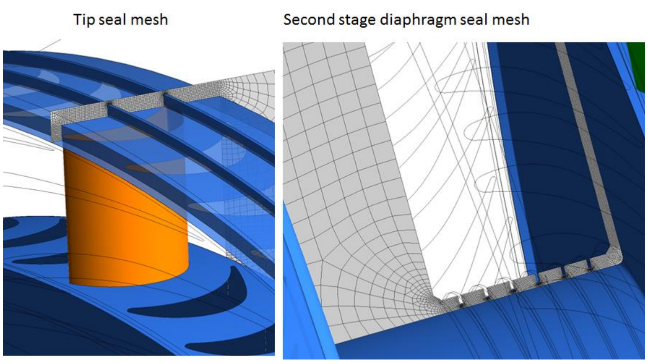

Fig.8. Mesh examples for tip shroud and diaphragm seals

<u>2D tip leakage model correction</u>

In the 2D approach, the value of leakages is calculated with:

$$\Delta G = \mu F \sqrt{\frac{P_0}{v_0}\sqrt{\frac{1-\varepsilon^2}{z}}} \quad (3)$$

Table 2 below represents the leakages comparison for CFD data and the 2D aero design algorithm for the uncorrected tip leakages formula.

Parameter	CFD	2D	Δ, %
1st rotor tip leakage, %	5.1	7.8	2.7
2nd nozzle gland leakage, %	4.8	6.1	1.3
2nd rotor tip leakage, %	4.0	4.1	0.1

Some discrepancies were observed between the values of relative leakages (leakage/Inlet MFR) calculated with 2D and CFD approaches. The biggest difference was for the first stage rotor tip leakage. This was caused by the fact that there is no pressure difference at the inactive part of the arc; consequently there are no significant leakages. At some locations, the flow even goes in the reverse direction. But the leakage calculation approach used in 2D supposes that the leakage takes place over the whole circumference, thus it overestimates the rotor tip leakages and the error is the more the less is partial admission. Another aspect attributed to leakages calculation is related to pressure reduction at the arcs ends, which also leads to leakages overestimation.

Both of these factors were considered with the correction coefficient $k=f(e, N_{segm)},$ which was determined as a correlation function of two parameters – partial admission ratio and number of segments.

$$\Delta G = k \mu F \sqrt{\frac{P_0}{v_0}\sqrt{\frac{1-\varepsilon^2}{z}}} \quad (4)$$

Such corrections resulted in a leakage value decrease to 3.9%, ipso facto decreasing the discrepancy between CFD and 2D calculations from 2.7% to -1.2% for first partial stage.

EFFECT OF ADMISSION ARCS POSITIONING ON DOWN STREAM STAGE PERFORMANCE

There are two main aspects about partial admission stages related to turbine aerodynamics, which need to be accounted for during turbine design.

Firstly, in the partial admission stages, additional aerodynamic losses are generated - windage and sector end losses. These losses can be easily accounted for in the design process by using the above mentioned equations (1) and (2). Secondly, in multistage turbines the first partial admission stage generates substantial circumferential non-uniformity and downstream stages experience circumferential pressure gradients and flow angle variations that produce additional losses in the downstream located stages. There is no proposed simple approach/method to evaluate this type of loss so far in the open sources, besides experimental study and 3D CFD analysis.

The effect of circumferential flow non uniformity and pressure gradients were experimentally investigated by Lewis [9] in a 4 stages test turbine – see picture below.

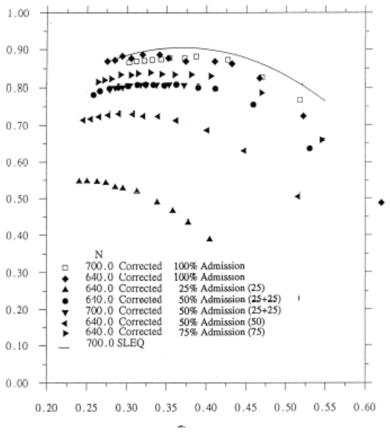

Fig.9. 4 stages test turbine efficiency at partial admission, Lewis (1993)

According to Lewis, the results of the circumferential flow deformation was rapidly weakening downstream-wise with the two front stages operating significantly off-design and the two rear stages were less affected. Lewis concludes that the effect of partial admission on the downstream stages can be significant and much of it depends on partiality and the mixing capacity of chambers between the first and downstream stages.

Another conclusion was that the use of several admission arcs is a preferable option, comparing to a single arc of admission for a multistage turbine. The same conclusion was also confirmed in some other studies for partial admission stages. The efficiency improvement of the overall performance for multistage turbines by using an increased number of admission arcs at a given admission rate was observed in CFD analyses by He, L., [10] and Hushmandi, B.N. [11] and confirmed in a test by Jens Fridh [12]. Multiple admission arcs with uniform circumferential location enhance the tangential mixing further downstream and thereby facilitate an improved performance of downstream stages, although the performance of the first partial stage evidently decreases. The results from Jens Fridh's [12] Doctoral Thesis are shown at Figure 7 for experimental turbine set up and second stage penalty loss, where the efficiency decrease is less from one to two admission arcs, for the two-stage turbine at 52.4% admission, than for the single stage turbine.

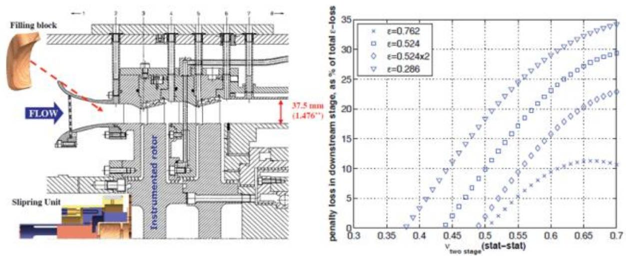

Fig.10. Test turbine set up and second stage penalty loss under different partial conditions, Jens Fridh, 2012

The effect in 6-7% for the second stage efficiency improvement was observed, which resulted from two admission arcs instead of a single one [12].

To evaluate the effect of the multiple arcs option for the turbine modernization project, the 3D CFD analysis was done based on the above mentioned CFD model.

The variants studied included admission sectors varying from one to five uniformly distributed in circumferential direction -Figure 8.

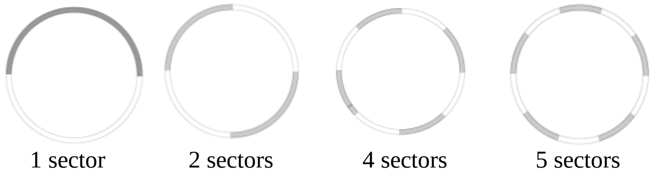

Fig.11. Variants of admission sectors distribution (total 50% partiality).

3D flow circumferential- structure

The CFD results, presented below at Figure 12 shows the 3D flow structure (streamlines) in the mixing chamber for a single sector variant. The main flow, which leaves the first row blades at an active sector, goes into the second stage nozzles. This flow is schematically shown with black arrows. Additional flow is developed in the mixing chamber at the inactive sector due to a combined effect of blade pumping and disk friction that moves in circumferential direction and is schematically shown with red arrows. At the active sector windward end, these two flows are met with strong vortices formation, which propagates circumferentially, interacting with the main and additional flows. Such a complex flow structure resulted into significant parameters non-uniformity in a tangential direction in the mixing chamber, which is demonstrated at Figure 13. Two variants of different admission sectors distribution are presented – one and five sectors. As it was mentioned, [12] the multiple sectors variant shows more uniform parameters distribution. As an example, pressure field is shown in the mixing chamber for a single sector and five sectors variant, which clearly demonstrates a positive effect of multi sectors arrangement.

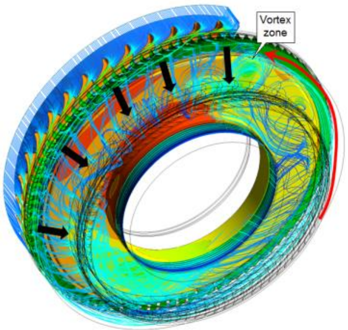

Fig.12. 3D flow structure in mixing chamber with single sector

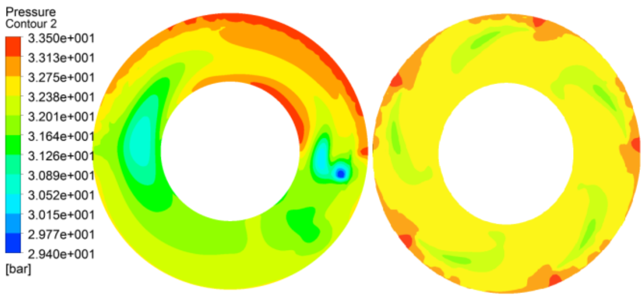

Fig.13. Pressure distribution at first stage outlet for single sector and five sectors variants

Effect of admission arcs number on second stage performance

More uniform flow parameters in the mixing chamber provide better inlet conditions for the second stage nozzles and improve the second stage blade efficiency. Figure 14 shows the total pressure distribution at the second stage inlet for a single sector and five sectors arrangement. The multi-sector variant shows a more uniform flow field, which results in higher second stage performance.

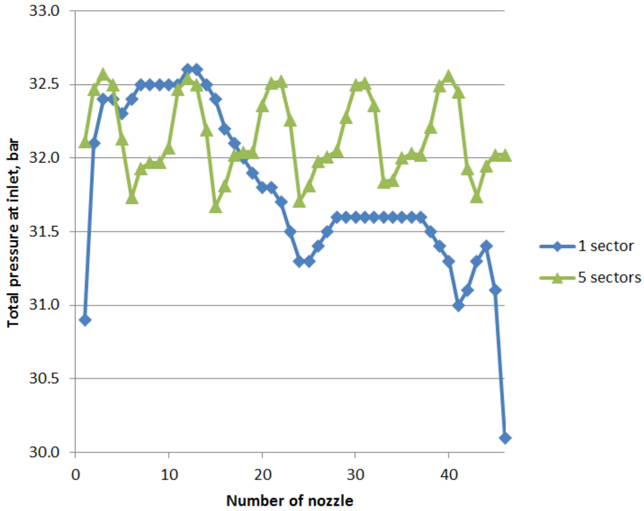

Fig.14. Total Pressure distribution at second stage inlet for single sector and five sectors variants

Figure 15 demonstrates a multi-sector variant advantage compared to a single sector by producing more power, which could be observed from the power per channel distribution chart below.

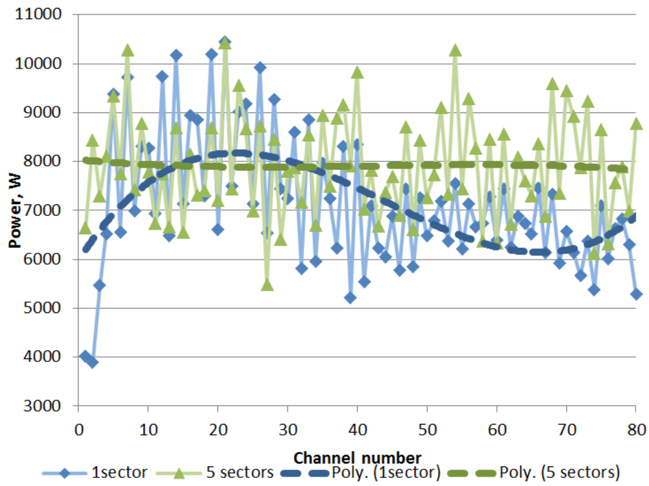

Fig.15. Power per channel distribution for one and five sectors variants

The two stages compartment performance variation with different numbers of segments was determined for all variants in comparison with a single sector case. The first stage performance was determined by a 2D algorithm and the second stage performance was determined based on 3D CFD results. The summary table is shown below.

In spite of the first stage performance drop, the second stage demonstrated better performance for the multiple sector arrangement. In total, the two stages power increased by 43.4 kW for the 5 sectors variant.

Table. 3 Performance summary for different number of segments

Number of segments	1	2	4	5
ΔN stg1, kW	0	-5.14	-15.18	-20.24
Δηtt stg1, %-	0	-0.46	-1.39	-1.86
ΔN stg2, kW	0	20.1	50.4	63.6
Δηtt stg2, %	0	1.66	4.15	5.11
ΔNstg1 + ΔNstg2, kW	0	14.96	35.22	43.36

CONCLUSION

The calculation study was undertaken for a two stages compartment with a partial admission control stage for a 10 MW industrial turbine with the purpose to identify potential options for a future turbine upgrade.

For the first partial admission control stage the proposed upgrade option was to change the blade height from its original 17.75 mm to a new 14 mm, which resulted in an additional 7 kW power for a two stages compartment.

For the second stage, the multiple admission sectors arrangement was studied with 3D CFD. A five sectors variant resulted in 43.36 kW of additional power.

The final decision of both considered options should be done based on a cost versus performance trade-off analysis.

Additional results included the 2D design algorithm extension with regards to more accurate tip leakage calculation for partial admission stages.

A detailed 3D CFD model developed for a turbine upgrade project and analysis of CFD results offer more possibilities to investigate such a complex flow behaviour for partial admission stages. It was done and presented in the course of this work. Authors will continue to study and reveal the numerous aspects of the partial admission turbines related to aerodynamics and blades forcing analysis in their future publications.

BIBLIOGRAPHIC REFERENCES

[1] H. R. M. Craig, H. J. A. Cox, “Performance Estimation of Axial Flow Turbines”. The Sust. Of Mech. Eng/ Proc, 1970/71, vol. 185, 32/71, P. 407-424

[2] Stepanov G.U. Turbomachinery Cascades Hydrodynamics, 1962

[3] Leonid Moroz, Yuri Govorushchenko, Petr Pagur, “Axial Turbine Stages Design. 1D/2D/3D Simulation, Experiment, Optimization”, ASME Turbo Expo 2005: Power for Land, Sea and Air, GT2005-68614.

[4] Moroz L., Govorushchenko Y., Pagur P., 2006, A Uniform Approach to Conceptual Design of Axial Turbine/Compressor Flow Path, Future of Gas Turbine Technology, 3rd International Conference, Brussels, Belgium

[5] Stodola, A., Louis C. Loewenstein Trans., 1927, “Steam and Gas Turbines Vol. I”, McGraw-Hill Book Co., Inc., Reprinted by Peter Smith, 1945

[6] Traupel, W., 1966, “Thermische Turbomaschinen”, Vol. 1, 2nd Ed., Springer-Verlag, Berlin, in German

[7] Traupel, W., 1977, “Thermische Turbomaschinen”, Vol. 1, 3rd Ed., ISBN 3-540-07939-4, in German

[8] Roelke, R.J., 1973, “Turbine Design and Application”, NASA SP-290, A.J. Glassman, Vol.2, Chap. 8

[9] Lewis, K.L., 1993, “The influence of partial admission of a multistage turbine”, Internal report, Whittle Laboratory, Cambridge University

[10] He, L., 1997, “Computation of unsteady flow through steam turbine blade rows at partial admission”, Proc. of ImechE Journal of Power and Energy, 211, pp. 297-305

[11] Hushmandi, B.N., 2010, “Numerical Analysis of Partial Admission in Axial Turbines”, PhD thesis, KTH, Stockholm, Sweden, ISBN 978-91-7415-390-3, ISSN 1100-7990, ISRN KTH-KRV-R-10-02-SE

[12] Jens Fridh, Experimental Investigation of Performance, Flow Interactions and Rotor Forcing in Axial Partial Admission Turbines, Doctoral Thesis 2012, Division of Heat and Power Technology, Department of Energy Technology Royal Institute of Technology, Stockholm, Sweden