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Thermodynamic Considerations for Large Steam Turbine Upgrades and Retrofits

Leonid Moroz, Kirill Grebennik
15 New England Executive Park, Burlington, MA, 01803
SoftInWay Inc, USA

Abstract

Upgrading and retrofitting of power plants have long been a focus of utilities, industrial companies and independent power producers. Normally, it was possible to obtain OEM "like new" performance levels, or purchase OEM upgrade products. New approaches to restoring or increasing performance of existing in-service turbines are presented here. The ability to analyze aerodynamic and thermodynamic parameters of existing machines and make adjustments to original airfoil shapes is ensured by new developments in turbomachinery flow path design, analysis and optimization software AxSTREAM.

Redesign of large steam turbines for power generation with capacity range from 100 to 1000 MW is presented in the current article.

A few main upgrade and retrofitting tasks could be distinguished, such as replacement of blading to increase turbine performances, turbine re-rate to increase capacity or retrofitting to obtain highest performances with changed inlet or outlet conditions. For each kind of these tasks it's possible to define upgrade limitations like keeping existing meridional dimensions for all or for group of stages, maintaining existing axial dimensions or retaining existing casing and rotor. Optimal selection of seals and clearances for the replacement is also an important task because these parameters are usually limited by manufacturing and operating criteria. Also, it's necessary to clearly predict the effectiveness of investment into selected kind of upgrade and choose it with the respect to expected financial return volumes and terms which we want to receive. For power generation turbines, depending on turbine capacity, application and working time, different range of increase in percentage of efficiency after retrofitting could be acceptable.

A retrofitting procedure could be divided into the following main phases: data retrieval and original flow path model recovery, analysis of upgrade possibilities and performing changes within existing limitations, comparing and estimating of such upgrade economic effect and reasonability. The software used here allows engineers to cover all steps of this cycle having full control on all flow path geometry estimating integral and detailed thermodynamic and kinematic performances on each step for design and off-design modes. This can significantly decrease time and increase final efficiency of upgrading procedures and allows completing retrofitting procedures independently from OEM that produced the current turbine.

In current paper the retrofitting procedure and the results of a redesign with new optimized performances for 160 MW steam turbine are presented.

Nomenclature

DoE – design of experiment

HPC – high pressure cylinder

IPC – intermediate pressure cylinder

LPC – low pressure cylinder

1. Introduction

Nowadays, fossil-fired power plants are one of the world's major sources of carbon dioxide emission that produces significant part of pollutions. According to latest available data [1], [2], fossil fuels are used to generate about 67% of electricity in the world. The most of fossil-fired power plants are equipped with steam turbines with power ratings ranging from 100 to 1000 MW. In recent years, steam turbines generate about 80% of world's electricity. Increasing energy demand requires steam turbines OEM's and service companies to invest more in design and upgrade process to provide utilities with high-efficient turbines, that help to save fuel, produce higher output and reduce emissions.

Power plants capacities installed in late 1970-early 80s have reached their expected life span and operating them requires major maintenances. At this point, their design can be changed significantly to satisfy modern thermodynamic cycle conditions and requirements, using latest technological "know-hows" available. One of the most common tasks is conversion of steam turbines for usage with HRSG (heat-recovery steam generator) in combined cycle, increase of inlet temperatures, seals replacement, cycle upgrades with change of extraction and induction parameters and conditions.

Current article represents redesign process of 160 MW steam turbine for changed cycle conditions. All phases and challenges starting from data collection and original turbine model creation, feasibility and manufacturability study of proposed upgrades variations, final machine redesign study and evaluation.

2. Original Turbine Model Description

Original turbine specifications are next:

- Rated power = 160 MW
- Inlet pressure before turbine = 124.5 bar
- Inlet temperature before turbine = 536 C
- Mass flow rate at inlet = 139 kg/s
- Rotation speed = 3600 rpm
- Pressure in condenser = 0.0556 bar
- Reheat temperature = 536 C
- 3 cylinders with next configuration: HP: Double-row control wheel+7 stages, IP – 7 stages, LP – 6 stages
- HP and IP cylinder are in one casing, LP – double-sided
- Regeneration extractions: 3 (3 kg/s, 3.96 kg/s, 3.6 kg/s) in IPC, 2 (3.85 kg/s) in LPC

In present article, redesign and upgrade of HP and IP sections is described, therefore only those cylinders model will be created and used for turbine analysis.

2.1 Data Collection and Retrieval

A few main steps of data collection and retrieval procedure can be distinguished, such as:

1. Thermodynamic cycle scheme information collection and analysis – details about inlet and outlet conditions, extraction positions and flow rates
2. Turbine geometry dimensions (hub and tip diameters, heights, etc)
3. Clearances and seals actual conditions

4. Blades 3D model scanning (or profile XY coordinate measuring on sections spanwise)
5. Profiles geometry properties (throat, chord, LE and TE radii etc) recognition

There is a minimal set of geometrical data, required for machine aerodynamic analysis. Most important of them, such as diameters, heights, number of blades, have to be measured as accurate as possible. Correct clearances and seals condition measurement is crucial in case of redesign and upgrade task, because seal type and conditions improvement may bring significant impact on resulting performances.

Currently turbines reverse engineering and redesign process have some limitation due to impossibility of complete and correct profile data recovery. High accuracy of data obtained is also critical, because even small error in cascade data may significantly affect resulting performances. Error in throat distance measurement is in direct proportion to mass flow rate, and therefore, power. Depending on particular stage configuration 1 degree error in gauging angle determination can result in mass flow difference about 3-5%. Main problem is a lack of technologies that can process digitized data and extract necessary information for aerodynamic analysis with reliable quality within short time. The special technology of 3D blade models recognition was utilized in this article. It helps to eliminate existing limitations and process scanned data in special way to accurately extract required properties (angles, radii, etc) from it. The technology essence will be described.

When collection of data is done (by laser scanning or any other technology), blades 3D model recognition and importing procedure was performed to extract required profile geometry data in numerical format. Scanned points coordinates obtained by laser scanning were taken to special profile processing software to extract necessary geometry data from 3D model on selected spanwise sections. The example of profile obtained with this procedure is below.

Figure 1 shows 3D blade model with hub and shroud lines defined and sliced on 5 spanwise sections. Profile coordinates are extracted on selected sections and profile shapes with all theirs properties could be recognized basing on the points sequence extracted, see Figure 2.

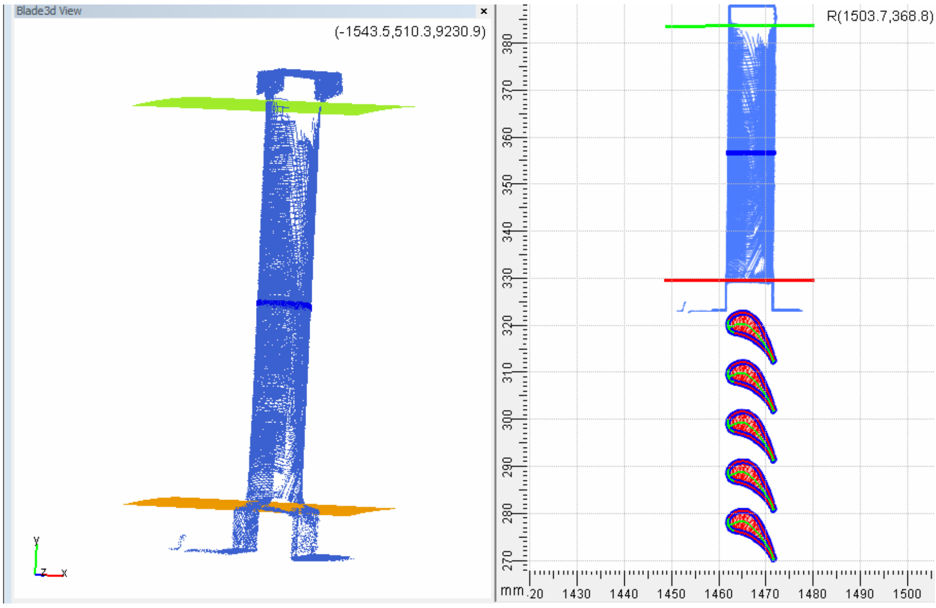

Figure 1. Blade 3D model sliced on sections spanwise

On the next step profiles coordinates recognized are ready to import in aerodynamic performances analysis software.

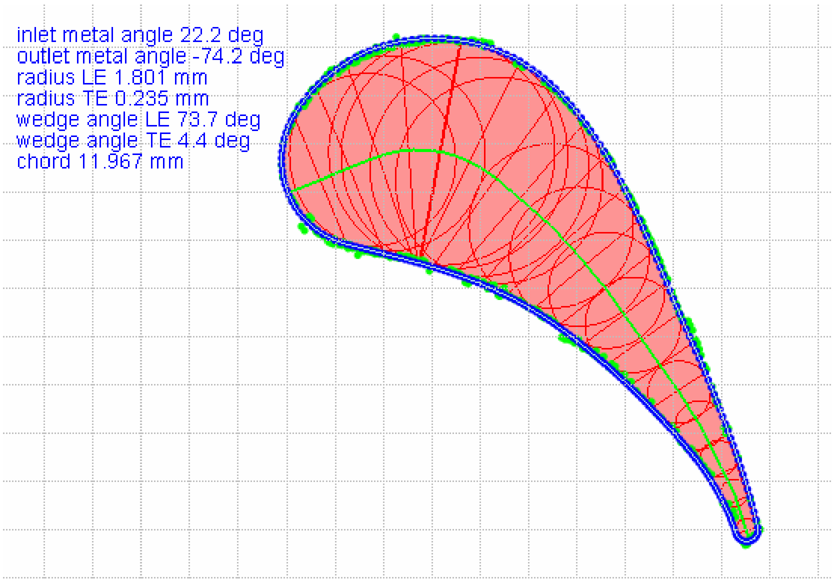

Figure 2. Recognized profile shapes and properties

When recognition is done for all stages and other data is prepared, machine model is ready for analysis.

2.2 Turbine Model Benchmark

The major step to be done before performing redesign is baseline model creation and benchmarking. Turbine aerodynamic performances evaluation for existing machine in its current conditions is necessary to check correspondence between created model results and real output data, taking into account current flow path conditions deteriorated from original ones. When this analysis is done, compared and evaluated, reference flow path conditions such as power, thermodynamic and kinematics in elements will be known for further references.

According to redesign plans HP and IP cylinders with steam reheat between them were simulated in aerodynamic calculation. Data acquisition and model creation process for them was presented above.

Figure 3 represents original turbine HPC and IPC aerodynamic conditions layout including current extractions in IP sections, as well as place for extraction, which is supposed to be added during machine redesign.

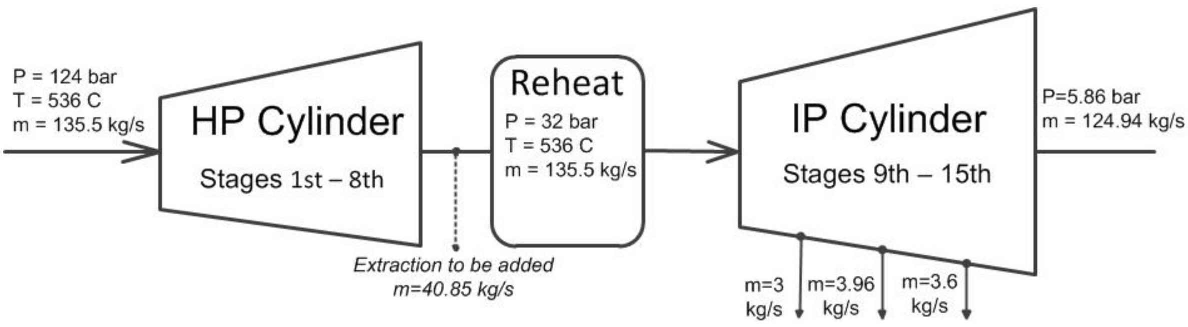

Figure 3. Turbine HP and IP cylinder scheme

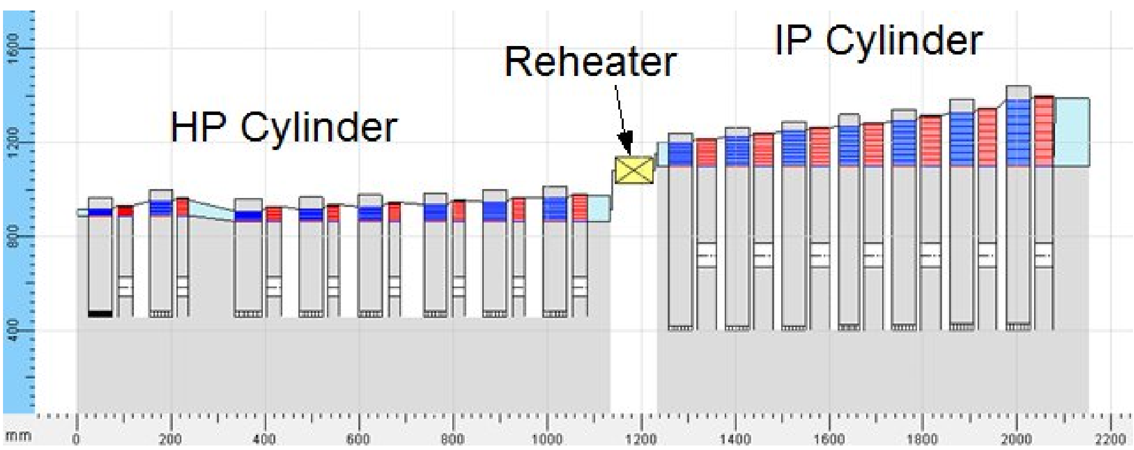

Figure 4. Turbine cross-section model

Table 1. Original turbine model benchmark results

Total pressure at inlet	bar	124.54
Total enthalpy at inlet	kJ/kg	3444.14
Total temperature at inlet	°C	536.5
Static pressure at HP outlet	bar	32.33
Static pressure at IP outlet	bar	5.86
Mass flow rate at HP inlet	kg/s	135.54
Mass flow rate at IP inlet	kg/s	135.54
Mass flow rate at IP outlet	kg/s	123.53
Shaft rotational speed	rpm	3600
Averaged isentropic velocity ratio	-	0.620
Volume flow rate at outlet	m^3/s	56.86
HP cylinder power	MW	40.06
IP cylinder power	MW	59.16
Total power (HP+IP cylinders)	MW	**99.22**

Reheating after HP section increases steam temperature to 536 C, a pressure loss of about 1% in reheater pipings were assumed. The difference between flow rates at IP cylinder inlet and outlet is caused by extractions in IPC between stages. Because turbine will be redesigned to have 30% flow rate extraction (40.85 kg/s) after HPC, then additional calculation with this extraction added has to be done for original model in order to have baseline for its comparison with upgraded machine. When initial analysis is done, this result can be used for further comparison.

Table 2. Original turbine model with extraction added benchmark results

Total pressure at inlet	bar	124.54
Total enthalpy at inlet	kJ/kg	3444.14
Total temperature at inlet	°C	536.5
Static pressure at HP outlet	bar	22.16
Static pressure at IP outlet	bar	5.861
Mass flow rate at HP inlet	kg/s	138.69
Mass flow rate at IP inlet	kg/s	97.53
Mass flow rate at IP outlet	kg/s	85.94
Shaft rotational speed	rpm	3600
Averaged isentropic velocity ratio	-	0.620
Volume flow rate at outlet	m^3/s	41.87
HP cylinder power	MW	51.59
IP cylinder power	MW	32.93
Total power (HP+IP cylinders)	MW	**84.52**

When extraction after HP was added, pressure at HP outlet dropped - from 32.33 to 22.16 bar approximately, as it's evident from calculation results. Pressure and heat drop were redistributed between cylinders: HPC got higher pressure drop, than in case with no extraction. Therefore, power produced by HPC in case with extraction will be increased from 40.06 to 51.59 MW, and IP cylinder power drops correspondingly from 59.16 to 32.93 MW, as pressure drop across IP cylinder decreases.

The turbine redesign process will be started from IP cylinder, taking pressure after HP cylinder, obtained with 30% extraction included, as initial approximation.

3 Turbine IP Cylinder Redesign

3.1 IP Cylinder Redesign Task Formulation

The aim of first part of machine redesign is to adjust and optimize IP cylinder for changed extraction conditions before IP section. Cycle modification project requires having extraction of 30% flow rate after HP section before reheater. Thus, current IP section has to be redesigned accordingly to new mass flow rate conditions before it. Limitations set for IP section redesign are next:

1. Preserve similar constant hub diameters for blades to use current disks for new blading
2. Not to exceed existing casing radial and axial dimensions.
3. Minimal number of blades

3.2 IP Cylinder Redesign Study

This section is dedicated to IP cylinder redesign study. As it was discussed above, the only one major limitation in current case is constant hub diameter. The rest of geometry parameters including elements angles and heights aren't strictly limited. For such case, complete redesign of IP cylinder for new conditions was done by meaning of design from scratch procedure with next initial parameters:

1. Inlet total pressure to IP cylinder = 22.4 bar (obtained as result of thermodynamic calculation on original model)
2. Inlet total temperature = 536 deg C

3. Static pressure at IP cylinder outlet = 5.86 bar
4. Mass flow rate at IP cylinder inlet = 97.53 kg/s
5. IP cylinder hub diameter is constant = 1100 mm
6. Maximal tip diameter = 1390 mm
7. Number of stages = 7

Preliminary design procedure allows creating new flow path according to boundary conditions and constraints applied. Detailed description and theoretical background of turbine preliminary design procedure is presented in [3].

In current case criterion for preliminary design procedure was maximum power produced by this cylinder. Basing on data above, new design candidates were generated and compared to obtain optimal design that satisfies all geometry constraints, have minimal exit energy losses, i.e. optimal outlet flow angle from last stage. When design candidate was selected, it could be adjusted to satisfy actually conditions in terms of seals, clearances, balance holes and stages axial and radial dimensions if necessary. IPC thermodynamic calculation has to be performed for final turbine performances evaluation.

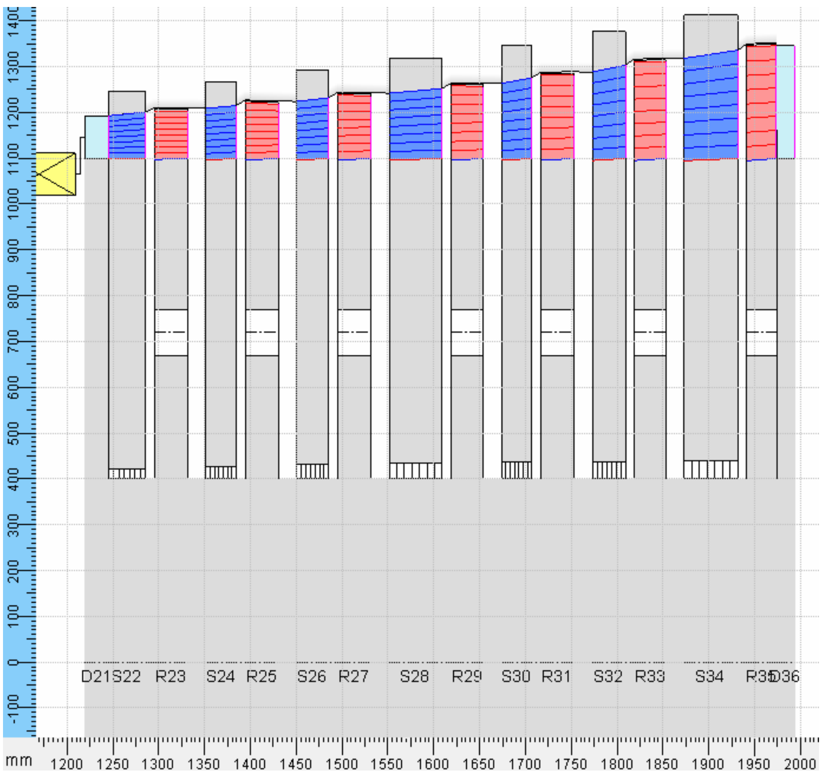

Figure 5. Redesigned IP cylinder cross-section view

Table 3. Redesigned IP cylinder performances

Total pressure at inlet	bar	124.54
Total enthalpy at inlet	kJ/kg	3444.14
Total temperature at inlet	°C	536.5
Static pressure at IP outlet	bar	5.861
Mass flow rate at inlet	kg/s	140.79
Mass flow rate at IP inlet	kg/s	99.93
Mass flow rate at IP outlet	kg/s	88.21
Shaft rotational speed	rpm	3600
Averaged isentropic velocity ratio	-	0.617
Volume flow rate at outlet	m^3/s	42.04
HP cylinder power	MW	53.55
IP cylinder power	MW	37.50
Total power	MW	**91.05**

Such improvements in cylinder performances were achieved by next flow path adjustments:

1. Modification of IP cylinder stages heights and throats in order to handle changed flow better than in original one
2. Inlet blade angles adjustments to minimize losses from non-optimal incidence angle
3. Clearances and seals improvement to decrease leakage losses
4. Redistribution of heat drops between stages to equalize their specific work (see Figure 9)
5. Last stage flow discharge angle adjustment to decrease exit energy losses

To compare original and redesigned IPCs', power and efficiency of stages before and after redesign can be considered. Note: stages numbering starts from HP inlet, i.e. first stage in IP section is 9, last - 15.

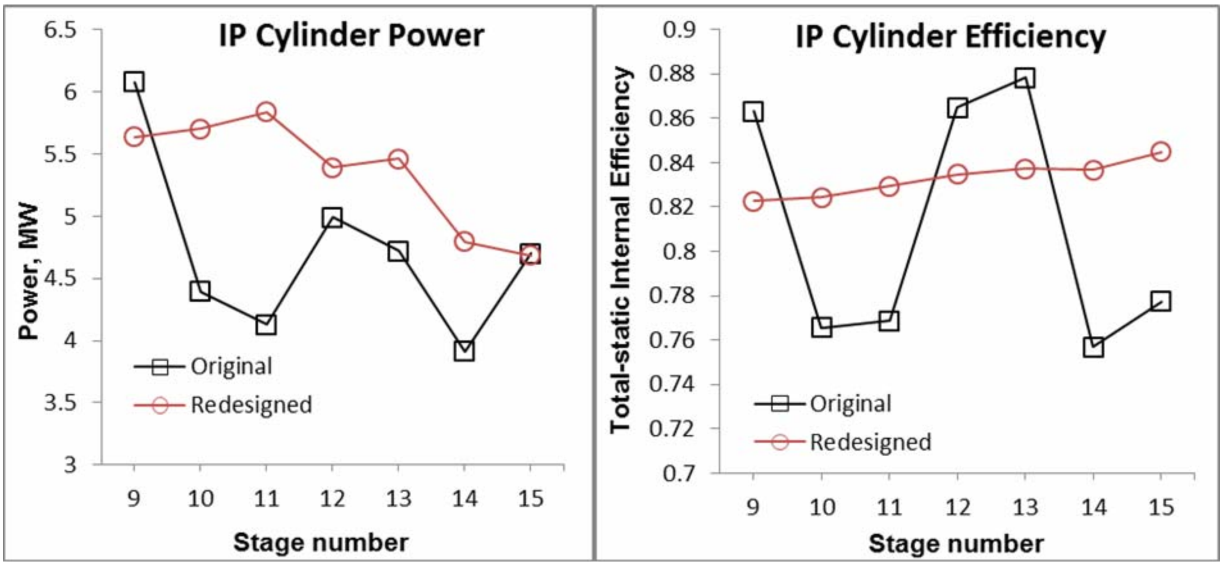

Figure 6. Power and efficiency comparison for original and redesigned IP section

Since hub diameter remained constant (1100 mm), the major modifications between original and redesigned IP cylinder were made to blade heights and gauging angles (Gauging angle = ASIN (throat/pitch)).

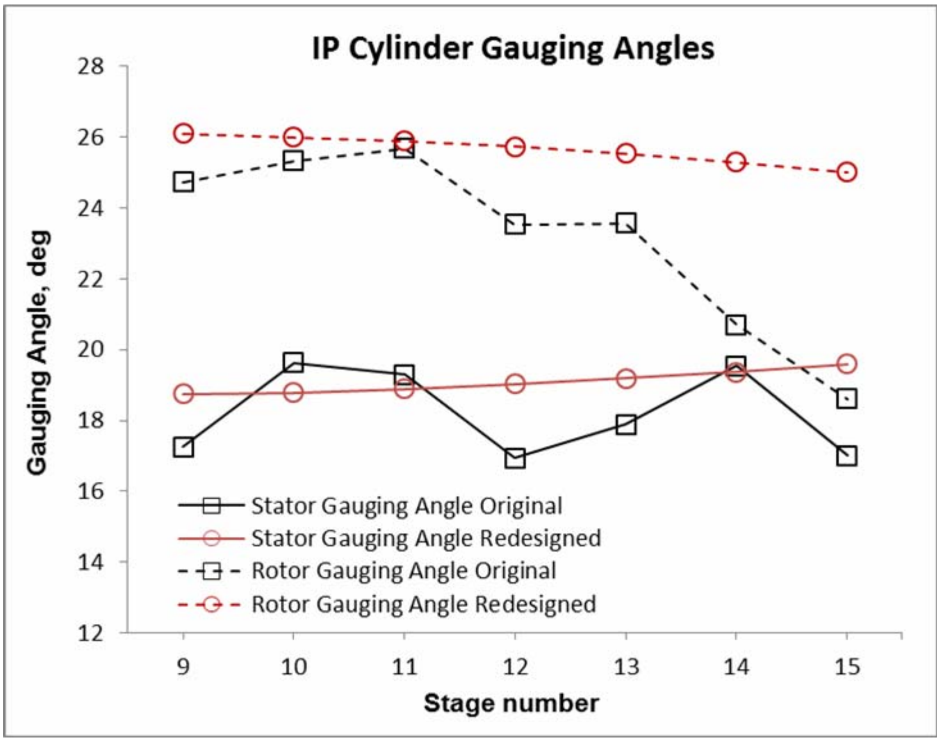

Figure 7. Original and redesign IPC's gauging angles comparison

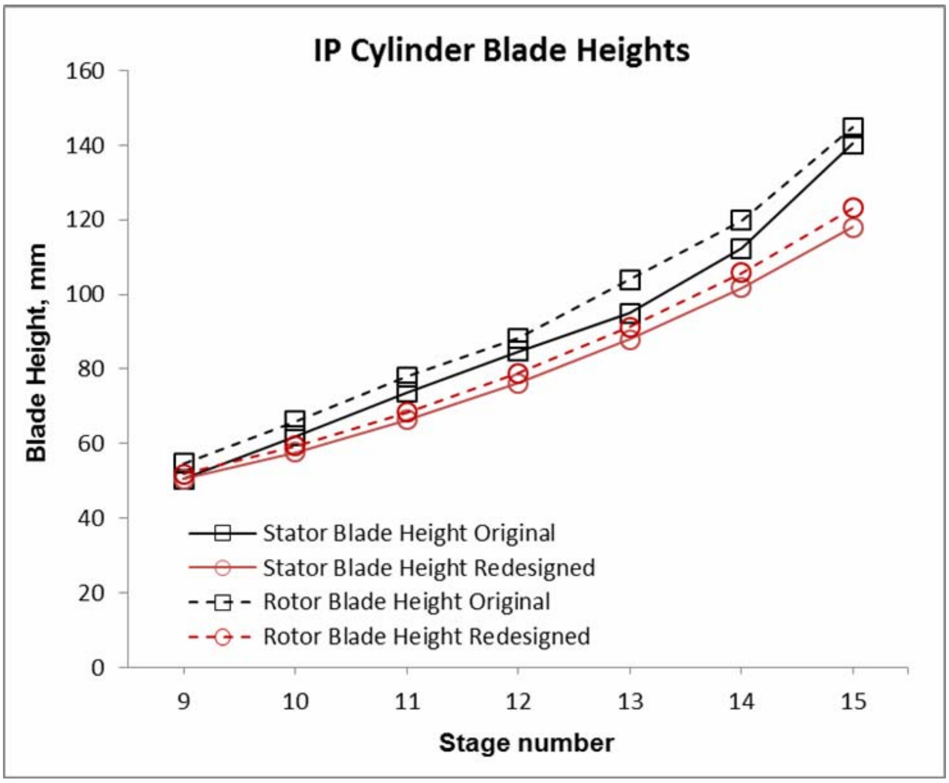

Figure 8. Original and redesign IPC's blade heights

Table 4. Original and redesign IP cylinders geometry parameters comparison

	Blade height at TE, mm		Gauging angle, deg	
	Original	Redesigned	Original	Redesigned
Stator Stage 9	50.55	50.5	17.26	18.76
Rotor Stage 9	54.61	51.87	24.74	26.10
Stator Stage 10	61.98	57.76	19.62	18.78
Rotor Stage 10	66.04	59.37	25.34	26.01
Stator Stage 11	73.66	66.29	19.30	18.90
Rotor Stage 11	77.98	68.32	25.69	25.89
Stator Stage 12	84.71	76.28	16.94	19.03
Rotor Stage 12	88.27	78.83	23.55	25.74
Stator Stage 13	95.00	88.00	17.89	19.19
Rotor Stage 13	103.89	91.20	23.58	25.55
Stator Stage 14	112.27	101.80	19.54	19.37
Rotor Stage 14	119.76	105.82	20.70	25.31
Stator Stage 15	140.46	118.09	17.01	19.60
Rotor Stage 15	144.78	123.16	18.61	25.01

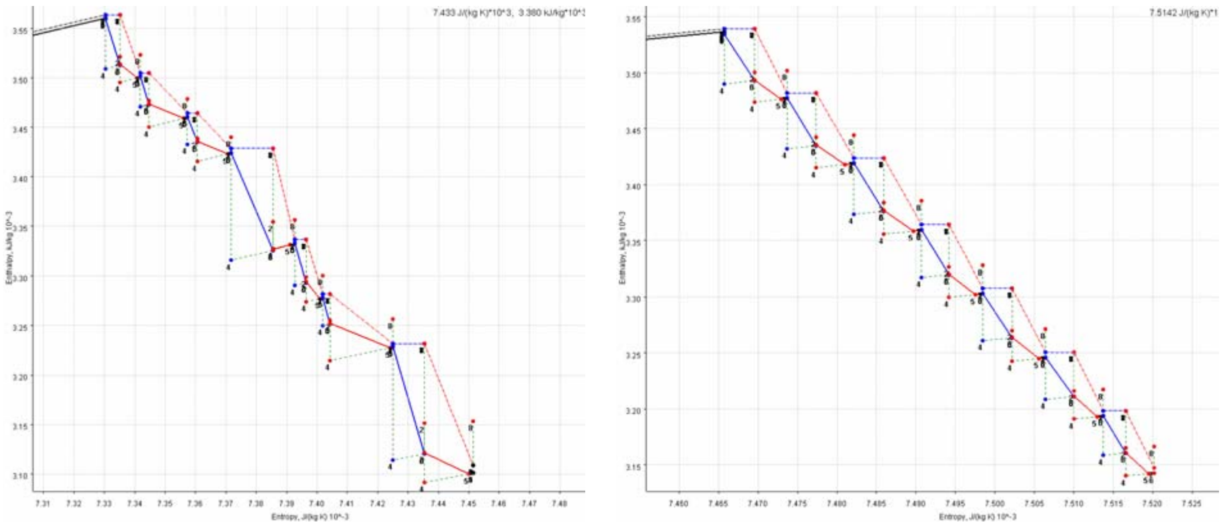

Figure 9. Mollier (enthalpy-entropy) diagram for original (left) and redesigned (right) IP cylinder

Redesigned IPC was created to better handle changed conditions: pressure and mass flow rate. Complete redesign of stages heights and angles, seals improvement, resulted in increasing IP cylinder power by 6.54 MW compared to original design.

4 Turbine HP Cylinder Upgrades Estimation

4.1 HP Cylinder Redesign Task Formulation

The change of HP cylinder parameters after cycle modification is shown above. Pressure after HPC dropped from 32.13 bar down to 22.19 bar after extraction was added. Correspondingly power, produced by HP cylinder, increased from 40 to 51.6 MW. When redesign of IP cylinder was done, calculation showed increase of power in HP cylinder up to 53.55 MW. To improve HP cylinder performances with changed outlet pressure, redesign possibilities study can be performed.

The main requirements for HP cylinder modification are:

- HPC blades and nozzles diameters and heights have to be maintained in existing dimensions
- Mass flow rate 140 kg/s has to be kept constant

Due to requirements set, designer is limited to modify only profile shapes. The main issue for HPC redesign is that any action may have impact on actual IPC's conditions at stage, when its redesign has already been done. Therefore, all adjustments and modification of HPC should be performed and evaluated only together with IPC.

4.2 HPC Performances Improvement Study

When possible modifications constraints are defined, redesign strategy and approaches can be developed. To start work on HPC redesign next options was selected for study:

- Adjust inlet metal angles to current flow angles to minimize incidence angles losses
- Optimize gauging angles on HPC stages by Design of Experiment study
- Clearances and seals renovation and improvement

Current incidence angles for HPC stages are presented in table below.

Table 5. Incidence angles in HPC, deg

	Stator	Rotor
Stage 1	0	-22.3
Stage 2	-17.4	13.8
Stage 3	28.5	-4.0
Stage 4	24.8	-15.9
Stage 5	7.6	-30.5
Stage 6	-13.8	-21.3
Stage 7	-4.1	-30.8
Stage 8	-17.4	-19.0

Basing on this data, it was decided to modify next elements: stators of stages 3 and 4, rotors of stages 5 – 7. Adjustment of inlet metal angles was done to have:

Incidence angle = Blade inlet metal angle – inlet flow angle = 0 deg

This made possible to decrease profile losses on selected rows and improve their efficiency. To compare modifications on these stages total pressure loss (pressure discharge) factor on stators and rotors was used as indication:

$$Y_S = \frac{P_0^* - P_1^*}{P_1^* - P_1}; \; Y_R = \frac{(P_1^*)_{REL} - (P_{2S}^*)_{REL}}{(P_{2S}^*)_{REL} - P_{2S}};$$

Table 6. Total pressure loss factor

	Original HPC	Redesigned HPC
Stator Stage 3	0.1458	0.1075
Stator Stage 4	0.1061	0.0847
Rotor Stage 5	0.2551	0.1133
Rotor Stage 6	0.2573	0.1035
Rotor Stage 7	0.2553	0.1149

On the next step, design of experiment was performed to estimate possibilities of HPC improvement by gauging angles adjustment. To decrease number of elements that is supposed to be completely redesigned, study was made only to stators of stages 3 and 4. The variables for this study are gauging angles on stators of 3rd and 4th stage, range of angles variation is from 14 to 22 deg. Objectives are next: power as main optimization criterion and mass flow rate, because it has to be the same after optimization for correct task accomplishment.

Design of Experiment was performed using Box and Behnken [4] Design of Experiment (DoE) approach. Response surface of objectives was generated basing on reference points values, selected and calculated accordingly to this DoE plan. Response surface calculated is presented in Figure 10 below. Gauging angles of 3rd and 4th stator corresponds to X and Y axes, power corresponds to Z axis.

Multi-parametric search on this response surface was performed using power as optimization criterion, and found that maximal power of 92.1 MW corresponds to angles shown in Table 8.

Table 7. Gauging angle comparison before and after DoE study, deg

	Initial	DoE
Stage 3 Stator	18.13	20.00
Stage 4 Stator	18.07	14.00

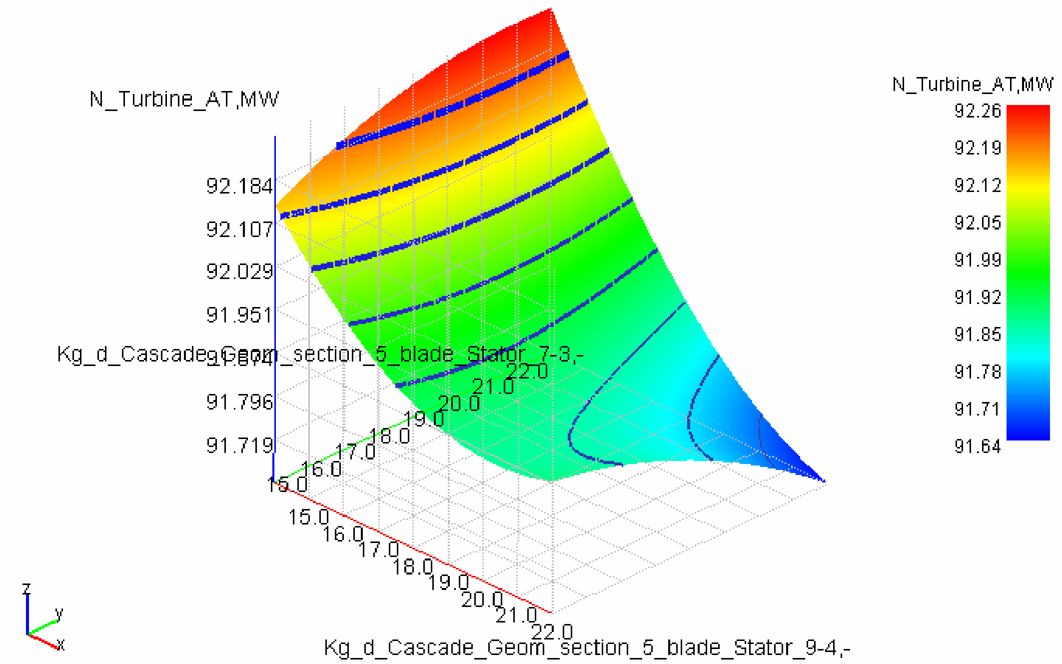

Figure 10. DoE response surface

Resulting performances of turbine obtained after HPC modifications are presented below.

Table 8. Turbine power after HP cylinder modification, MW

HP cylinder power	54.68
IP cylinder power	37.42
Total power	**92.1**

5 FINAL DESIGN EVALUATION

Final design evaluation was performed to confirm that all modifications made for HPC and IPC are providing their reliable operation in all range of working modes, as well as all possibilities of upgrade were used within existing limitations. Following modifications were made to HPC:

- Inlet blade metal angles adjustment on 3rd and 4th stages stators, 5th to 7th stages rotors
- 3rd and 4th stages stator gauging angle optimization with design of experiment methods

Extraction of 30% flow was added after HPC. Pressure distribution between cylinders was changed due to it, i.e. pressure drop on HPC increased while it reduced in IPC. IPC was modified to upgrade it for new inlet conditions after HPC with next changes made:

- Complete IPC redesign using preliminary design procedure
- Change of heights, inlet angles, throats and number of blades
- Clearances and seals improvement

Resulting redesigned flow path performances are presented in Table below.

Table 9. Resulting performances of HPC and IPC

Total pressure at inlet	bar	124.54
Total enthalpy at inlet	kJ/kg	3444.1
Total temperature at inlet	°C	536.5
Stat. pressure at outlet	bar	5.86
Mass flow rate at inlet	kg/s	140.58
Mass flow rate at IP inlet	kg/s	97.53
Mass flow rate at IP outlet	kg/s	88.21
Flow angle at inlet	deg	90
Shaft rotational speed	rpm	3600
Averaged isentropic velocity ratio	-	0.616
Volume flow rate at outlet	m^3/s	41.93
HP cylinder power	MW	54.68
IP cylinder power	MW	37.42
Total power	MW	**92.1**

6 CONCLUSION

This article presents all redesign steps of large steam turbine for changed steam conditions and 30% extraction added after HPC. To create turbine model 3D scanning with further blade profile characteristics recognition utilizing new data extraction approach was performed. Required machine data acquisition and organizing process was completed and turbine model ready for analysis and optimization. Evaluation of conditions on initial machine was done to have baseline for comparison. During redesign process complete modification of IP cylinder was performed by using preliminary design procedure. Number of stages, radial and axial dimensions was kept as initial constraints to use existing casing and disks. These improvements helped to increase power of IPC by 6.54 MW and got final power 92.1 MW for both HP and IP cylinders.

Analysis performed here shows that the various methodologies and approaches can be applicable in redesign process depending on task complexity, requirements and existing constraints, goals that have to be obtained. Redesigned turbine cylinder can be designed from scratch with further adjustments in case of complete cylinder modification. If designer is free to modify only a few particular elements with very strict constraint, then other approaches, such as Design of Experiment can be used to find optimal solution within limitations set.

Modern hardware (measuring equipment) and software technologies can significantly decrease redesign time and increase upgraded machine power and efficiency.

LITERATURE

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