Int. Symp. on Heat Transfer in Gas Turbine Systems
9-14 August 2009, Antalya, Turkey

COMPARISON OF COUNTER – ROTATING AND TRADITIONAL AXIAL AIRCRAFT LOW-PRESSURE TURBINES INTEGRAL AND DETAILED PERFORMANCES

Leonid Moroz, Petr Pagur, Yuri Govorushchenko, Kirill Grebennik
SoftInWay Inc. 35 Corporate Dr., Burlington, MA 01803, USA
(* Contact author: k.grebennik@softinway.com)

ABSTRACT. Raising requirements for aircraft engine efficiency and fuel consumption level combined with strong restrictions to engine weight and geometrical dimension pose serious challenges for engineers who are working under the new generation of engine development. These tasks require brand new flow path design approaches. The usage of a counter-rotating turbine is one of the possible ways to successfully match all these requirements. Modern aerodynamic design computational and optimization methodologies allow to fulfil this task in the shortest period of time with the highest gain in turbine performances.

A counter-rotating turbine means that blade rows are joined to two shafts with opposite rotation direction and different rotation speeds. Vanes elimination in a counter-rotating turbine helps to solve three important tasks of turbine improvement:

- Increasing turbine efficiency by eliminating vanes and correspondingly losses in vanes;
- Decreasing turbine blading weight;
- Decreasing turbine axial length;

These improvements are impossible without such fundamental design changes.

In the current paper the steps of counter-rotating turbine aerodynamic design, optimization, and off-design performances estimation are described. The comparison of traditional and counter rotating turbines integral and detailed thermodynamic performances are presented.

INTRODUCTION

Traditional axial turbine consists of stationary vanes (stators) and rotating blades rows consequently placed in the flow path. Vanes are guiding and accelerating the flow in the required direction and moving blades are converting kinetic energy of moving fluid into mechanical work on the shaft. An example of traditional axial turbine stage is presented in the figure below.

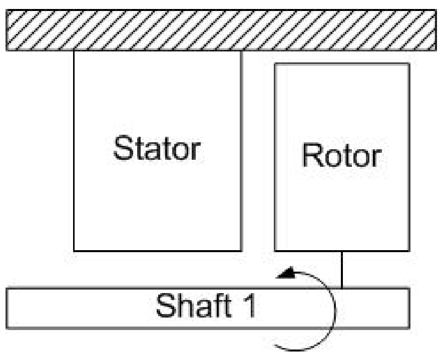

Figure 1. Traditional axial turbine sketch

The history of counter-rotating turbines starts at the beginning of 20th century. This very distinctive form of multistage turbine was invented by Fredrik and Birger Ljungstrom in Sweden in 1910. Instead of the flow being parallel to the axis of rotation through blade rows as in an axial-flow machine, the flow is radial in this kind of turbine, but this example could be considered as invention of counter-rotating machine prototype

First modern-time theoretical works in the field of axial counter-rotating turbine development for propulsion application are started in 1980's and historically we can distinguish the following works dedicated to this problem such as B.A. Ponomariev and J.F. Louis papers [1] and [2] correspondingly.

The general idea of counter-rotating turbine is usage of two shafts which rotate in opposite directions. Moving blades are located on these shafts successively without stators between them, i.e. the previous stage blade playing the role of guiding vane for the next one. Such placement of flow path elements gives benefits which were described above but needs special approach to organize flow inside the turbine. Special attention must be paid to selection of optimal rotation speed and flow radial equilibrium conditions. Layout presented in the figure below is one of a few possible shaft and blade arrangements, but others can also be found depending on overall engine design. We assume that at inlet to our counter-rotating turbine a single stator vane to guide the flow to first blade is present.

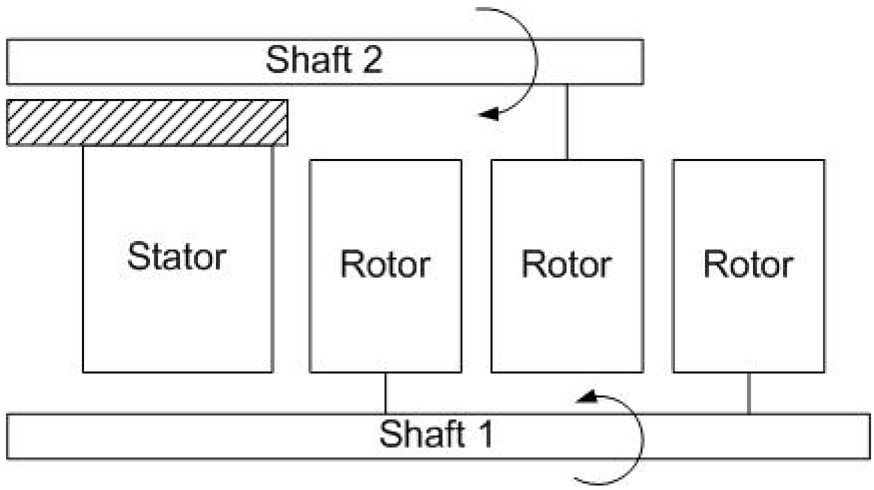

Figure 2. Counter-rotating axial turbine sketch

STREAMLINE THROUGHFLOW DESIGN AND OPTIMIZATION OF COUNTER – ROTATING TURBINE

Initial Design A four-stage aircraft LP turbine was chosen as an initial design. On the next step the redesign of this LP turbine keeping radial (meridional) dimensions in the same ranges was performed. The design and optimization procedure of a counter-rotating prototype is described in the figures below.

Initial turbine design data presented in table 1 and flow path sketch is presented below.

Table 1
Initial design data

1	total pressure at inlet	Pa	533293
2	total enthalpy at inlet	J/kg	1203438
3	total temperature at inlet	K	1103
4	stat. pressure at outlet	Pa	172000
5	mass flow rate at inlet	kg/s	91.91575
6	inlet flow angle in abs frame	deg	90
7	isentropic velocity ratio	-	0.504
8	capacity	MW	22.79
9	total-static pressure ratio	-	3.015
10	axial length	m	0.42

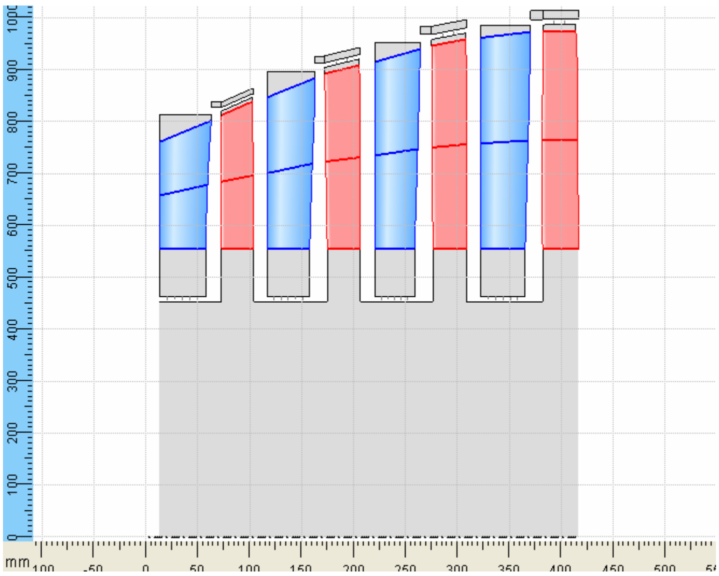

Figure 3. Initial turbine design

Counter-rotating Design (4 stages) Counter-rotating turbine design procedure was performed in the following way: the vanes of the initial turbine were removed, and the number of stages was kept in the counter-rotating turbine as in the prototype. We assume that the most correct comparison of two designs could be made only in this case. In the figure below counter-rotating turbine flow path obtained after redesign is presented. Blades which are joined to the first and the second shafts are colored red and purple colors correspondingly.

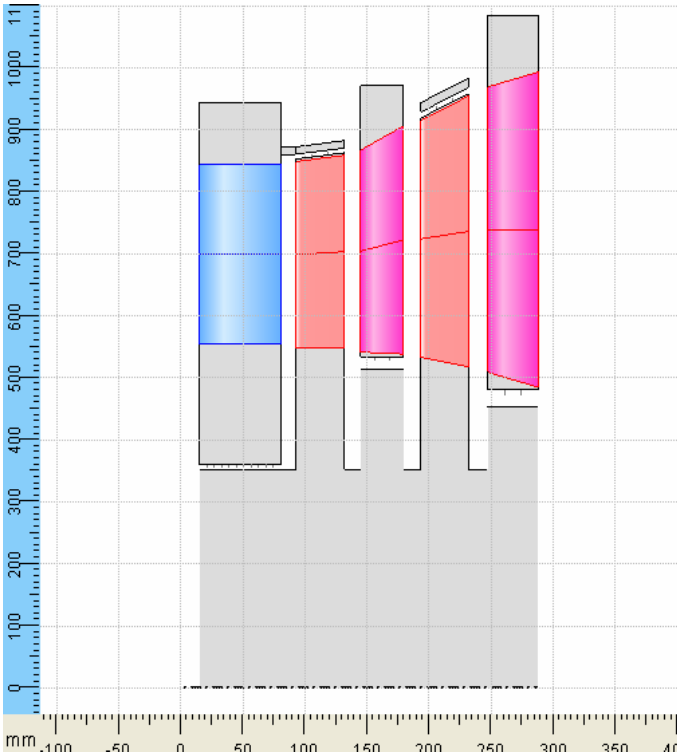

Figure 4. Counter-rotating turbine design

Selection of second shaft rotation speed is dictated by overall engine layout, i.e. joined compressor part rotation speed, gearbox presence or absence etc. It's natural that higher rotation speed allows to obtain more specific work from every stage of turbine.
Search region for the second shaft rotation speed was chosen in ranges 10-15% less than primary shaft rotation speed. To receive exact resulting speed the design of experiment study was performed. In the figure below response surface between total-to-total efficiency vs first and second shaft rotation speeds (X and Y axis correspondingly) is shown. Basing on the highest total-to-total efficiency criterion rotation the speed of 4500 RPM was chosen for the second shaft.

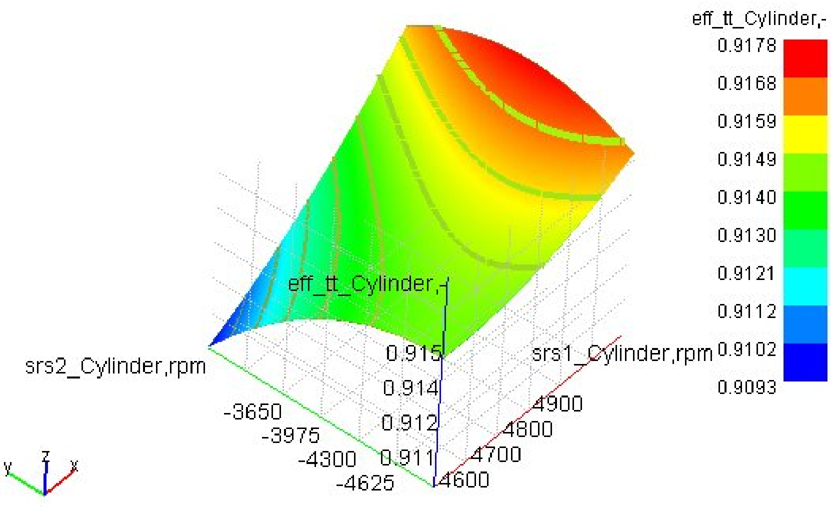

Figure 5. Efficiency chart vs rotation speeds (4 stages)

As it is clear from flow paths comparison applying counter-rotating allows to decrease axial length from 0.42 to 0.29 meters (by 30% approximately) and decrease total estimated mass of vanes and blades up to 40% approximately.

Initial and counter-rotating turbine rows view on mean station is presented in the figures below with velocity triangles.

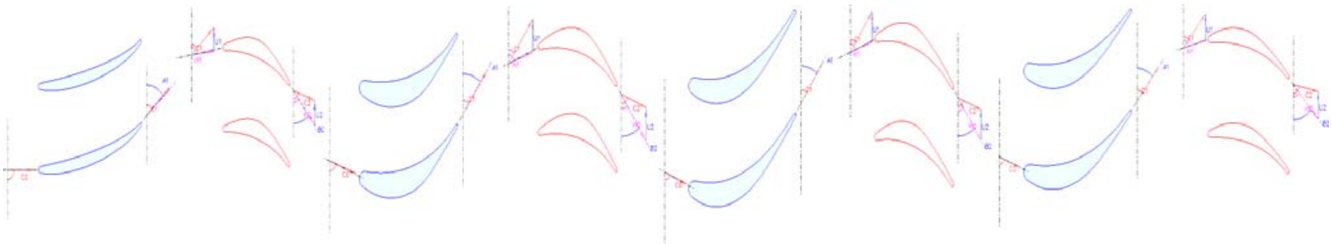

Figure 6. Initial turbine rows and velocity triangles

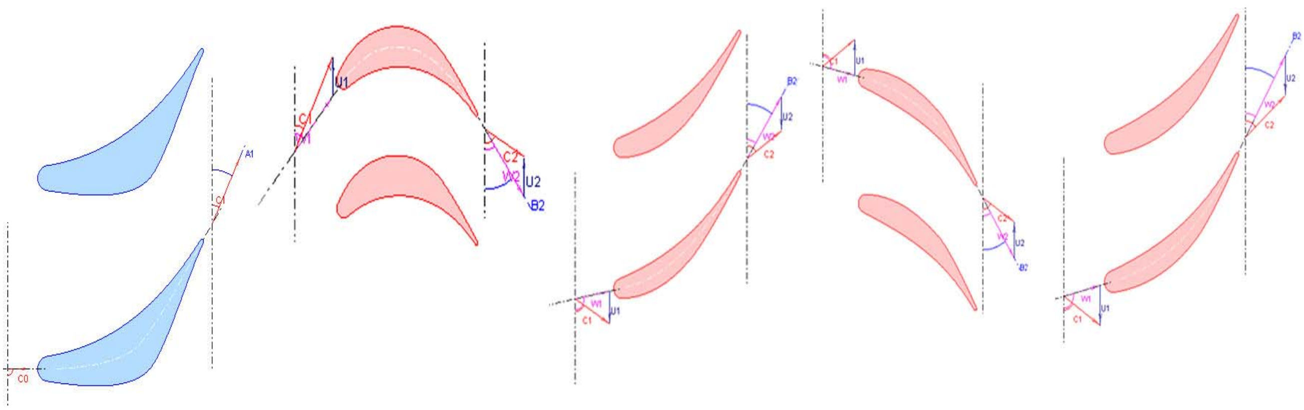

Figure 7. Counter-rotating turbine rows and velocity triangles

Counter-rotating Design (6 stages) We found that counter-rotating design with 4-stage obtained for direct comparison of flow paths gives significant decrease in length and weight of turbine. Considering different design possibilities we assume that another variation of counter-rotating design with 6 stages is applicable and will not exceed initial axial sizes and mass limitations. 6-stage design was obtained by performing axial turbine design from scratch, removing vanes, smoothing meridional dimensions and inlet metal angles adjustment to match minimal incidence angle.

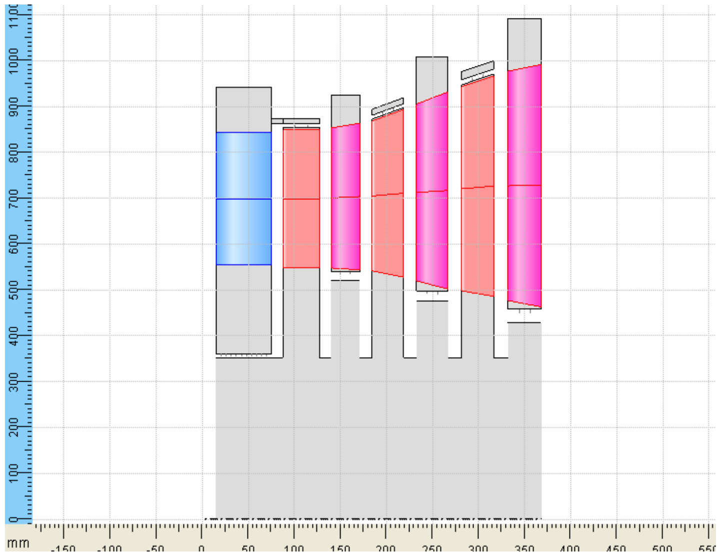

Figure 8. 6 stages counter-rotating turbine design

Selection of rotation speed for this design was performed in the same way, i.e. using DoE study for range of rotation speed variables basing on the highest efficiency criterion. For 6 stages second shaft rotation speed was chosen equal to 4500 RPM

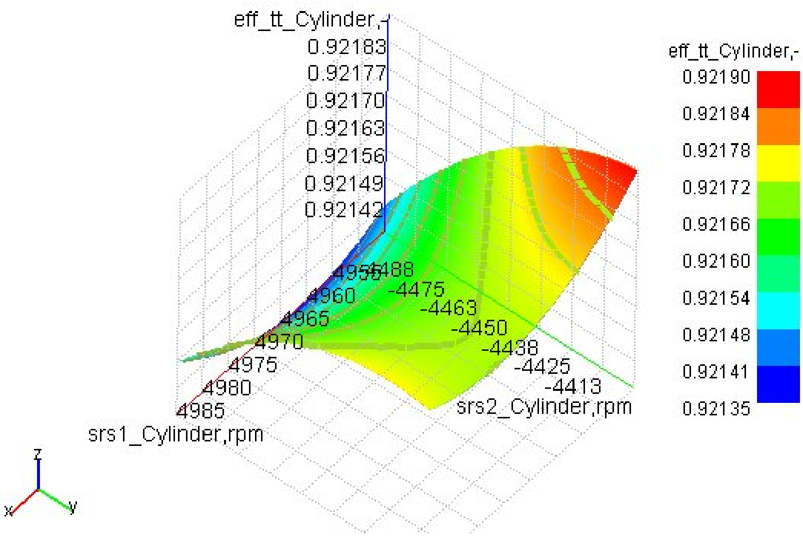

Figure 9. Efficiency chart vs rotation speeds (6 stages)

COMPARISON OF INITIAL AND COUNTER-ROTATING TURBINE PERFORMANCES

Integral Performances Comparison Comparison of integral and detailed thermodynamic performances of both turbines is presented in this part. All results were obtained in 2D streamline calculation on 7 stations spanwise along all flow path. Task formulation used for calculation was “Finding mass flow rate for static outlet pressure”, calculation was performed with assumption of turbine operating on design mode, i.e. with incidence angles corresponding to minimal losses.

Table 2
Turbines Integral Performances Comparison

		Unit	Initial Design	Counter-Rotating Design (4-stage)	Counter-Rotating Design (6-stage)
1	mass flow rate at inlet	kg/s	91	91	91
2	inlet flow angle in abs frame	deg	90	90	90
3	shaft1 rotational speed	rpm	4983	4983	4983
4	shaft2 rotational speed	rpm	-	-4500	-4500
5	isentropic velocity ratio	-	0.5050	0.2312	0.2273
6	volume flow rate at outlet	m^3/s	129.570	129.651	130.018
7	capacity	MW	22.779	22.841	23.048
8	internal total-to-static efficiency	-	0.7962	0.7972	0.8041
9	internal total-to-total efficiency	-	0.9143	0.9157	0.9214
10	axial length	m	0.42	0.29	0.37

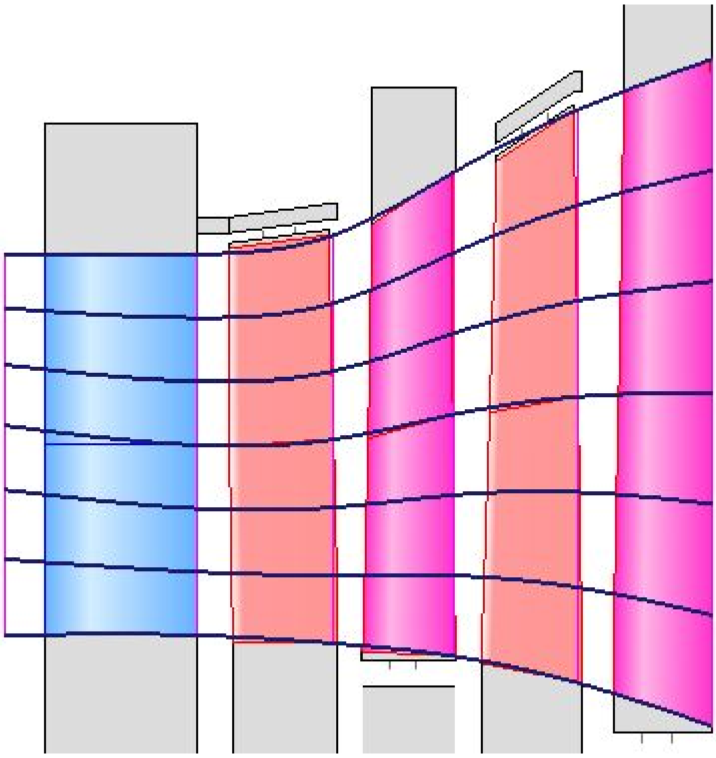

Figure 10. Calculated streamlines for 4-stage counter-rotating turbine

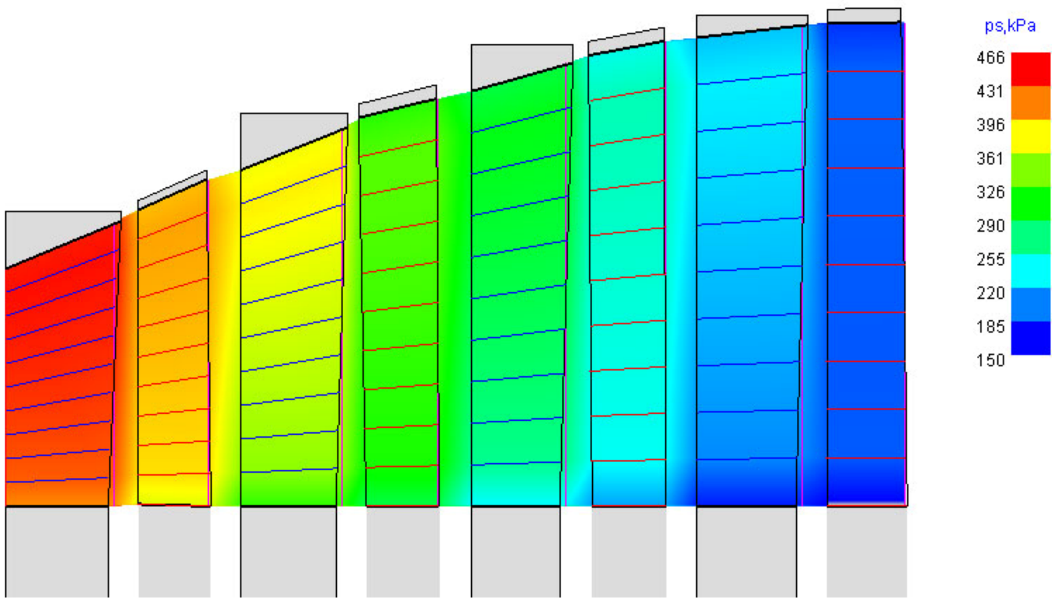

Figure 11. Static pressure distribution in traditional axial turbine

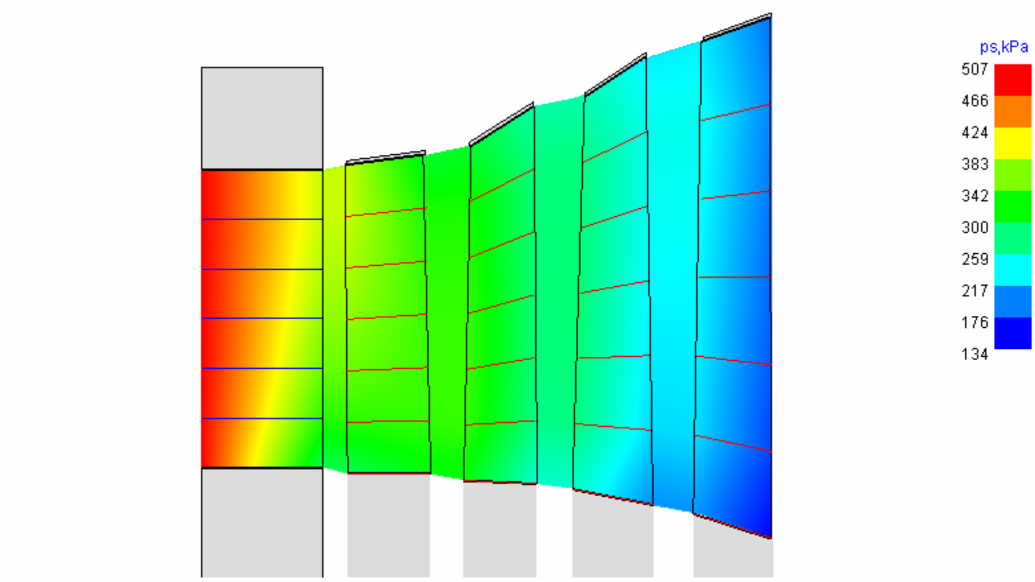

Figure 12. Static pressure distribution in 4-stage counter-rotating turbine

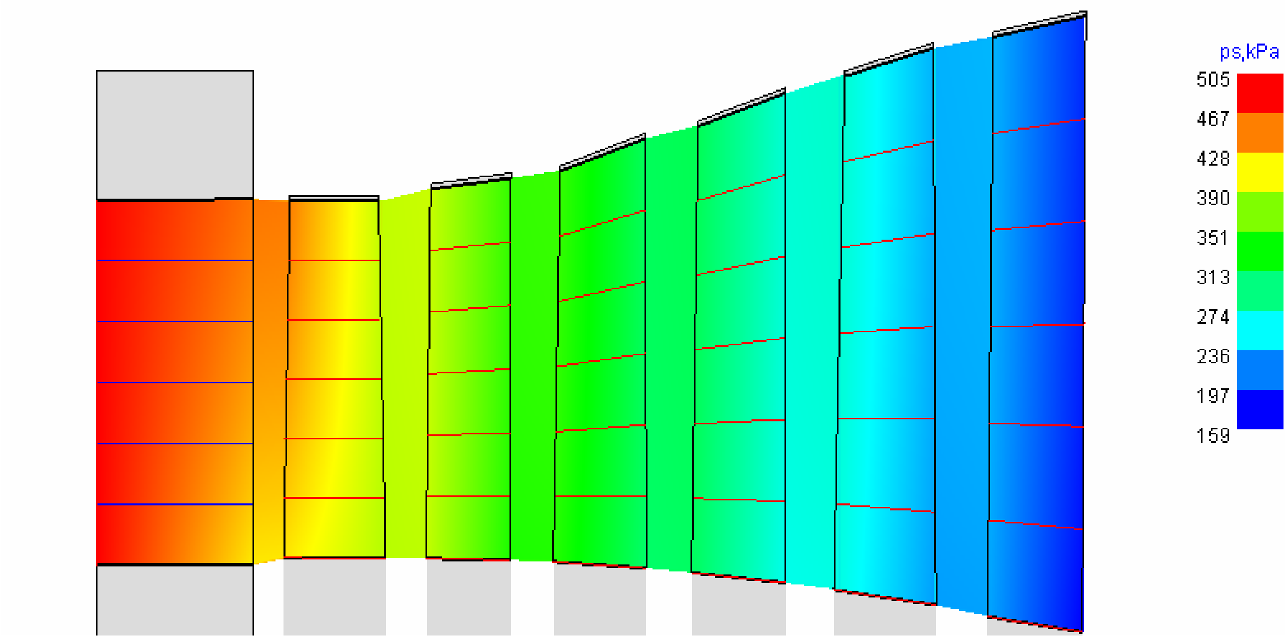

Figure 13. Static pressure distribution in 6-stage counter-rotating turbine

Aerodynamic design summary Flow path calculations show integral results presented above and distribution of thermodynamic and kinematic parameters. Kinematic data investigation showed that Mach number is in subsonic ranges for both designed counter-rotating turbines. This means that we avoid additional shock losses and the necessity of applying special supersonic profiles.
The main resume from summarizing the integral performances is that it's possible to obtain more efficient counter-rotating flow path compared to traditional prototype with decreasing overall flow path length. Counter-rotating turbine capacity with 6 stages is increased up to 0.2 MW (1%) and total-to-total efficiency by 0.57% compared with 4-stage design and by 0.7% compared with initial design.

Off-design performance analysis Initial axial turbine design and 4-stage counter-rotating design was chosen for preliminary comparing off-design performances with the initial ones. The analysis of off-design performances is the essential moment for every gas turbine, because they operate on off-design modes a significant part of their working time. Of course, the analysis of jet engine performances is incomplete without taking into account compressor maps, but in this article we will consider only turbine characteristics. Initial data for off-design performances calculations is given below. We assume that on partial modes the inlet pressure is lower due to nature of gas turbine engine cycle and for off-design calculations we select the next conventional inlet pressures.

Table 3
Off-design performances study initial data

Inlet total pressure, Pa	450000	480000	510000	533293
Initial axial turbine (AT)				
Primary shaft rotation speed, RPM	3000	3800	4200	4983
4-stage counter-rotating turbine (CR)				
Primary shaft rotation speed, RPM	3000	3800	4200	4983
Secondary shaft rotation speed, RPM	-2600	-3200	-3700	-4500

All results were obtained in 2D streamline calculation on 7 stations spanwise along all flow path for geometry presented above. Task formulation used for calculation was “Finding mass flow rate for static outlet pressure” and off-design thermodynamic and kinematic detailed parameters were calculated individually for each point in the map.

As it is clear from the obtained performance charts the designed counter-rotating turbine (red lines in the figure below) is more efficient on off-design operating modes than the traditional one (blue lines). Especially it’s clearly seen for 3000 RPM charts (60% design rotation speed), where efficiency of counter-rotating turbine is more than 4% higher.

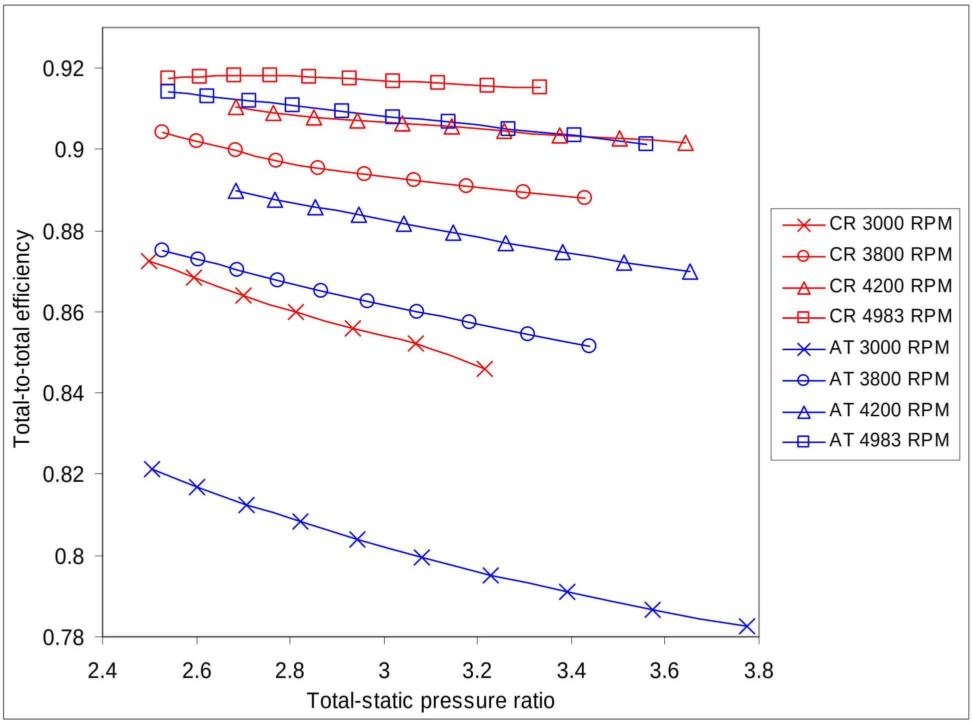

Figure 14. Off-design performances comparison

Heat transfer coefficient comparison Heat transfer conditions are very important for the reliability of the turbine at various operating modes. Influence of its conditions on design as well as off-design modes of the whole turbine will be considered in detail; such as thermal stresses and extensions, on further parts of this investigation. On the current stage we are evaluating and trying to find the major differences between heat transfer coefficients for both designs. In the figures below the heat transfer coefficient charts are presented for 2nd stage blade of initial and counter-rotating turbines. The heat transfer coefficient presented in these charts is calculated in the gas boundary layer.

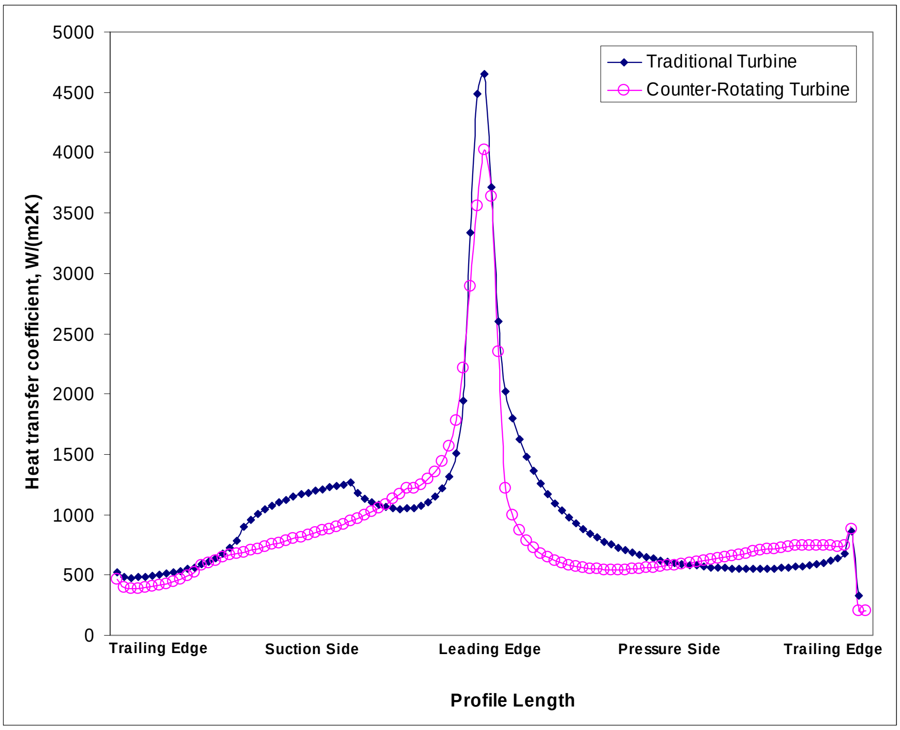

Figure 15. Heat transfer coefficient distribution on 2nd stage blade meanline section

It's clear from the charts that the gas heat transfer coefficient in a counter-rotating turbine is lower than in the traditional axial flow turbine with the same temperature conditions, it's especially noticeable at the leading edge. If we consider the heat balance the equation for a cooled turbine blade in its simplest form can be written:

$$h_g S_g (T_g - T_b) = h_c S_c (T_b - T_c) \tag{1}$$

$$T_c = T_b - \frac{h_g S_g}{h_c S_c}(T_g - T_b) \tag{2}$$

Where T_b, T_g and T_c is blade, gas and coolant temperature, h_g and h_c are the gas-side and coolant-side heat transfer coefficients, and S_g and S_c are the wetted perimeters of the blade profile and combined coolant passages respectively.

Discussion of obtained results As a result of the heat transfer coefficient comparison sources a few potential benefits from these differences, which could be useful in counter-rotating turbines in case of equality of other thermal conditions in flow and materials. If it is necessary to apply cooling to this blade, we can consider that lower gas heat transfer coefficient enables to use less cooling air or higher cooling air temperatures to obtain the same resulting blade temperature as in the initial case [3], [5].

Also, the second benefit is coming from the counter-rotating turbine part arrangement specialties. As counter-rotating blades are joined to the second shaft in the tip section, they suffer not tensile, but compression stresses and that's why it could be possible to use ceramic materials for the second shaft blades, due to the nature of ceramic materials to resist to compression stresses.

Software used All calculations presented in this article were performed in commercial AxSTREAM turbomachinery design and optimization software developed and distributed by SoftInWay Inc. (Burlington, MA, USA). Theoretical background of the software is described in [7].

Future development prospects Development of counter-rotating turbine for aircraft engine gives a new opportunity for creation of counter-rotating fan or compressor, which also can significantly decrease engine length and mass. This research is now intensively performed by scientific institutions of different countries.
Possible improvement of counter-rotating turbine could be performed by adding variable-area vane before it, but we need to keep in mind that this also increases design complexity.

CONCLUSIONS

Differences between traditional axial and counter-rotating turbines from prospects of integral and detailed thermodynamic parameters in the flow path are summarized. Three designs were considered: initial traditional axial turbine design which was chosen as prototype, and two variations of counter-rotating designs with 4 and 6 stages. Summarizing the results obtained we can find that using counter-rotating turbine with the same number of stages as in the prototype it's possible to obtain the same performance level, but to decrease turbine axial length by 30% and weight by eliminating vanes. In this case we can balance all disadvantages which are coming from the complication of design in counter-rotating turbine. The second counter-rotating design with 6 stages was considered as an example increasing performances keeping in the prototype dimensions ranges, but increasing efficiency by 0.7% compared to initial design.
Calculation for off-design operating modes showed that counter-rotating turbine efficiency is higher in wider ranges of rotation speeds which gives additional benefits, but for the final conclusion this part of research needs to be conducted jointly with compressor performance calculations.
This means that use of a counter-rotating turbine gives the opportunity not only to decrease turbine weight and length but also to create more advanced designs in terms of aerodynamic quality and overall cost-efficiency.

REFERENCES

1 B. A. Ponomariov, Y.U. Sotsenko [1992], Using contra-rotating rotors for decreasing sizes and component number in small GTE., ASME, 92-GT-414
2 J. F. Louis [1985], Axial flow contra-rotating turbines, ASME, 85-GT-218, Houston, TX, USA
3 Cohen H., Rogers GFC, Saravanamuttoo HIH [1996], *Gas Turbine Theory*, 4th Edition, Longman, Harlow Essex, UK
4 Fang Xiang-Jun, WANG Ping [2008], Research of Supersonic Axial Vaneless Rotor-Rotor Turbine, GT2008-50509, Proceedings of ASME Turbo Expo-2008, Berlin
5 Lakshminarayana, B. [1996], *Fluid Dynamics and Heat Transfer of Turbomachinery*, John Wiley & Sons, Inc., New York
6 Tang Fei, Zhao Xiaolu, Xu Jianzhong [2008], The Application of Counter-Rotating Turbine in Rocket Turbopump, International Journal of Rotating Machinery, Volume 2008, Article ID 426023
7 L. Moroz, Y. Govorushchenko, P. Pagur [2006], A Uniform Approach To Conceptual Design Of Axial Turbine / Compressor Flow Path, The Future of Gas Turbine Technology 3rd International Conference 11-12 October 2006, Brussels, Belgium