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

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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 which 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 complete 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 of turbine blading weight;
- Decreasing of 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.

STREAMLINE THROUGHFLOW DESIGN AND OPTIMIZATION OF COUNTER – ROTATING TURBINE

A four-stage aircraft LP turbine was chosen as an initial design. The next step is to redesign this LP cylinder 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.

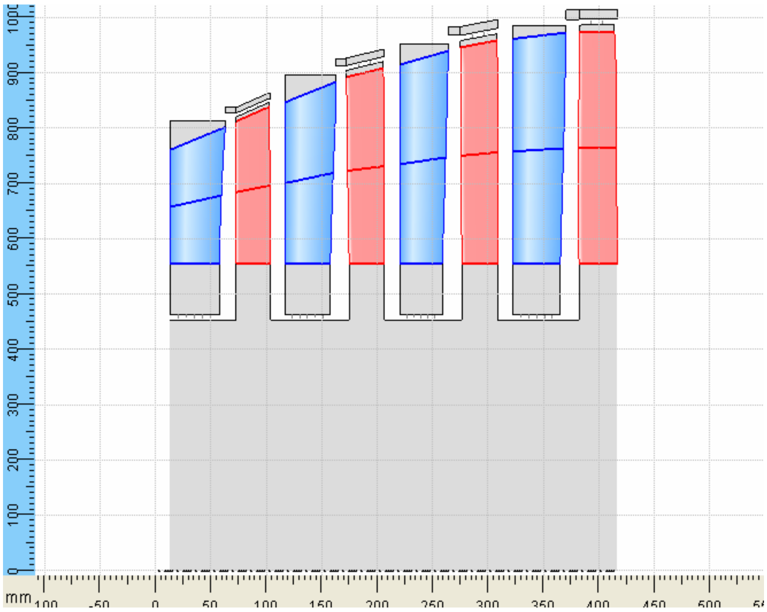

Figure 1. Initial turbine design

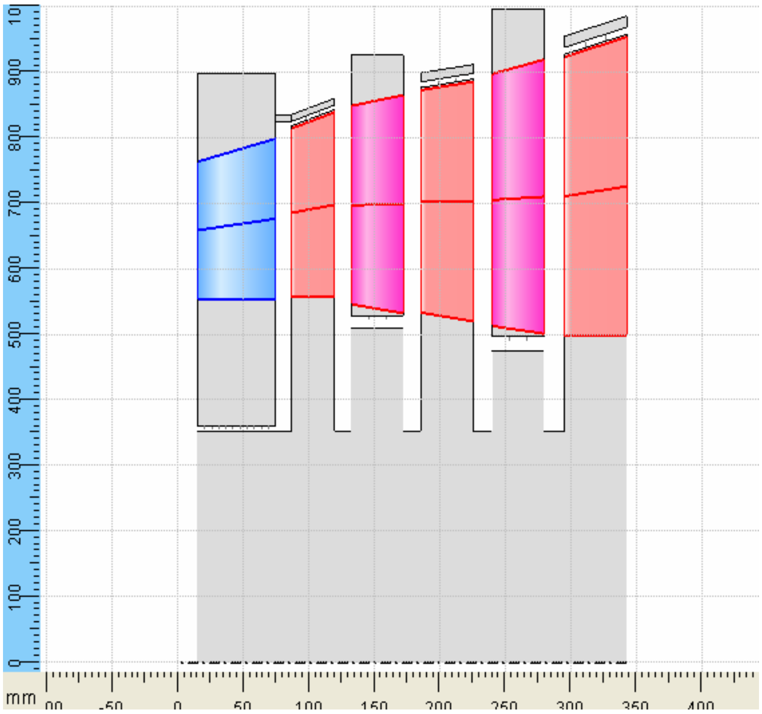

Figure 2. Counter-rotating turbine design

COMPARISON OF INITIAL AND COUNTER-ROTATING TURBINE PERFORMANCES

Comparison of integral and detailed thermodynamic performances of both turbines are presented in this part.

Table 1
Turbines Integral Performances Comparison

		Unit	Initial Design	Counter-Rotating Design
1	mass flow rate at inlet	kg/s	91	91
2	inlet flow angle in abs frame	deg	90	90
3	shaft1 rotational speed	rpm	4983	4983
4	shaft2 rotational speed	rpm	-	-2400
5	isentropic velocity ratio	-	0.5050	0.2254
6	volume flow rate at outlet	m^3/s	129.570	130.018
7	capacity	MW	22.779	23.4294
8	internal total-to-static efficiency	-	0.7962	0.8170
9	internal total-to-total efficiency	-	0.9143	0.9152
10	axial length	m	0.42	0.35

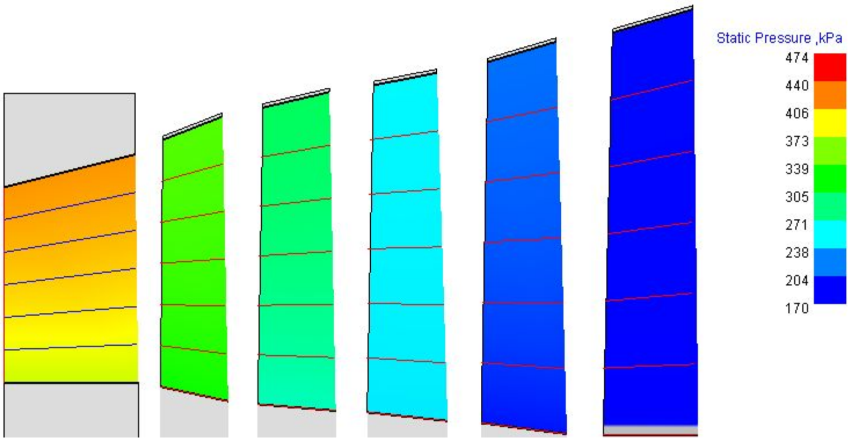

Figure 3. Static pressure distribution in Counter-rotating turbine

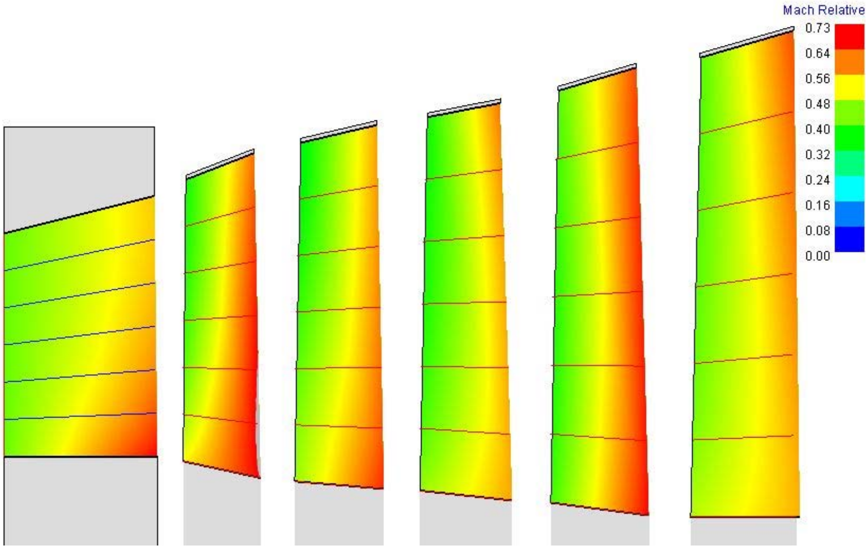

Figure 4. Relative Mach number distribution in Counter-rotating turbine

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 different modes on design as well as on off-design modes on whole turbine will be considered in detail; such as thermal stresses and extensions, on further parts of this investigation. On the current step 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.

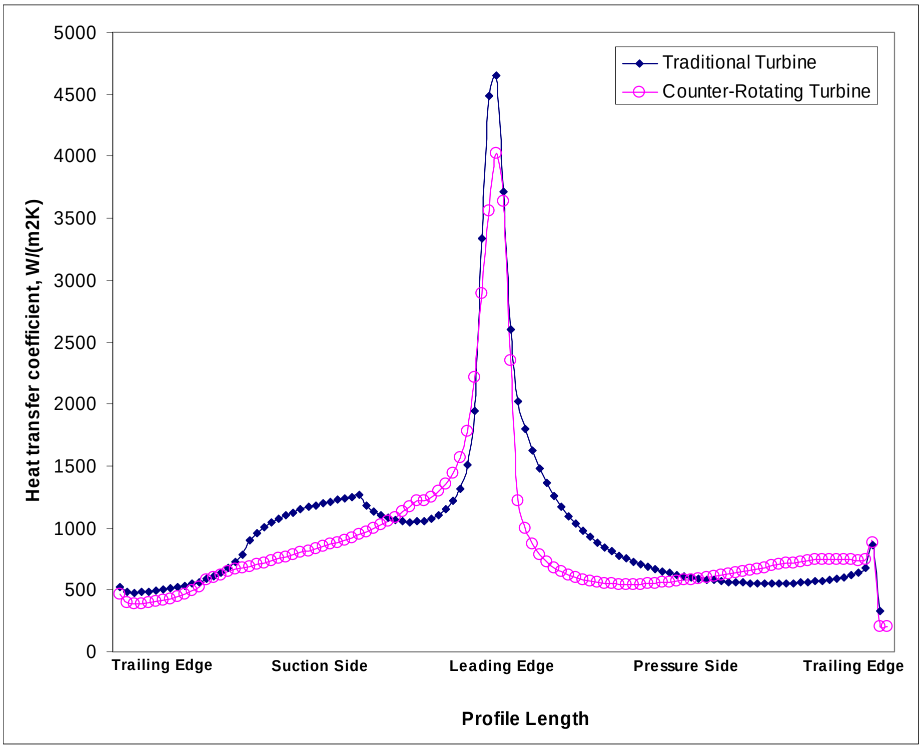

Figure 5. Heat transfer coefficient distribution on 2nd stage blade meanline section

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

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

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

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 initial case.

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

Software used All calculations presented in this article was performed in commercial AxSTREAM turbomachinery design and optimization software developed and distributed by SoftInWay Inc. (Burlington, MA, USA)

CONCLUSIONS

Differences between traditional axial and counter-rotating turbines from prospects of integral and detailed thermodynamic parameters in flow path are summarized. Heat transfer coefficient distribution in profile boundary layer was investigated. Use of a counter-rotating turbine gives the opportunity to create more advanced designs in terms of aerodynamic quality and overall cost-efficiency.

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