Proceedings of the International Gas Turbine Congress 2003 Tokyo
November 2-7, 2003

IGTC 2003 Tokyo TS-116

MICROTURBOCOMPRESSOR FUEL CELL – BASED PLANTS FOR HYBRID ENGINE APPLICATIONS

A.Soudarev[1], A.Souryaninov[1], A.Molchanov[1], P.Avran[2], L.Lelait[2]

1). Boyko Research-Engineering "Ceramic Heat Engines" Center (NIZ KTD), 195221, S.Petersburg, Russia, E-mail:soudarev@boykocenter.spb.ru, Telefax: 7(812)225 34 53
2). EDF, Paris, France, Site des Renardieres, 77818 Moret–Sur-Loing Cedex, E-mail: Laurent.lelait@edf.fr, Telefax: (49)72161051330

ABSTRACT

The hybrid engine on the basis of the solid oxide fuel cell (SOFC) and gas turbine unit (microGTE) is the most promising power system because it allows to Increase the efficiency and to improve the environmental and life values through a joint positive effect achieved by combination of the thermo-dynamical schemes of the both components. A feasibility to use microGTE as a part of hybrid engines based on use of blade-free type turbomachines (turbine and compressor) where the conventional stationary and rotating circular blade systems are replaced by the three-dimensional conic channels (tunnels) systems is demonstrated in the paper. The thermo-technical, hydro-dynamical and strength data obtained by the tunnel turbine stage calculations are presented as well. They indicate that a similar design could be actually embodied in practice. It is also demonstrated that with reduction of the absolute sizes of microGTE, it becomes possible to increase the efficiency and improve the life values for the plant through elimination of the losses linked with the internal leaks and overleaks from the turbomachines (Soudarev et al,2003).

INTRODUCTION

The turbocompressor microGTE must be adapted to a combined operation with FC since the conventional design GTEs do not meet the main requirements to hybrid engines formulated even at the stage of their initial studies, namely:

-turbomachines must have small sizes and high speed and, at the same time. Efficient;

-there must be used the air lubrication to ensure a high dynamical stability of the turbomachine rotors when the peripheral velocity in the bearings is up to 600-700 m/s;

-with TIT being up to 1000-1350^{0}C, the turbine flow passages should be uncooled;

-compact heat exchangers must be available (the compactness factor is up to 1000 m^2/m^3 and higher);

-the combustors must be environmentally friendly (first, it applies to NOx and CO emissions;

-use of high speed electrogenerators with reduction gear-free connection with the turbocompressor rotors must be envisaged.

Use of the radial gas turbine in small power plants such as micro- and nano-gas turbine engines is preferable compared to the axial design both in terms of structure simplicity and, in some applications, arrangement convenience and, due to improved control adaptability, lower aerodynamic losses at low media flows.

With gas-turbine engine power lowering, the absolute sizes of the turbomachines (engine turbine and compressor) are getting smaller, too. In the meantime, i.e. with axial or radial blading, a negative effect of the gaps between the rotor elements of the flow passage tends to increase whereby reducing essentially the turbomachine efficiency(Isomura et al, 2001), (Epstein et al.,2002). Hence, there is a need in finding a design approach that would allow to reduce leaks and overleaks of the media across their flow passage.

OBJECTIVE OF STUDY

The objective of the studies conducted is, finally, to verify the results of the gas-dynamic calculations for the radial tunnel turbine using an experimental gas-dynamic cold test bed. At such tests, the tunnel turbine inlet conditions differ from those of a conventional design engine.

While conducting the like studies it is very important to ensure a reliability and strength of the structure tested. So, the objective of the given paper is to study **the heat-stressed state** of the tunnel turbomachine with maintaining both the geometric and **gas-dynamic similarity** which preconditions are an equality of the reduced flows and speeds for the standard and model rotors.

Manuscript Received on March 31, 2003

TUNNEL TURBINE DESIGN

In the tunnel turbocompressor turbine and compressor, the conventional axial or radial blades are replaced with the conic channels. The gas-dynamic and thermal- technical calculations indicate a feasibility to produce competitive turbocompressors for microGTE. Though to achieve speeds required , it would be necessary to use supports having a very low friction loss, i.e. gas-static bearings. The studies performed revealed that the 1% bleed-off from the air flow in the compressor allows not only to safeguard a stable operation of the radial and thrust bearings but, at the same time, to provide a cooling for the stator parts. Furthermore, the same air is used to eliminate a reduction in the gas-dynamic efficiency of the tunnel compressor through increasing the compression operation at heating of air that flows across the compressor tunnels as a result of heat overflows from the "hot" tunnels of the turbine part to the "cold" tunnels of the compressor part of the turbocompressor.

The design scheme of the tunnel stage of the turbomachine is shown in fig.1.

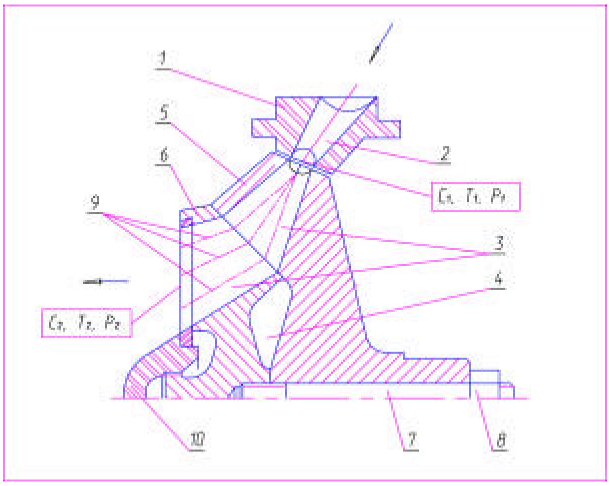


Fig.1 Schematic diagram of a turbomachine stage

1- nozzle vanes (NV) with tunnel channels 2; 3 – tunnel channel of turbine wheel (TW); 4 – TW; 5 – peripheral disc of TW; 6 – near-axial disc of TW; 7,8 – elements of TW disc connection; 9 - axes of 3 groups of channels of various geometry; 10-deflector; ← - stream direction in tunnel turbine stage).

The high temperature working media stream (gas) in the turbine stage supplies from the GTE combustor to the stator device 1 formed by fixed tunnels 2 made as circular cones with straight axes where it is accelerated. Due to gas expansion, its absolute velocity C_1 increases while pressure P_1 and temperature T_1 decrease. Further on gas supplies to the rotating tunnels 3 (of the same geometry) of the turbine wheel 4 that includes two interconnected discs 5 and 6. In the rotating turbine tunnels 3, gas keeps on expanding with its pressure and temperature decreasing to P_2 and T_2 whereby converting the potential gas energy into the mechanical energy of the turbine wheel. The expansion and exit of gas from the turbine wheel take place across three groups of axes of 9 rotating tunnels 3.

In the compressor stage of the turbomachine, the direction of the working media (air) reverses; thus, it is compressed first in the system of the rotating casing tunnels 3 and then in the stator device 1. So, you may regard the tunnel stage of the turbomachine in the 1st approximation as a reversed one.

THERMAL-TECHNICAL AND GEOMETRIC PARAMETERS OF TUNNEL TURBINE MODEL

Based on the engineering capabilities of the current experimental test bed, a power of the standard 108KW turbine was chosen and the initial media parameters were specified. Then, a recalculation was performed for its model (table 1) with maintaining the research duties on the cold gas-dynamic test bed maintaining the complete geometric (scale 1:1) and gas-dynamic similarity.

Table 1. Main parameters of tunnel turbine model

N	Name	Value
1.	Reduced flow at rating, $\frac{kgK^{0,5}}{s \cdot MPa}$	27.29
2.	Reduced speed at rating, $\frac{rpm}{K^{0,5}}$	2978.7
3.	Air temperature at turbine inlet, ^{o}K	320…360
4.	Air pressure at turbine inlet, MPa	0.348
5.	Air temperature at turbine inlet, ^{o}K	250….295
6.	Air pressure at turbine outlet, MPa	0.1013
7.	Air flow across turbine, kg/s	0.53…0.49
8.	Rotor speed, rpm	53500..57500
9.	Inlet diameter, mm	130
10.	Number of nozzle vanes channels, pc.	17
11.	Nozzle vanes diameter at outlet, mm	105
12.	TW diameter at inlet, mm	105
13.	Number of groups of TW channels of the same geometry, pc	3
14.	Absolute velocity at nozzle vanes inlet, m/s	367
15.	Absolute velocity at nozzle vanes outlet, m/s	156
16.	Peripheral velocity at TW inlet, m/s	300

The standard turbine geometry was calculated on the basis of the initial parameters of the thermal-dynamic cycle of a standard GTE. As a result of the multi-version iterations and by the combined calculation of the geometry and gas-dynamics of the turbine wheel, all the geometries of the external profiles, channel sizes and channel locations were identified. A major factor allowing to opt for an optimum version is the interchannel baffle thickness, the larger it is the higher is the disc strength.

STUDIES OVER THE HEAT-STRESSED STATE OF THE MODEL TURBINE WHEEL OF THE TUNNEL TURBINE

The stressed state of the peripheral disc was performed for speed n=60 000 rpm, i.e. for a duty somewhat higher the maximum for the tests planned Even the 1st calculations of the strength of the designed turbine wheel demonstrated a potential in improving its stressed state. As follows from the results of calculation of the base-line version of the peripheral disc (fig.2), the maximum stresses on the conic part (along radial directions in the sections of location of openings at the exit of a group of channels) of the disc periphery amounted to 780 MPa.

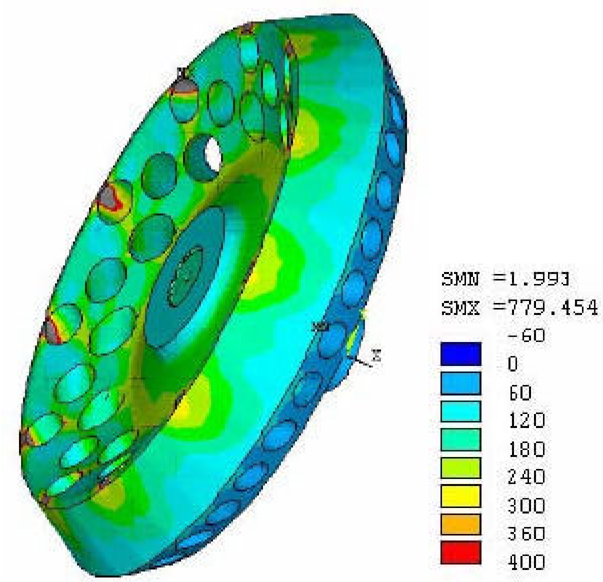


Fig.2 Distribution of equivalent stresses in the peripheral disc of the tunnel turbine

Provided smooth borings are made in these locations (fig.3), the maximum stresses will be around twice lower (375 MPa). In the meantime, the main stresses in the near-axial disc will be not above 268 MPa (fig.4,5) at the same design conditions (n=60 000 rpm). The performed research of the stressed state of discs allows selection of a material that would best ensure their reliable operation during the cold gas-dynamic tests.

The model tunnel turbine stage (NV + TW) was manufactured (fig.6) of the aluminum alloy AK-4 which main properties are in Table 2. A similar alloy AK-6 could be employed.

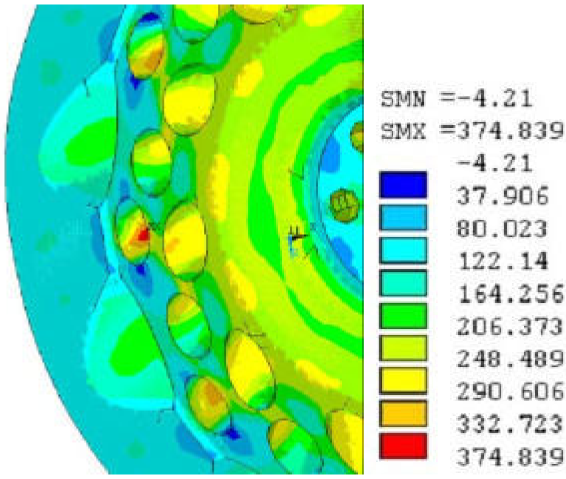


Fig.3 Distribution of main stresses in the peripheral disc at interface with the near-axial disc

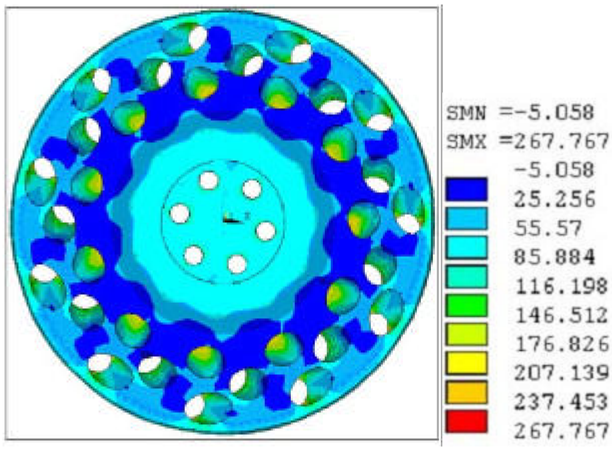


Fig.4 Distribution of main stresses in the near-axial disc (at interface with the peripheral disc).

Table 2 Main mechanical characteristics for aluminum alloys AK-4 and AK-6

N	Name	Value	
		AK - 4	AK - 6
1.	Tensile strength, MPa	440	480
2.	Yield strength, MPa	320	380

These materials are capable to provide a fairly admissible safety factor for the experimental model turbine wheel. The manufactured tunnel turbine stage is supposed to be supplied to the test bed during the 2nd quarter of 2003. At present, the bed is being equipped with the measuring instruments in view of the approaching studies. The 1st experiments findings will be presented at the conference.

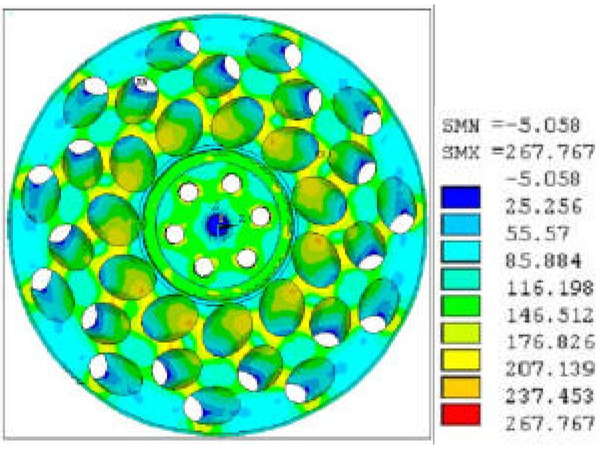
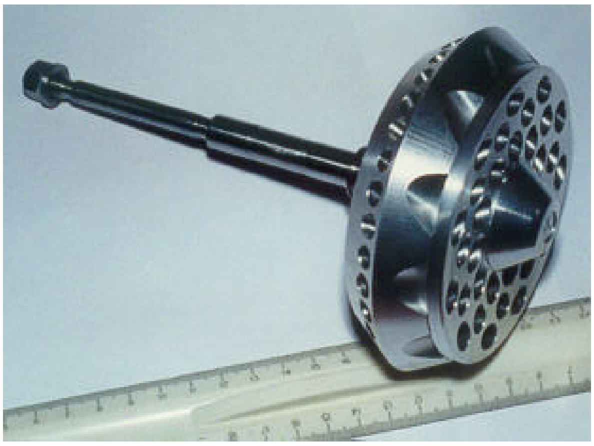


Fig.5 Distribution of main stresses in the near-axial disc (stream exit plane)

a)



b)

Fig.6 Nozzle vanes (a) and turbine wheel (b) of the tunnel turbine stage

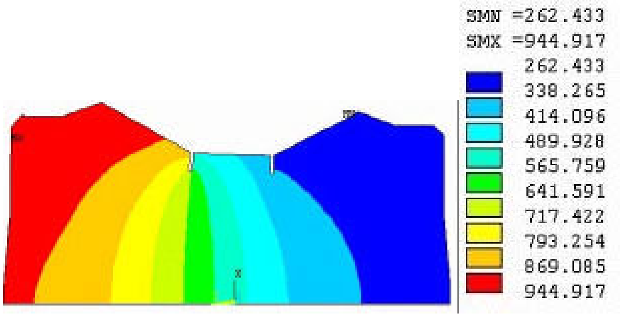
CERAMIC TUNNEL TURBOCOMPRESSOR (HEAT-STRESSES STATE)

A burning challenge of the current gas turbine manufacture is to produce robust high temperature elements for gas turbine plants using structural ceramics (SCMs).

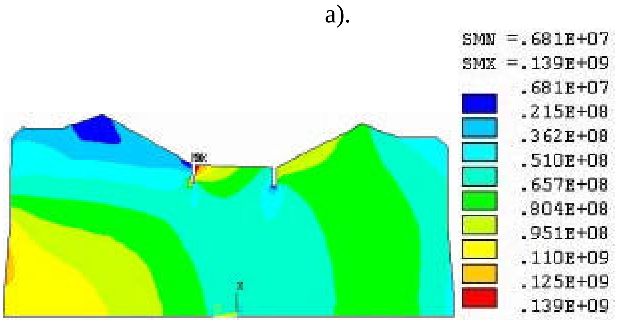
In so doing, the plant efficiency is increased due to a considerable increase in the initial gas temperature upstream of the turbine, this efficiency being notably higher that for a similar metal structure since parts and components of SCMs do not need a cooling. At present, the gas temperature of 1623^{0}C can be achieved (Epstein,1999), (Isomura et al.,2001). Since the SCMs compression strength is essentially higher the tensile strength, it is necessary to embody such designs of the gas turbine rotors where the **rotor blades do not stretch** under the effect of the centrifugal forces but **compress.**

The most rationally the like design is integrated with axial turbines when a heat resistant thermally stable ceramic shell is mounted loosely on a metal core secured in the disc. The core has a platform on the periphery against which the ceramic shell (Tatsumi et al., 2002) is pressed by the centrifugal forces at rotation. It is a more complex task to find an engineering approach relative to the radial turbines (RT) with radial blades. In the case a metal-ceramic centripetal tunnel turbine is applied, you can provide **a ceramic lining for tunnels** which is achieved through using ceramic conic sleeves that are mounted loosely in the conic tunnels of the turbine wheel. The ceramic sleeves are compressed by the centrifugal forces of the masses proper that are generated at the rotation of the CPT wheels. The like design approach would be best applied to the 5-500kW mictoGTEs. If it deals with a nano-GTE, then the absolute sizes of the turbomachines are so small (the turbine and compressor diameter for a nano-GTE is 4 to 10 mm (Epstein et al., 1999), (Soudarev et al.,2001), (Soudarev et al., 2002a), (Soudarev et al., 2002b) that the entire turbocompressor is manufactured as an all- ceramic structure. In this case the TW design must meet not only the strength targets but also those of thermodynamics since a heat transfer from the “hot” turbine part of TW to the “cold” compressor part will result in the engine efficiency decreasing due to augmenting of the compression work of the heated air. Various design approaches for the turbine-compressor connection were examined that would minimize the heat flux from the turbines to the compressor. The limitations to identify the temperature pattern in the tunnel turbocompressor rotor were taken from the gas-dynamic calculations of the flow along the conic channels and in the gas-static bearings. The heat-stressed calculation was carried out by two stages. First, the simplified axisymmetrical model of the turbocompressor was examined; in doing so, the conic channels were replaced with the equivalent annular channel which allowed identification of the heat-stressed state of the rotor and option for optimum geometries and design of the turbocompressor. At the 2nd stage, some versions of the tunnel turbocompressor at 3-D statement of problem were studied which enabled to pin-point the heat state of the rotor and determine the local stress concentrators with their subsequent minimization.

The numerical study of the heat-stressed state of TW at various rotor designs indicates that with decreasing the TW cross section area through which a heat exchange proceeds between the turbine and compressor, the efficiency of the turbomachine rises which makes the processes proceeding therein closer to isothermal but the maximum stresses in the TW rotor increase sharply.



a).



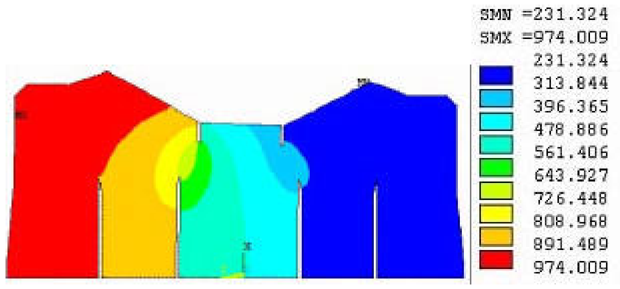
b).

Fig.7 Temperature distribution in ^{0}C (a) and maximum stresses in Pa (b) in nano-turbocompressor (version 1) at:

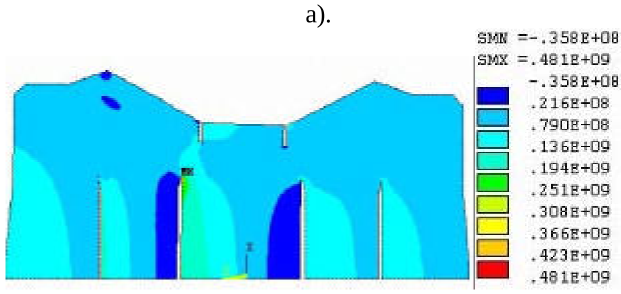
-peripheral velocity – 300 m/s

-TIT – 1200^{0}C.

Thus, with the TW rotor made with two cuts on periphery (fig.7), the maximum temperature (TW compressor) is 944^{0}C while the minimum temperature (TW compressor) is 262^{0}C. With adding four more radial cuts from the central axis of the rotor (fig.8), these temperatures vary up to 974^{0}C and 231^{0}C, accordingly while their difference varies from 682 to 743, i.e. there is a 10% increase. At the same time, an optimum approach could be found at the expense of a rational trade-off between the strength and heat exchange values. It will be linked with such characteristics of SCMs as heat resistance and thermal stability.



a).



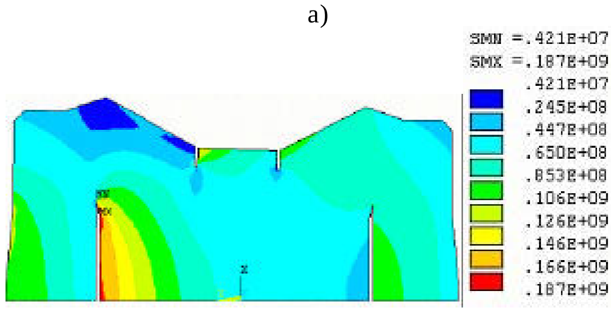
b)

Fig.8 Temperature distribution in ^{0}C (a) and maximum stresses distribution in Pa (b) in nano-turbocompressor (version 3) at;

-peripheral velocity – 300 m/s;

-TIT - 1200^{0}C.

a)



b)

Fig.9 Temperature distribution in ^{0}C (a) and maximum stresses distribution in Pa (b) in nano-turbocompressor (version 3) at;

-peripheral velocity – 300 m/s;

-TIT - 1200^{0}C.

E.g., with use of the alumo-boron-nitride SCM (Patent of France, 2002) to manufacture TW, an optimum design version would be one with two extra central radial cuts (fig.9). Then the temperature difference will increase by 20^0C and the stress will increase by 35%.

SUMMARY

1.Structure, designing basics and techniques of manufacture of the tunnel type turbine which prototype is being now prepared for gas-dynamic cold tests were developed.

2.The numerical study over the heat-stressed state of the tunnel turbocompressor of the structural ceramics was carried out. The effect of the design variations for the tunnel turbocompressor rotor on the temperature pattern and its stressed state was demonstrated.

REFERENCES

Epstein A.N. et al,1999, USA Patent, No 3 443 792, NKI 416-500.

Epstein A.N., 2002, “Micro-turbine engines for soldier power”, Defense Science and Technology Seminar Future Warrior Systems”, Massachusetts Institute of Technology.

Isomura K., Murayama M., Kawakubo T., 2001, “Feasibility Study of a Gas Turbine at Micro Scale”, ASME paper, 2001-GT-101, New Orleans, L.A, p.8.

Tatsumi T., 2002, “Development of the 300kW ceramic gas turbine CGT 302”, Ceramic Gas Turbine Design and Test Experience. Progress in Ceramic Gas Turbine Development, ed. Mark van Roode, Mattison K., Ferber, David W.Richerson, ASME PRESS, vol.1, ch.15, pp.331-360.

Soudarev A.V., Grishaev V.V., Souryaninov A.A., Molchanov A.S., 2001, “Research into technological and structural capabilities of alumo-boron-nitride ceramic material for gas turbine application”, ASME paper, 2001-GT-464, New Orleans, p.9

Soudarev A.V., Tikhoplav V.Yu., 2002a, “Ceramic Gas Turbine Design and Test Experience. Progress in Ceramic Gas Turbine Development”, ed. Mark van Roode, Mattison K., Ferber, David W.Richerson, ASME PRESS, vol.1, ch.32, pp.683-707.

Soudarev A.V., Souryaninov A.A., Tikhoplav V.Yu. et al, 2002b, “Cooled metal-ceramic gas turbine rotor blade”, Patent, Claim No 99100216/20 0f 10.01.99, filed 03.04, 2002.

A.Soudarev, A.Souryaninov, V.Yu.Tikhoplav et al., 2003, International Gas Turbine Congress, IGTC 2003 Tokyo TS-041, Tokyo, p.8.

Patent of France, 2002, No 02 10977, 75INPI Paris.