DESIGN-POINT, OFF-DESIGN MEANLINE PERFORMANCE ANALYSIS AND CFD COMPUTATIONS OF THE AXIAL TURBINE TO MICRO GASTURBINE ENGINE

Wojciech Bar

Rzeszow University of Technology, Department of Aircraft and Aircraft Engines
Powstancow Warszawy Street 8, 35-959 Rzeszow
tel.: +48 17 965 1674
e-mail: wbar@prz.edu.pl

Michał Czarnecki

Rzeszow University of Technology, Department of Aircraft and Aircraft Engines
Powstancow Warszawy Street 8, 35-959 Rzeszow
tel.: +48 17 965 1609
e-mail: czarnecki_m@prz.edu.pl

Abstract

Presented paper consists detailed analysis of the turbine to micro gasturbine. Calculatios was made for axial turbine with given mass flow rate m=0.15 [kg/s] and total inlet temperature T_{3*}=950 [K]. The results of Design-Point, Off-Design Meanline Performance analysis and CFD computation are presented. The differences between typical size gasturbine turbines and turbines to micro-gasturbines are highlighted and some guidelines for designers of microgasturbines are given. As a major conclusions for designers of microgasturbines there is a considerations that require to be very carefull and forewarn that typical corelations for commercial gasturbines could't be take it as granted.

In particular, compression ratio – solution possibility, micro gasturbine rotor speed, turbine expansion ratio Π_T as a function of compressor and turbine efficiency, efficiency of turbine as a function of Reynolds number for constant specific speed coefficient and constant specific diameter coefficient, turbine to micro gasturbine performance map for different normalized rotational speed, 2D and 3D contour plots of entropy, relative Mach number at middle cut of the blade, static pressure at middle cut of the blade are presented in the paper.

***Key words**: turbine, gasturbine, axial turbine, micro, CFD*

1. Introduction

In literature we can find that micro-turbines are usually referred to units of less than 350kW but the micro gasturbine engines are referred to units of less than 4.45 [daN] of thrust or 0.5 [kg/s] of mass flow. A turbine to micro-gasturbine engine is a unit of around 10kW. That gives one order of magnitude reduction in Reynolds number. Unfortunately this reduction leads to lower efficieency. A comparison of the today full scale gasturbine and micro gasturbine solutions show other big differences which has big impact on turbines to micro-gasturbines engines. Principal differences are presented below:

Engine:
- Mass flow rate – 100-600% smaller,
- rotational speed – 300-800% larger.

Compressor section:
- Pressure ratio for compressor – 100-150% smaller,
- Target efficiency for radial compressor – 15-20% smaller.

Turbine section:

- Pressure ratio for turbine per stage - 15% smaller,
- target efficiency for axial turbine – 15-20% smaller,
- blade height from tip to hub – 80% smaller,
- Reynolds number – full scale turbine $Re>10^6$, micro turbine $Re\approx10^5$.

2. Design parameters studies of micro gasturbine

To fully understand the design of turbine to micro-gasturbine engine we must analyze design parameters of micro gasturbine. Design process starts from design studies, at starting point number of available information is limited. Presented studies were made for micro gasturbine equal to "Schreckling's design". This type of design consist one stage radial compressor and one stage axial turbine (1R-1T).

2.1. Required power

As a principle in general case balance between compressor and turbine should be equal or insensibly greater than power required by compressor. First step consists verifying of solution possibility. Fig. 1 presents compression ratio as a function of mass flow rate for produced micro gasturbine.

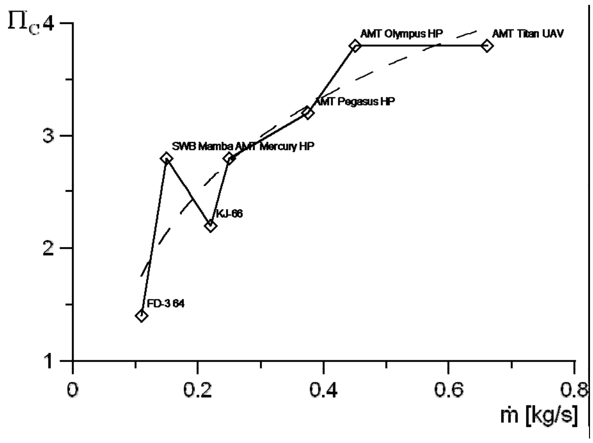

Fig. 1. Compression ratio – solution possibility

Available compression ratio for micro gasturbine ratio may be approximate be following equation:

$$\Pi_{C(\dot{m})} = 1.23 \cdot ln(\dot{m}) + 4.47, \tag{1}$$

where:

Π_C - expansion ratio of compressor,

$\dot{m}$ - mass flow.

Equation (1) is useful because there is a lack of useful information about micro gasturbine compressors. Automotive characteristics don't match for micro gasturbine. Rotor of compressor is similar but diffuser differs considerably. Efficiency of compression process in first assumption for estimated compression ratio is $\eta_C = 0,7$.

2.2. Rotational speed

Simplified design case when compressor rotor is taken from turbocharger puts on limitation on rotational speed. For radial compressor rotor rotational speed is in range from 100 k to 180 k rpm. Lower rotational speed values correspond to pure axial design.

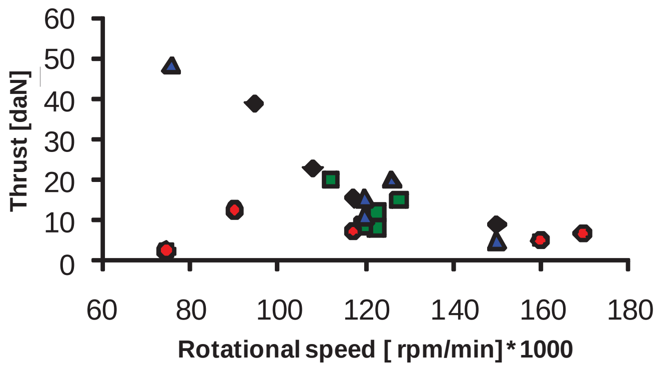

Fig. 2. Micro gasturbine rotor speed

Figure 2 shows shaft rotational speed in reference to thrust of micro turbine. There are a four groups of manufactures, diamonds – AMT Netherlands, square-JetCAT USA, triangles-SWB Turbines, circles – remaining manufactures. Majority developments of the micro turbines come from KJ-66 project and similar designs have rotational speed from in 120k rpm range.

2.3. Degree of reaction

Degree of reaction affects, and is affected by, the design in many ways. The meanline velocity triangles are determined by the choice of reaction, and this in turn determines some of the principal features of the blade design. The velocity triangles show that a low-reaction design will require a large acceleration and turning of the flow in the stator blades, and a high inlet velocity to the rotor blades. In a very high reaction design, the acceleration in the stator is small, but that in the rotor is large. Both effects lead to a lower efficiency than is possible with blades of moderate reaction where both the stator and the rotor contribute significantly to the acceleration of the fluid. The choice of reaction also has an influence on the mechanical design of the blade. Micro gas turbines are developed with degree of reaction for one stage axial turbine Λ=0.5. That's the only proposal for design presented by Thomas Kampst. This type solution is characterized by symmetric divided stage work and almost symmetric velocity triangles.

2.3. Efficiency & expansion ratio

Lack of performance map for axial micro gas turbines required searching all area for estimate design point. Search method was based on static optimization. For our application systematic search method was used that is perfect for searching unknown areas. In presented charts step 0.1 was taken for each axis that gives 256 points by chart. That was a compromise by speed and accuracy of the method. Estimating expansion ratio was taken for turbine inlet temperature 950 [K], and mass flow ratios 0.15, 0.30, 0.45 [kg/s] was chosen.

As a goal function turbine expansion ratio $\Pi_T < 1.95$ was taken. In practice, automotive industry is using expansion ratio $\Pi_T \approx 1.5$ in radial turbines. Lined areas on Fig. 3 shows designs that required number of turbine stages greater than one with compression ratio taken by presented approximation. In this area solution with one stage is also possible, but that hits engine compression ratio greatly.

2.4. Diameters

Limit on turbine stage diameter comes from relative velocity U that must be smaller than < 400 [m/s]. In practice turbine outer diameter for micro gasturbine solution is smaller than 100 [mm].

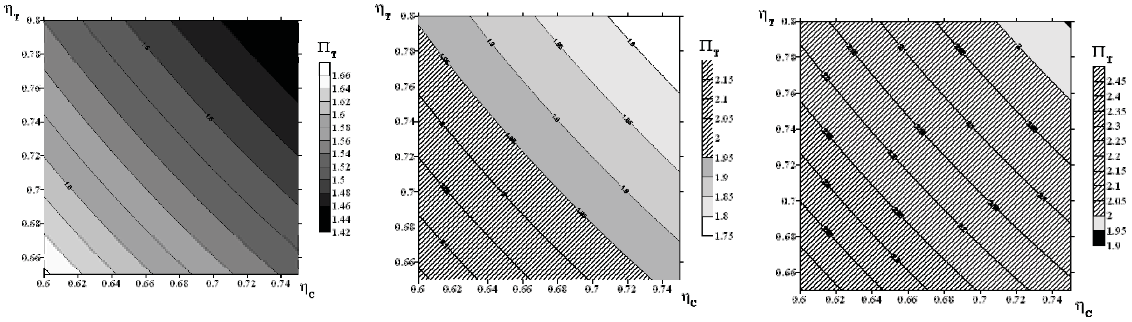

Fig. 3. Turbine expansion ratio Π_T as a function of compressor and turbine efficiency for mass flow rate = 0.15, 0.30 and 0.45 [kg/s]

2.5. Reynolds Number

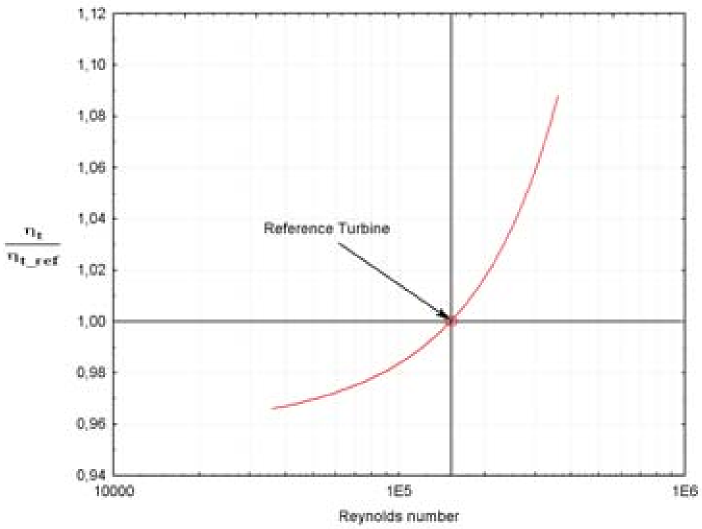

Fig. 4. Efficiency of turbine as a function of Reynolds number $\mathrm{Re} = \dfrac{\rho \cdot U \cdot D}{\mu}$ *for constant specific speed coefficient* $N_s = \dfrac{\omega \cdot \sqrt{Q}}{\Delta H^{3/4}}$ *and constant specific diameter coefficient* $D_s = \dfrac{D \cdot \Delta H^{1/4}}{\sqrt{Q}}$

where:

D - rotor diameter,
Q - volumetric flow,
ρ - density,
μ - viscosity,
U - blade speed,
ΔH - total enthalpy rise,
ω - angular velocity.

Figure 4 shows turbine efficiency as a function of Reynolds number, assuming constant specific speed and specific diameter coefficients. A chart was created by rescaling to smaller and bigger mass flow of a reference turbine to micro gasturbine. As we can see on graph the smaller mass flow the smaller efficiency of a turbine. We can improve efficiency by choosing bigger speed coefficient in the design but this leads to bigger rotational speed. Or we can use radial inflow turbine but this leads to enlargement of radial diameter of engine. In both cases the side effects of improving efficiency are undesirable. So the design of turbine to micro gasturbine is a compromise between radial diameter of engine, its rotational speed and desirable efficiency.

3. Meanline performance analysis

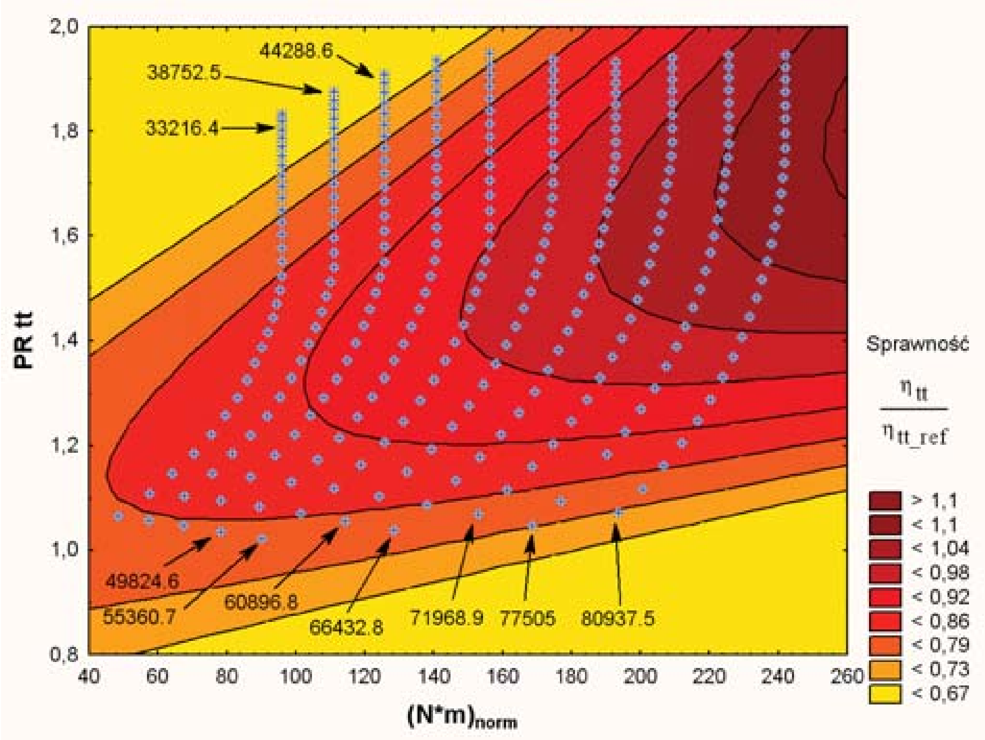

Fig. 5. Turbine to micro gasturbine performance map for different normalized rotational speed $N_{norm} = \frac{N}{\sqrt{\frac{a_{in}}{a_{std}}}}$

$$(\dot{m}_{norm} = \dot{m}\left(\frac{a_{std}}{a_{in}}\right)\left(\frac{Q_{in}}{Q_{std}}\right) \text{ and } (N * m)_{norm} = N_{norm} \cdot \dot{m}_{norm})$$

where:
$\dot{m}$ - mass flow,
N - rotational speed,
a - speed of sound,
subscripts:
norm - normalized,
std - standard,
in - inlet.

Figure 5 presents performance map of turbine to micro gasturbine (design point values: $\dot{m} = 0{,}15 kg/s$, $PR\_tt = 1{,}6$, $P = 10kW$, $N = 110000 RPM$). The chart was created with use of Concepts NREC AXIAL solver. AXIAL utilizes a reduced throughflow technique, which solves the flow using three reference streamlines (hub, RMS, and tip). Thus, the AXIAL solver can be placed between meanline and multiple streamline solvers by its capabilities. The performance of machine is predicted with use of correlations. The chosen loss model was based on Ainley/Mathieson (ARC, R&M 2974, 1951), Dunham/Came(ASME, J. of Eng. for Pwr, 1970), Kacker/Okapuu(ASME, J. of Eng. for Pwr, 1982), Moustapha/Kacker(ASME, J. of Eng. for Pwr, 1990) and Benner, Sjolander, Moustapha(ASME 95-GT-289, 1995). Some correction coefficient, based on CFD results, was used during computations. The goal of corrections was to make the results more accurate for Reynolds number approx. $\mathrm{Re} = 10^5$. Presented results are very useful during preliminary design of micro – gasturbine engine. They let to determine points of mating for compressor-turbine system and in result determine off-design performance of designed micro gasturbine engine. Although the results may differ from real turbine performance they are very good approximation and there is a lack of performance maps of machines in this scale.

4. CFD results

The CFD calculations were made with use of Concepts NREC AXCENT software. AXCENT uses hybrid multi-block structured grid full Navier-Stokes solver. The solver is appropriated for all types of turbomachines. The solver includes low-speed precoditioning technology developed by Merkle. It can deal with all flow regimes (compressible, incompressible and low speed). The governing equations used in solver are the Reynolds-Averaged full Navier-Stokes equation, Low Mach number preconditioning, and Spalart-Allmaras 1-Eqation Turbulence model. The grid is O-grid which is perfect for the area in the immediate vicinity of the blade. The outer boundary of the O-grid is the mid-passage. This kind of grid gives very accurate results at leading edge and trailing edge of the blade. Fig. 6 shows the CAD geometry of turbine with grid.

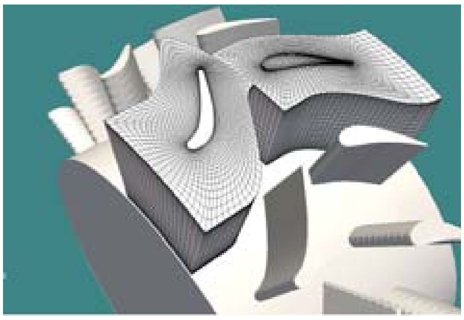

Fig. 6. Turbine geometry with O-Grid

Figure 7 shows contours of entropy at different cuts and 3D contours of largest entropy in flow. Entropy is chosen to visualize to loss in flow because it is the most suitable measure of irreversibility because its value is independent of the frame of reference and is a convected quantity. Entropy, which is created during an irreversible process, may be compared to smoke and diffuses into the surrounding fluid as it is convected downstream. We can see that the biggest losses in the flow are due to tip leakages. In turbines to micro gasturbines the ratio of tip clearance to blade height is much bigger than in normal scale turbines. The clearance should be as small as it is possible to reduce leakages. Although in case of turbines to micro gasturbine the limiting parameter is usually the accuracy of manufacturing. If we consider 1mm tip clearances on 100 mm and 10mm blade we will get 1% and 10% of blade high which is making a loot of difference.

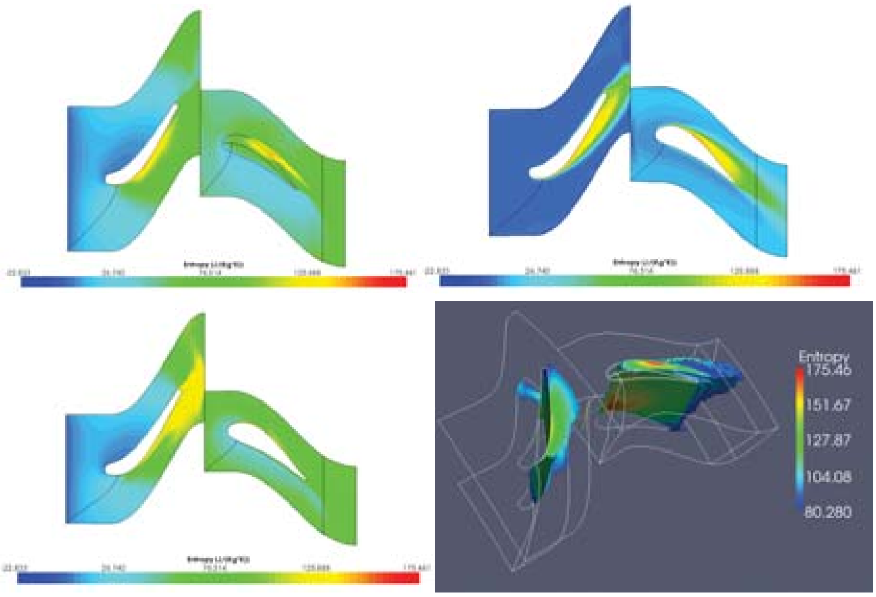

Fig. 7. 2D and 3D contours plots of Entropy (left upper corner - shroud, right upper corner - middle, left lower corner - shroud, right lower corner - 3D contours of largest entropy)

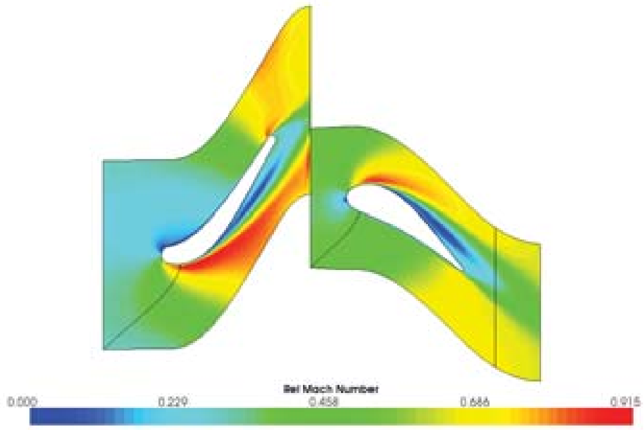

Fig. 8. Relative Mach number at middle cut of the blade

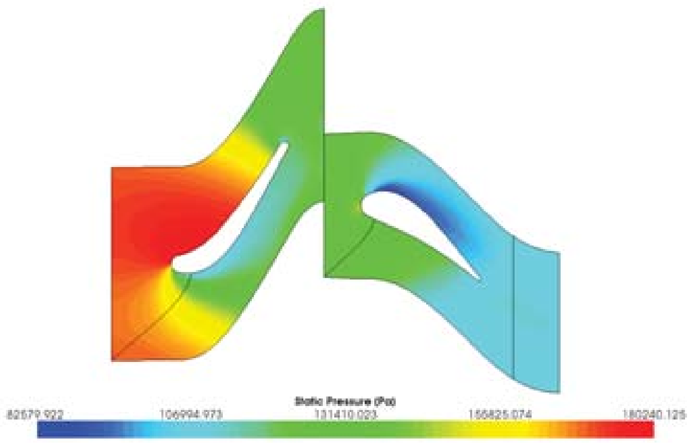

Fig. 9. Static Pressure at middle cut of the blade

Figure 8 and 9 shows two other important parameters the relative Mach number and the static pressure. The relative Mach number in turbines to micro gasturbine engines should not exceed the limit of $M_{rel} = 1$. And the static pressure at turbine outlet should be around atmospheric pressure. The main component of thrust in micro gas turbine engines is axial velocity at engine outlet. This velocity will be the biggest when the pressure after turbine will be as much close to atmospheric as it is possible.

5. Summary

Design of turbine to micro gas turbine engine differs significantly from design of commercial full scale gas turbine. First difference is initial parameters of design and the lack of existing design guidelines and test results. The small number of existing machines makes preliminary design a process with high number of unknown. That is why all early design meanline performance and CDF computations are very important. It is obvious that this methods need to be verified in laboratory tests but as a first assumption it is a powerful toll which can give us an answers to all our unknown. The Ns-Ds diagrams for Reynolds number $\mathrm{Re} > 10^6$ are well known and widely presented in many papers. However there is a lack of this data for Reynolds number of $\mathrm{Re} \approx 10^5$. The next step in development of micro gasturbine engines should be laboratory tests of very small turbines and compressors, especially mentioned Ns-Ds diagrams. During design of turbines to micro gasturbines the designer must focus on Reynolds number effects. Keeping as high as it is possible Reynolds number will give higher efficiency. Also as high as possible, specific speed coefficient will give higher efficiency, too. Another important thing during design is to focus on reducing tip clearance effects because they generate the biggest amount of losses. Next step in design of presented turbine will be laboratory tests which should give as answer about the right or wrong assumptions that were made based on presented numerical analyze.

References

[1] Meherwan, P., Boyce, *Gas Turbine Engineering Handbook*, Second Edition, 2002.
[2] Sehra, A., Shin, J., *Revolutionary propulsion system for 221 century*, NASA TM-2003-212615, 2003.
[3] Czarnecki, M., *Turbine micro engines in UAV application*, Jurnal of Aeronautica Integra, No. I, pp. 13-28, 2008.
[4] Mustapha, H., Zelesky, M., Baines, N., Japikse, D., *Axial and radial turbines*, Concepts NREC 2005.
[5] Kamps, T., *Model jet engines*, Traplet publications, 2005.
[6] Saravanamuttoo, H., *Gas turbine theory*, Pearsons education limited, 2001.
[7] *Axial Help*, Concepts NREC 2008.
[8] *AxCent Help*, Concepts NREC 2008.
[9] Denton, J. D., *Loss Mechanism in Turbomachines*, The 1993 IGTI Scholar Lecture, Journal of Turbomachinery, Vol. 115, pp. 621-656, 1993.
[10] Schlienger, J. P., *Evolution of Unsteady Secondary Flows in a Multistage Shrouded Axial Turbine*, Thesis, Diss. ETH, No. 15230, 2003.

The scientific work financed from resources on science in 2007-2011 as a research project.