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DIRECT OFF-DESIGN PERFORMANCE PREDICTION OF MICRO GAS TURBINE ENGINE FOR DISTRIBUTED POWER GENERATION

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ABSTRACT

Central-station power plants (CSPP) are the main provider of energy today. In the process of power generation at central-power stations, about 67% of primary energy is wasted. Distributed cogeneration or combined heat and power (CHP) systems are an alternative to central-station power plants. In these systems, an electrical generation system located in a residence or at a commercial site consumes natural gas to generate electricity locally and then the exhaust heat is utilized for local heating needs (in contrast to being wasted at central-stations). Microturbines offer a number of potential advantages compared to other technologies for small-scale power generation. For example, compact size and low-weight leading to reduced civil engineering costs, a small number of moving parts, lower noise and vibration, multi-fuel capabilities, low maintenance cost as well as opportunities for lower emissions. Inverter generators allow using micro-turbines of different shaft rotation speed that opens opportunities to unit optimization at off-design modes. The common approach to predict the off-design performance of gas turbine unit is the mapping of the compressor and the turbine separately and the consequent matching of common operation points. However, the above-mentioned approach might be rather inaccurate if the unit has some secondary flows. In this article an alternative approach for predicting off-design performance without using component maps is presented. Here the off-design performance is done by direct calculation of the components performances. On each off-design mode, the recalculation of the characteristic of all scheme components, including a compressor, gas turbine, combustor, recuperator and secondary flow system is performed. The different approaches for obtaining the

performance at off-design modes considering the peculiarities of the gas turbine engine are presented in this paper.

INTRODUCTION

The amount of publication and research that is done in last few decades indicate there is an increased interest in design and analysis of micro gas turbines [1, 2, 3]. To improve the efficiency of existing gas turbines, it is important to know the influence of regime parameters on unit performance. Micro gas turbines (MGT) for central-station power plants work with inverter generator that gives the opportunity to get optimal shaft rotational speed, for example, to provide maximum efficiency or change the power of MGT as per to requirements of the customer. Therefore this task is of high interest.

Previous publications devoted to the prediction of off-design performances, mass flows, pressure ratios, etc. of the gas turbine are based on different mathematical models and methods. In order to obtain the accurate results, these models require utilization of compressor and turbine maps [4, 5]. Prediction of operating mode for subsequent investigation of the transient behavior of micro gas turbine utilizing compressor and turbine map is presented in [6]. Methods for cycle component matching are presented in [7, 8].

The analytical method of determination of part load performance of combined cycle by changing the temperature at gas turbine inlet is presented in [9]. The authors of [10] proposed the methodology of the integrated simulation of a gas turbine engine by using CFD to predict performance of the gas turbine engine components.

In [11, 12] the performance evaluation at off-design modes using fuel with lower low-heating value is done. The working diapason of the compressor in [11] is determined using compressor map. To improve the part load efficiency of single

shaft and twin-shaft gas turbines in [13] the variable value of rotation speed is used.

Generating component map cost not only money but is also a time-consuming process. It must also be noted, that presence of the secondary flow (cooling, leakages) can add to the inaccuracy of the results. Some methods to minimize utilizing of component maps were proposed earlier. The approach to performing off-design point analysis that is based on using a small number of input parameters without a detailed map is presented in [14]. However, this prediction is limited and can be inaccurate.

Considered above approaches allow predicting the off design behavior of gas turbines by different ways. However, they based on a map using time-consuming process without automatic connection between processes. This paper describes a new approach of direct (without using component maps) off-design performance prediction of micro gas turbine engines for distributed power generation.

The research is presented in two parts:

- comparison of results obtained using two different approaches (presented in this paper direct performance prediction approach and approach that utilizes machine maps) without secondary flows;
- comparison of the same approaches taking into account secondary flows.

NOMENCLATURE

- a* – air;
- c* – compressor;
- CHP* – cogeneration or combined heat and power;
- cool* – cold flow;
- CSPP* – central-station power plant;
- ex* – extraction;
- f* – fuel;
- G* – mass flow rate;
- g* – gas;
- i* – inlet parameters;
- l* – leakage;
- MFR* – mass flow rate;
- MGT* – micro gas turbine engine;
- N* – unit power;
- n shaft* – shaft rotation speed;
- o* – outlet parameters;
- P* – total pressure;
- r* – recuperator;
- T* – total temperature;
- t* – turbine.

OFF DESIGN CALCULATION APPROACH REQUIREMENTS AND THE OBJECT OF INVESTIGATION

In order to implement off-design performance prediction, the approach should have flexibility and opportunity to perform design and off-design tasks. It is also desired to have the following features;

- capability to integrate commercial and proprietary software products and customize their interaction for the different problem formulation;
- capability for scripting to add user’s functionality;
- allows integration of the different software without writing additional programs;
- opportunity to perform optimization and off-design performance estimation;
- the user should be able to set optimization tasks with minimal knowledge of optimization theory;
- capability to control the process at every calculation stage;
- inbuilt or customizable libraries of fluids;

Commercially available software, AxSTREAM® was selected for performing these tasks as all desired features were available in AxSTREAM® ION software. In this software, the data transfer between each process or software is realized based on the process diagram. The simplicity of defining the relationship between different modules allows creating projects of different complexity. The presented approach to predict off design performance was tested on a micro gas turbine unit for distributed power generation coupled with inverter generator. The scheme of the components of the micro-gas turbine unit is presented in Figure 1.

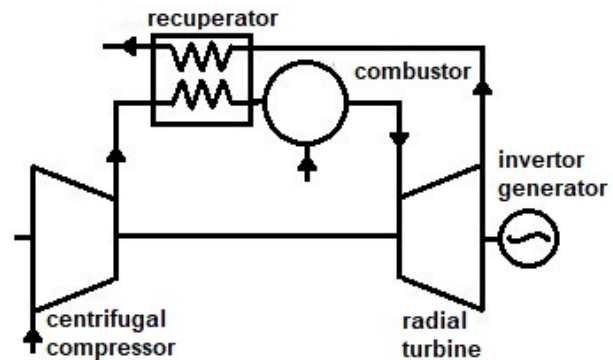


Figure 1 MGT scheme

It consists of single stage centrifugal compressor, recuperator, combustor and radial turbine. The design power output of MGT is equal 4.9 kW. Design parameters of MGT unite are presented in Table 1. The shaft rotation speed was used as an optimization parameter.

Table 1: Micro-gas turbine design parameters

Parameter	Unit	Value
$T_i c$	K	288.15
$P_i c$	kPa	101.3
$P_o c$	kPa	305.84
$P_i t$	kPa	287.765
$T_i t$	K	1263
n_{shaft}	rpm	184000
N	kW	4.9

MGT SHAFT ROTATION SPEED OPTIMIZATION WITHOUT TAKING INTO ACCOUNT SECONDARY FLOWS

In order to perform off-design performance prediction of MGT engine the flow path geometry of compressor and turbine were taken from design mode. Based on the description of the components of the MGT and optimization parameters the algorithm for off design investigation was created (Figure 2).

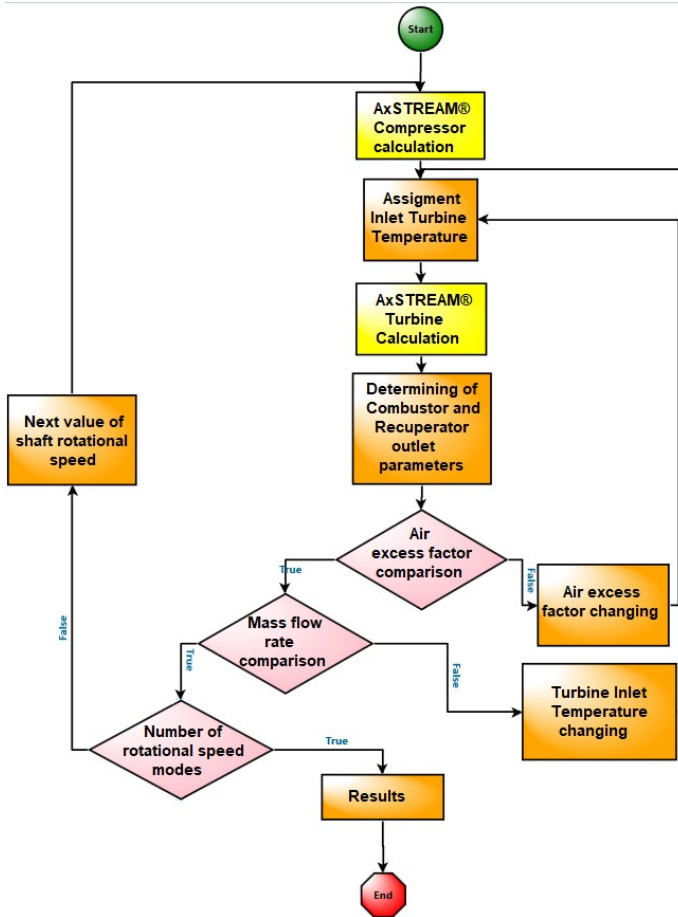


Figure 2 Calculation scheme without taking into account secondary flow

The first element of the calculation diagram is compressor analysis based on through flow solver with the problem formulation of finding mass flow rate (MFR) for given outlet total pressure. Assignment of inlet turbine temperature means determining the temperature at which mass flow enters the turbine inlet. In order to perform recuperator calculation, the turbine outlet temperature must be determined. For this, the direct calculation of turbine with the same problem formulation as compressor calculation is done. Air excess factor (AF) of working fluid and initial inlet temperature was set arbitrarily as an initial guess. Figure 2 shows the process of calculation, which is based on iterative method. All parameters that were set

arbitrarily as initial guesses were recalculated and clarified later during the calculation process. After obtaining the turbine outlet temperature, the temperature at the outlet of the recuperator at the hot outlet side is calculated:

$$Tro = e * (Tco - Tto) + Tto$$

where

Tro – outlet recuperator temperature;
 e – recuperator efficiency;
 Tto – outlet turbine temperature;
 Tco – outlet compressor temperature.

Next, the recuperator outlet enthalpy and temperature of the heated flow from compressor side using energy balance can be obtained as follows:

$$G_{cool} * (I_{o\ cool} - I_{i\ cool}) = G_{hot} * (I_{i\ hot} - I_{o\ hot})$$

$$T_{o\ cool} = f(P_{o\ cool}, I_{o\ cool})$$

where

G_{cool} – MFR of compressed air (in the case of leakage presence it equals $G_c - G_{ex}$);
 $I_{o\ cool}$ – enthalpy of air at recuperator outlet;
 $I_{i\ cool}$ – enthalpy of air at recuperator inlet;
 G_{hot} – turbine outlet MFR;
 $I_{i\ hot}$ – enthalpy of combustion products after turbine;
 $I_{o\ hot}$ – enthalpy of combustion products at recuperator outlet.

Given outlet temperature was used for combustor calculation. In order to provide turbine inlet temperature, the mass flow rate of fuel based on energy balance was determined. Here the air excess factor used is initial value and this needs to be recalculated which is done using the following equation,

$$AF = (G_{co}) / (G_f / I_0)$$

where

AF – air excess factor;
 G_{co} – outlet compressor mass flow;
 G_f – fuel mass flow;
 I_0 – stoichiometric air fuel ratio.

The refined value of AF is used in next iteration. The process of clarification of AF and fuel MFR continues until the difference between AF in previous and current iterations in relative values becomes less than 0.001.

The turbine inlet temperature is calculated iteratively utilizing “Bisection method”. Off-design turbine calculation is converged when values of MFR from combustor side and turbine inlet MFR are equal with discrepancy less than 0.001 in relative values. This requirement follows from the condition of MFR balance

$$G_g = G_a - G_l + G_f$$

where

G_g – gas mass flow rate at turbine inlet;

G_a – inlet compressor mass flow;
 G_l – leakage mass flow;

In the presented algorithm the calculation of recuperator and combustor chamber is performed using scripting module that allows setting user functions. The MGT scheme was created in commercial software AxCYCLE™ [15] and presented in Figure 3.

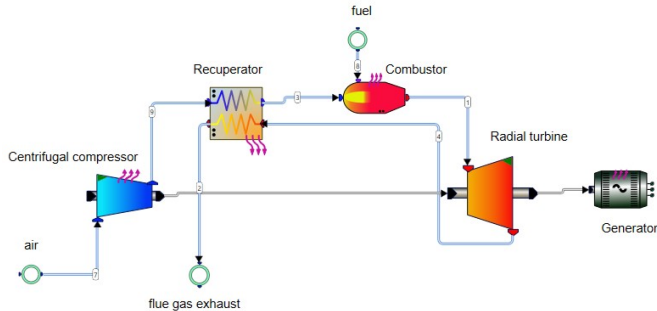


Figure 3 MGT scheme for calculations using maps

The results obtained using the direct calculation method were compared with the map based method. The distributions of compressor and turbine power and efficiency as a functions of shaft rotation speed are presented in Figure 4 & 5.

For the turbine, the shaft rotational speed rise leads to efficiency improvement and reaches the peak at about 185.5 krpm and then starts decreasing. Although we observed the increase in turbine efficiency, the overall efficiency of GTU was decreasing.

Compressor efficiency decreasing with shaft rotational speed rise is not significant in all research range. The decreasing of MGT net power is also observed with the shaft rotation speed rise.

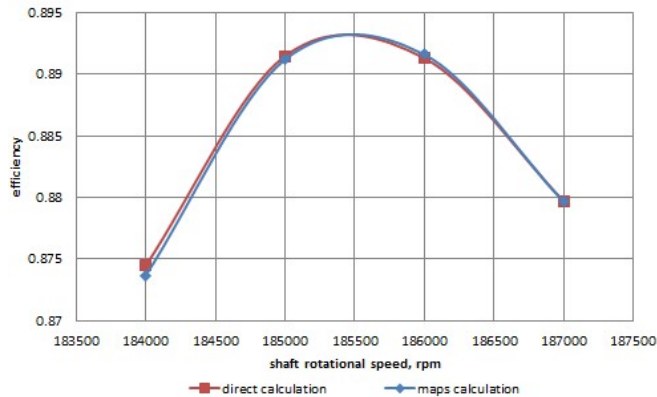


Figure 4 Turbine efficiency vs shaft rotational speed

Figure 6 shows the difference of power between the above mentioned approaches (maps and direct calculation) for different shaft rotational speed values. The maximal difference of power does not exceed 0.65% in relative values. Compressor and turbine efficiency error – 0.075%. By analyzing the

obtained results it can be concluded that the convergence of common approach (using maps) and an alternative approach (direct calculation) is insignificant for the case when secondary flows are not observed.

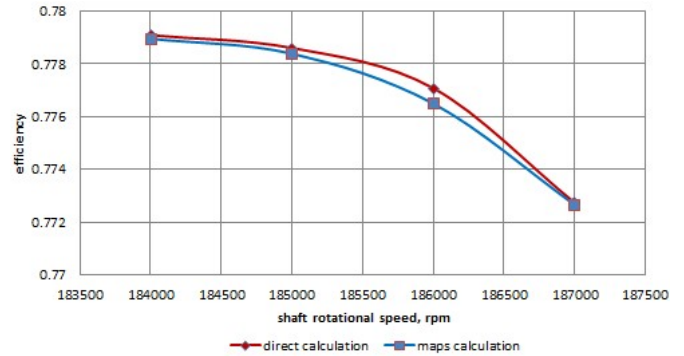


Figure 5 Compressor efficiency vs shaft rotational speed

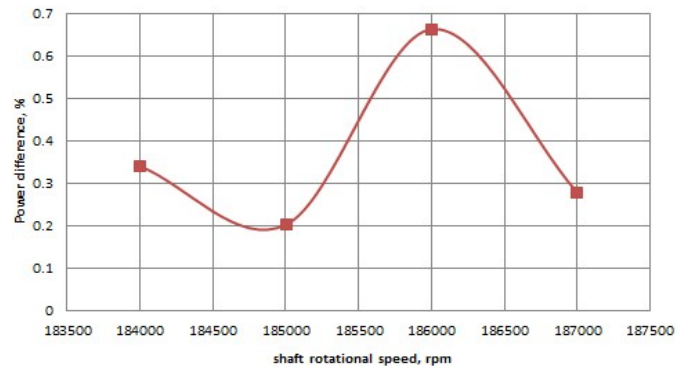


Figure 6 MGT net efficiency vs shaft rotational speed

MGT SHAFT ROTATION SPEED OPTIMIZATION TAKING INTO ACCOUNT SECONDARY FLOWS

Secondary flows at MGT flow path appear through clearances between stator and rotor elements of the flow path. This leakage moves through the area of bearing and then is being injected in radial turbine flow path. The influence of the secondary flow on thermodynamic parameters of main flow can be significant.

The conventional approach (maps matching) predicts off-design parameters based on maps, which are created with the absence of secondary flows or for some specific value of secondary flow. It is obvious that the results obtained by this approach might be inaccurate.

The aim of present research part is to determine the influence the secondary flows which have less energy, on efficiency and power of turbine. This part of paper, in contrast with previous part, is devoted to a comparison of direct calculation results with results generated using component maps considering the effect of secondary flows.

In the real machine, the scheme of secondary flow leakage is complex and includes a lot of elements. For presented research, the simplified scheme that accounts for leakage

moving along shaft through bearing was used (Figure 7). Secondary flow system was calculated in commercially available 1D flow and heat transfer analysis software AxSTREAM NET™ [16]. Following boundary conditions were used for calculations:

- total pressure at secondary flow system inlet;
- the total temperature at secondary flow system inlet;
- static pressure at secondary flow system outlet.

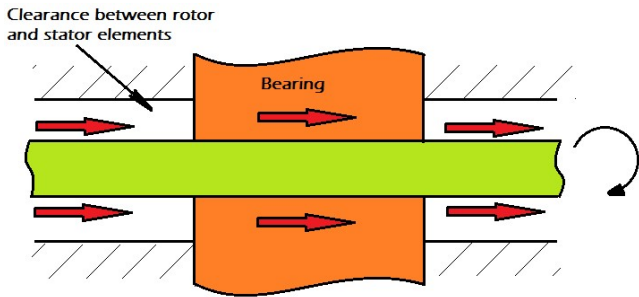


Figure 7 The scheme of secondary flow system

The clearance between stator and rotor elements and length of the channel were assumed to be 0.2 mm and 50 mm respectively. To take into account secondary flows in “maps” approach, the thermodynamic scheme of MGT was changed and is presented in Figure 9.

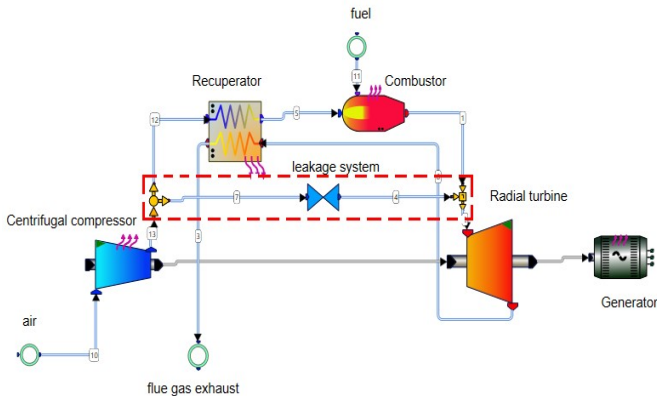


Figure 9 MGT scheme with secondary flow accounting for calculations using maps

Mass flow and outlet total enthalpy were taken from secondary flow system calculation and used in turbine calculation.

Diagram of calculation for the case of secondary flow presence is shown in Figure 8. The only difference here from the diagram presented in Figure 2 is in the module for secondary flows system calculation and comparison. The resistance coefficient for every calculation was defined as 0.05.

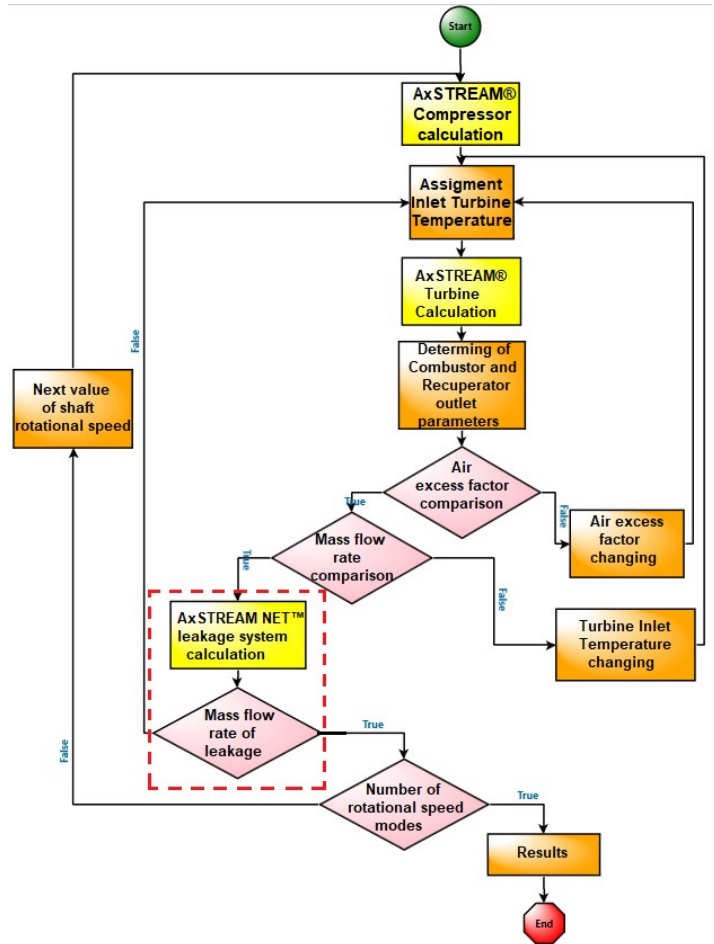


Figure 8 Calculation scheme with taking into account secondary flow

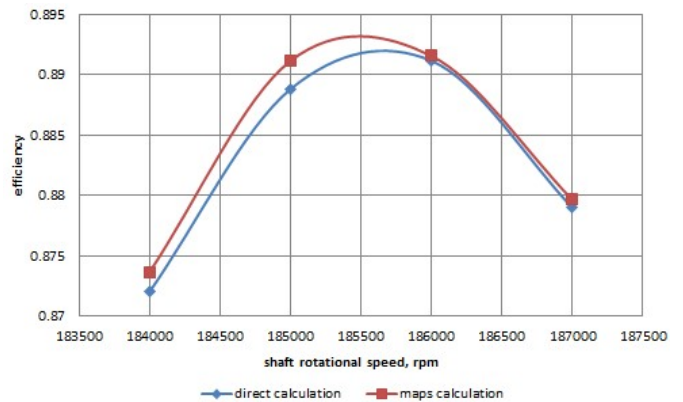


Figure 10 Turbine efficiency vs shaft rotational speed

Figure 10 presents the distribution of turbine efficiency versus shaft rotational speed. Accounting for secondary flows leads to decreasing of turbine efficiency. In this case, the maximum efficiency point corresponds to higher values of rotational speeds. Maps approach also shows the decreasing of efficiency on different shaft rotational speed but not very significant.

Net power distribution is shown in Figure 11. The inaccuracy of net power to both approaches is equal 1.05% in relative values that is about 1.5 % higher from the case when secondary flows are not accounted. The higher value of secondary flow corresponds to larger decrease of net power.

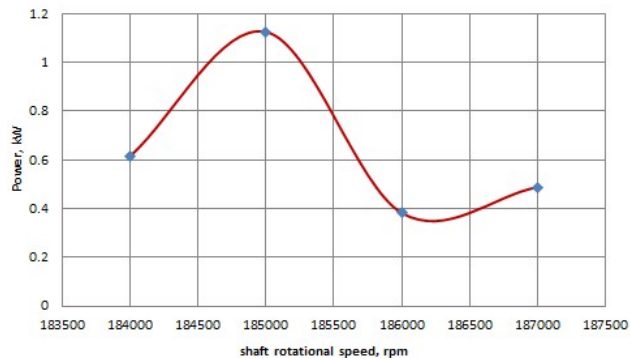


Figure 11 Turbine power vs shaft rotational speed

Also, Figure 11 shows that the values of net power for each value of shaft rotational speed is slightly higher than presented in Figure 6. Such increasing of net power is caused by specific work rise. Due to the fewer values of compressed air which is heated in the combustion chamber, the outlet combustor temperature becomes higher.

CONCLUSIONS

1. The alternative approach of MGT performance prediction based on the iterative process of direct calculation of each unit component is proposed.
2. It is shown that turbine efficiency can be improved by increasing the shaft rotational speed and the drops of compressor efficiency is insignificant.
3. Calculated MGT efficiencies using “maps” approach and the one presented in the paper were compared in two different task formulation: with leakages and without leakages.
4. Net power is significantly decreased by increasing shaft rotational speed. Therefore, during the optimization task, the summary influence of different objectives should be accounted.
5. Including the secondary flows into the calculation leads to decreasing of net power and efficiency of the turbine. The direct calculation of MGT shows a more significant drop of integral parameters.
6. The influence of secondary flows for given scheme of leakage and its resistance is not significant. Most likely, in the case of bigger leakage, decreasing of integral characteristic will be more significant.

REFERENCES

- [1]. Shan X. C., Qide Zhang and Yaofeng Sun, Zhenfeng Wang, 2006. “Design, fabrication and characterization of an air-driven micro turbine device”, International MEMS

- Conference, Journal of Physics: Conference Series 34, pp. 316–321.
- [2]. Thanate Sangsawangmatum, Udomkiat Nontakaew, 2017. “Design of Radial Inflow Turbine for 30 kW Microturbine”, MATEC Web of Conferences 95, 06005 (2017).
- [3]. Peirs, J., Reynaerts, D., Verplaetsen, F., Norman, F. and Lefever, S., 2003, “Development of a Micro Gas Turbine for Electric Power Generation”, proc. of MME 2003, pp. 215-218.
- [4]. Juha Kaikko, 1998. “Performance prediction of gas turbines by solving a system of non-linear equations”, Thesis for the degree of Doctor of Technology, University of Technology, Lappeenranta, Finland on the 6th of March, 100 p.
- [5]. Adam HEAD, 2011. “Conceptual Design and Simulation of a Microturbine; An Electric Car Range Extender Application”, Bachelor of Engineering Thesis, The University Of Queensland, 2011, 186p.
- [6]. Tuccillo R., 2005. “Performance and Transient Behaviour of MGT Based Energy Systems” VKI/LS on “Micro Gas Turbines,” RTO-MP-AVT-131, 56 p.
- [7]. Jeremy Martinjako, 2014. “Low Cost Gas Turbine Off-Design Prediction Technique”, MS Thesis, Arizona State University, May 2014, 95p.
- [8]. Saravanamuttoo, H. I. H., Rogers G. F. C., and Cohen, H., 2001. “Gas Turbine Theory”, 5th edition, Pearsons Education Limited, 2001.
- [9]. Cheng Yang, Zhifeng Huang, Zeliang Yang, Xiaoqian Ma, 2015. “Analytical Off-design Characteristics of Gas Turbine-Based CCHP System”, Energy Procedia 75 (2015) 1126 – 1131.
- [10]. Alexander Krivcov, Leonid Shabliy, Leonid Shabliy, 2014. “Account the mutual influence of the simulation components of GTE”, Proc. of the ASME 2014 gas turbine India conference.
- [11]. E. B. Silvaa , C. Bringhenti a, M. Assato b, and R. C. Limac, 2013. “Gas Turbine Performance Analysis Operating With Low Heating Value Fuels”, Thermal Engineering, Vol. 12 No. 2 • December 2013 • p.08-15.
- [12]. Aiguo LIU, Yiwu WENG , 2009. “Effects of Lower Heat Value Fuel on the Operations of Micro-Gas Turbine”, Energy and Power Engineering, 2009.
- [13]. Dan Li, Roger A., Dougal, Eshwarprasad Thirunavukarasu, 2013. “Variable Speed Operation of Turbogenerators to Improve Part-load Efficiency”, Electric Ship Technologies Symposium (ESTS), 2013 IEEE.
- [14]. Walsh, P. P., and Fletcher, P., 1998. “Gas Turbine Performance”, 1st edition, Blackwell Science Ltd, 1998.
- [15]. SoftInWay Inc, 2014, AxCYCLE user documentation.
- [16]. SoftInWay Inc, 2016, AxSTREAM NET user documentation