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Research Article

Prospects of centrifugal reaction turbines for microturbomachinery applications

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ABSTRACT
High capital cost of plant is one of the major factors impeding further development of microturbines for waste heat and pressure utilization. Thus, the reduction of cost is seen as vital for making the market of microturbines grow. There exist many applications where the overall plant efficiency is not high because of excessive waste sources and low power demand. Inexpensive and technologically simple centrifugal reaction turbine can be used in these applications.
Due to lack of materials about the above turbines, the present study seems to be relevant. The current paper provides a brief overview of the patents and papers on centrifugal reaction turbines, the CFD simulation method and its preliminary approbation. It was proved that the proposed simulation approach is in line with experimental results. The discussion chapter describes flow behavior and estimated efficiency of turbine and points out the directions of further development as well.

KEYWORDS: UNCONVENTIONAL TURBOMACHINERY, CENTRIFUGAL REACTION TURBINES, TURBOMACHINERY CFD3

NOMENCLATURE

a_{th}	nozzle throat (mm)	T	temperature (K)
D_m	mean diameter (mm)	u/C_0	stage specific speed
G	mass flow rate (kg/s)	Z	number of passages and nozzles
H_0	isentropic enthalpy drop (kJ/kg)	β_2^*	impeller nozzle outlet angle in relative frame (deg.)
l_{th}	nozzle height (mm)	ε	partial admission rate
M_R	rotor torque (N·m)	$\eta_{t\text{-}s}$	total-to-static efficiency
n	rotational speed (rev/min)	π	pressure ratio
p	pressure (MPa)		

Subscript
* total parameter

INTRODUCTION

Traditionally development of turbomachinery requires combining more than one complex systems to increase efficiency, performance and durability. Such approach is appropriate and reasonable for the systems that burn fossil fuels.

The constant challenge when dealing with unconventional- and micro turbomachinery is to solve the problem of economic feasibility. According to Bruce Hedman (2008), modern waste and pressure recovery systems become economically viable only when purchase prices include some incentives. The issue is particularly relevant for those auxiliary power systems and other power supply systems where electric demand is low (or unstable) as compared with potential of waste energy utilization. Therefore, now, and, at least, until some significant changes in the worldwide market of waste energy utilization come, the concept of reductionism seems to be a way of developing microturbomachinery.

Centrifugal bladeless reaction turbine instead of conventional turbine with nozzle vane and impeller can help in reducing cost and simplifying the design of a microturbine plant. The advantages of such centrifugal bladeless turbine are: high erosion resistance and simplicity in design and manufacturing. Another crucial feature is that centrifugal reaction turbine keeps almost constant efficiency in a wide range of pressure ratios because of its specific flow kinematics. This is especially important for pressure letdown turboexpanders. There are dozens of patents where centrifugal reaction turbines are described, dated from the early 20th century up to nowadays. Such type of turbines is often used for two-phase flows of wet steam in hydrothermal applications and for dry gases as well.

Palmer A. House (1982) described a ‘velocity pump reaction turbine’ designed for more efficient recovery of low- and medium hydrothermal energy and another waste heat sources. This invention consists of two concentric rotors one inside the other as shown in Figure 1. Inner rotor – a ‘velocity pump’ – is used to accelerate water flow up to circumferential velocity of outer rotor, that is a centrifugal turbine. Hot water enters the inner rotor in its center and moves radially outwards, accelerating in the nozzles. Then it enters the outer rotor and, moving radially towards the outlet, approaches a throat. Downstream the throat a mixture of water and steam appears due to pressure drop. The two-phase working fluid leaves the nozzle with high velocity generating reaction force and, consequently, momentum at the shaft. Unfortunately, only relative engine efficiency for different relations of inner and outer rotor diameters is presented by the author while the total-to-static efficiency is unknown.

Abhijit Date et. al. (2015) published the results of the rotating two-phase reaction turbine study, shown in figure 2. The isentropic efficiency was 17% within supply water temperature of 97°C and reached 25% due to an increase in water temperature of up to 113°C. The authors highlighted that supply water mass flow tends to increase within the rotational speed increase, which is a consequence resulting from centrifugal turbine behavior: working fluid pressure increases due to centrifugal force.

Jerry D. Griffith (1968) invented a small volumetric flow reaction turbine capable of operating with high temperature and high pressure fluids. An impeller contains a finite number of channels where the flow accelerates and then leaves the impeller. The turbine cross-sections are in figure 3. In accordance with the author’s description, the invention mostly meets the conditions of low mass flow rate, when partial admission is usually required. No data is provided for evaluating the machine efficiency.

Takeo S. Saitoh (2014) proposed ‘centrifugal reverse flow disk turbine’ for power generation. The turbine consists of an impeller with curvature channels and stator nozzles, which, in accordance with the author’s statement, are designed to provide extra power. The sketch of such turbine is shown in figure 4.

Victor Rassokhin (1991) from Leningrad Politechnic Institute studied centrifugal turbines both analytically and experimentally. More than 15 machines were designed and studied by means of experiments. As a result, plots of total-to-static efficiency vs. stage specific speed were obtained, shown in figure 5. It’s easy to observe that maximum values of the total-to-static efficiency are shifted by 40%.

Even though the authors of patents proposed many different alternate turbines, the design techniques and testing results are almost missing. It is challenging to evaluate efficiency and behavior of gas and steam centrifugal reaction turbineswith no publications about them. Therefore, the data obtained via numerical simulation and experimental testing contributed greatly to development of centrifugal turbines. The paper provides the results of numerical simulation of 30kWt letdown centrifugal reaction turbine operating on natural gas, including simulation method, and experimental apparatus for the ongoing turbine experimental study.

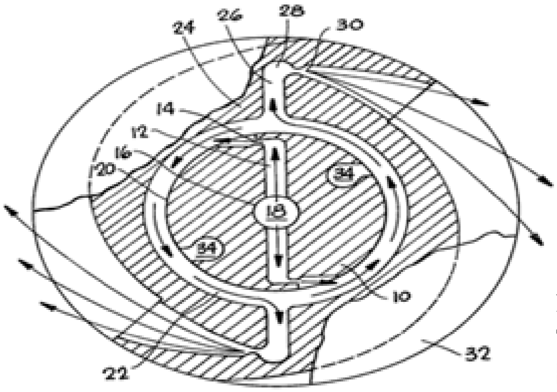


Figure 1: Velocity pump reaction turbine by Palmer A. House (1982)

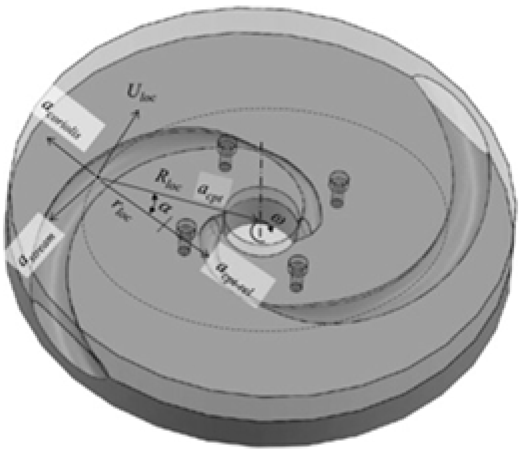


Figure 2: Two-phase reaction turbine by Abhijit Date (2015)

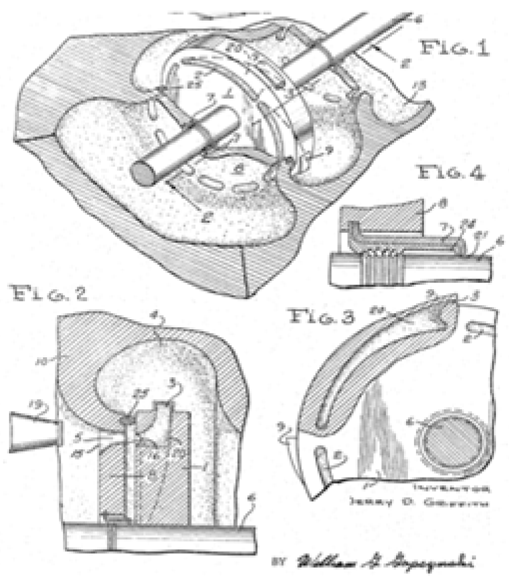


Figure 3: Small volumetric flow reaction turbine by Jerry D. Griffith (1968)

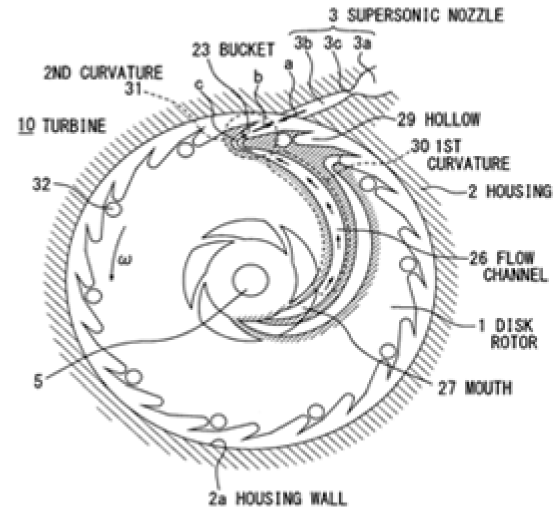


Figure 4: Centrifugal reverse flow disk turbine by Takeo S. Saitoh (2014)

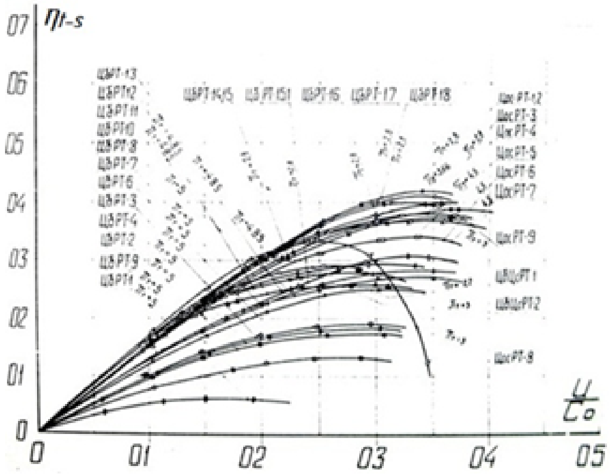


Figure 5. Total-to-static efficiency curves published by Rassokhin (1979)

METHODOLOGICAL FRAMEWORK

Turbine Design

Turbine consists of an impeller with 4 passages and a hood as shown in figure 6. The number of impeller passages is decreased as consistent with the concept of a low-cost turbine to reduce the production costs. Working fluid enters the impeller at the center, turns in the radial direction getting into the passages and moves radially outwards. The passage is shaped so as to meet the conditions of the working fluid compression due to centrifugal forces. Each passage terminates in a nozzle where the flow accelerates reaching the Mach number of not less than 1.9. It should be highlighted that the flow leaves the impeller in axial direction in order to provide both maximum rotor torque and minimum radial dimension of the machine.

The exhaust hood is designed to be straightforward for manufacturing. It consists of a hollow cylindrical bottom part where the impeller is located and a prismatic top part (collector) with a simple flow separator. As shown below, such a simple design gives flow non-uniformity and pulsations. When entering the hood, the working fluid runs almost tangentially due to small flow angle in the absolute motion. After having made a complete circle in the hood, the working fluid leaves it at the outlet in the top.

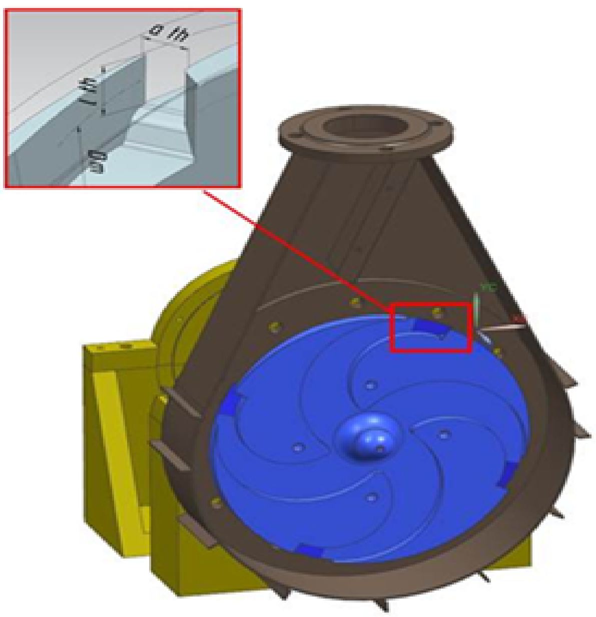
The details of the turbine geometry are presented in table 1. The turbine and exhaust hood layout is presented in figure 6.

Table 1: Design parameters of centrifugal stage

Parameter	Dimensions	Values
Throat shape	-	Rectangular
D_m	mm	396
ε	-	0.42
a_{th}	mm	10.00
l_{th}	mm	12.10
β_2*	deg.	5
Z	-	4

Table 2: Boundary conditions and operation parameters

Parameter	Dimensions	Methane-operated stage
p_0^*	MPa	1.90
T_0^*	K	288.0
p_2	MPa	0.60
π	-	3.167
H_0	kJ/kg	141.8
u/C_0	-	0.114; 0.836; 1.141
n	rev/min	3000; 22000; 30000



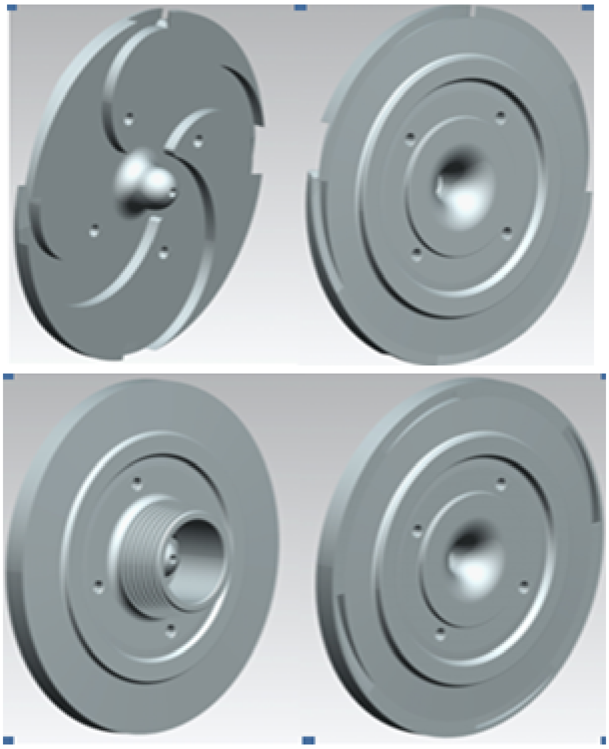


Figure 6. Turbine and exhaust hood layout

Numerical Simulation Method

ANSYS CFX commercial code was used to perform a numerical simulation. While the whole enthalpy drop takes place in the impeller nozzle, the outlet velocity is high and the outlet Mach number varies from 1.5 to 2.8. The discharge flow interacts with the flow in the exhaust duct, generating strong flow perturbations, which causes significant pressure losses and, as it was observed via transient simulation, mass flow unsteadiness. It is well known that steady state simulations tend to underpredict the losses in a transonic flow due to interaction with the downstream components, as shown by Roger L. Davis (2004).

To eliminate influence of turbine outlet conditions on the flow into the exhaust duct a direct simulation method needs to be applied, i.e. rotating impeller and static hood should be pared with using some rotor-stator interface. Whereas the exhaust hood is not axially symmetrical and discharge flow passes through a complete circle in the hood, full circumferential simulation is vital to properly resolve the flow behavior and disk friction losses. In the investigations, carried out by Gardzilewicz (2014), the advantages of the direct simulation of turbine and exhaust hood as opposed to the coupled simulations are clearly shown.

The study used a high-Reynolds version of the k-ω SST turbulence model. The grid independence study was not performed, thus the mesh was deliberately oversized to ensure it's fine enough. The total number of nodes in the grid is about 8 million. Maximum values of y^+ parameter were 60 for impeller parts and 120 for stator (hood) parts. Efficiency of stage was evaluated on the basis of impeller torque, as shown in equation (1). In order to resolve the behaviour of flow into the exhaust hood a transient rotor-stator simulation was carried out by performing URANS simulations. The time step was set so as to provide at least 10 time steps per one rotor passing period, while the rotor passing period is equal to the time of ¼ impeller revolution.

Monitoring of RMS residuals, imbalances and turbine efficiency and power output were used to control the convergence of a steady-state solution process. The convergence criteria for the steady-state solutions were:

– drop of RMS residuals more than 10^2 at each time step;
– imbalances less than 0.5%;
– periodic oscillations of the parameters at monitor points.

The chosen initial conditions and operation parameters are presented in table 2.

The total-to-static stage efficiency was calculated using the following equation:

$$\eta_{t-s} = \frac{M_R \cdot \pi \cdot n}{30 \cdot G \cdot H_0}$$

The $\eta_{t\text{-}s} = f\,(u/C_0)$ curve was obtained using linear character of the $M_R = f\,(u/C_0)$ function, as described by Kirillov (1972) and Traupel (1977).

DISCUSSION OF RESULTS

Flow structure

The first point of interest is an interaction between supersonic outlet flow and exhaust hood flow. Due to high pressure ratio the flow leaving impeller nozzles reaches the Mach number of up to 1.92. The maximum velocity occurs in the oblique cut downstream the nozzle throat. As shown in figure 7, while leaving the said oblique cut the flow meets working fluid that moves in a circle into the exhaust hood. As a result of a strong interaction oblique pressure shock appears, decreasing the flow velocity. That situation persists along the whole nozzle cut generating 3 pressure shock series.

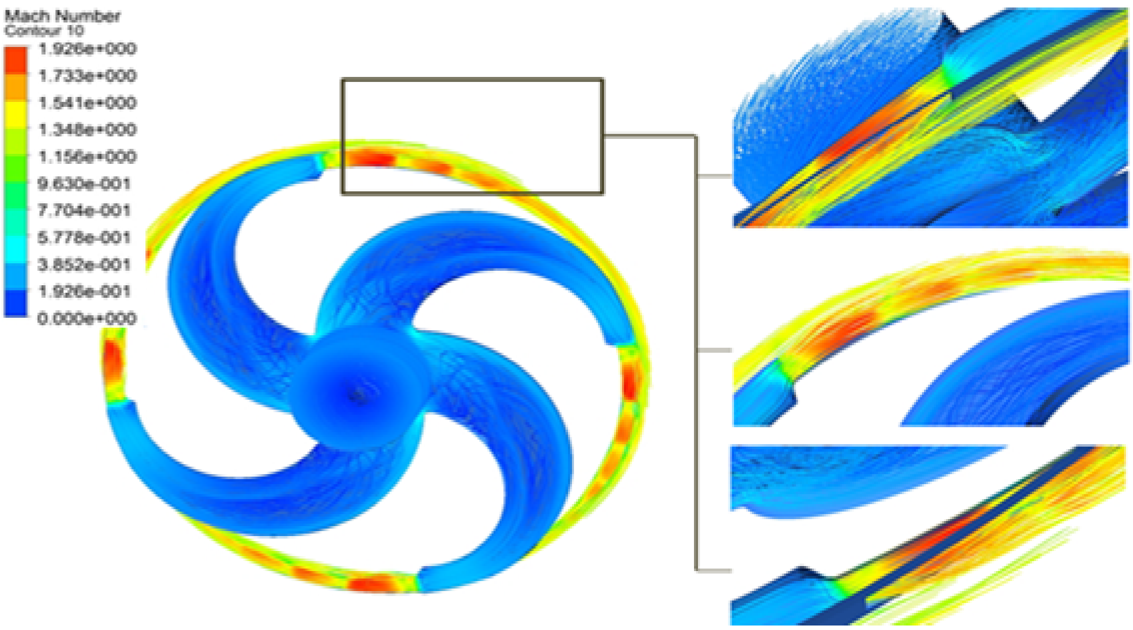


Figure 7. High-speed impeller outlet flow

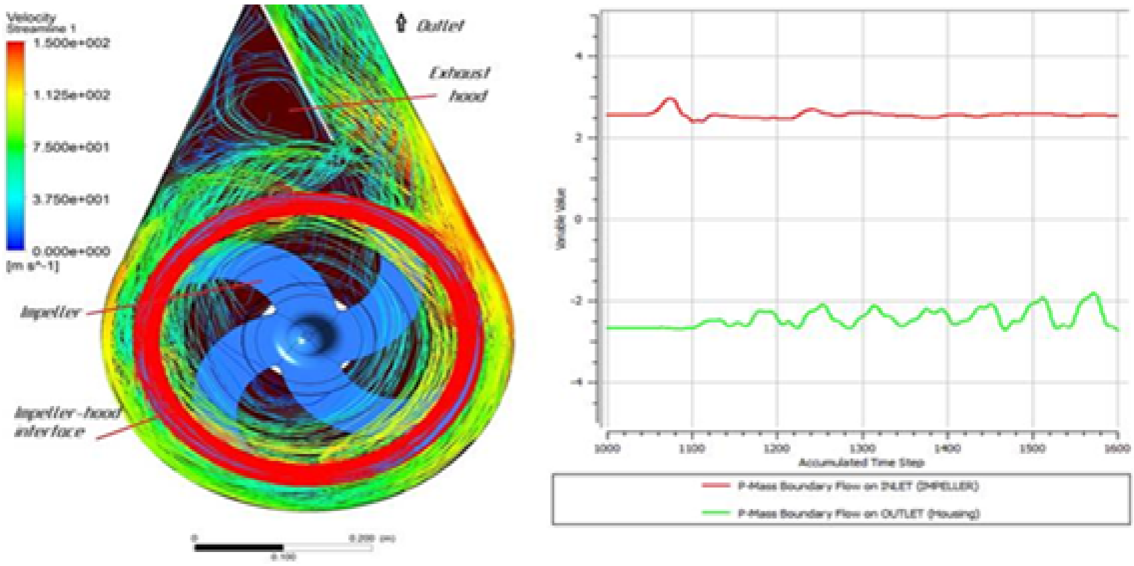


Figure 8. Flow into exhaust duct and outlet mass flow oscillations

The behavior of flow into exhaust hood contributes significantly to the machine efficiency and operation stability. Figure 8 shows scale-wise structures of vortex in both cylindrical part and collector. The most intense vortex interaction occurs at the border of cylindrical part and collector where the flow is divided to move circumferentially and towards the outlet. Strong velocity gradient is a reason for significant pressure loss in this area. A volute chamber is required to capture discharge flow with minimum flow disturbances but it is not applicable within the concept of a low-cost turbine. Considering the mass flow behavior at the outlet shown in figure 8, it is easy to observe the flow oscillations. It should be highlighted that only the transient simulation resolves this oscillation while the outlet flow is completely stable in the steady-state case. Since mass flow oscillation period is equal to ¼ of the impeller rotational period, it could be reasonably stated that despite damping effect of exhaust hood the periodic flow disturbances generated by the high speed impeller outlet velocity still exist at the outlet. They may affect negatively the outlet pressure control valve making it follow these disturbances. Since oscillating valve is operated at the same frequency as mass flow oscillations and explicit aerodynamic connection exists, some steps are required to stabilize the flow. One possible way is to use a diffuser in order to reduce discharge velocity, to provide the flow stability and to increase the stage enthalpy drop.

Turbine Performance

Turbine torque and total-to-static efficiency curves are presented in figure 9. Two curves are presented for the total-to-static efficiency: the first one, $\eta_{t\text{-}s\ teor}$, describes the turbine behavior without taking into account disk friction losses. The second one relates to efficiency with disk friction losses accounted for. As shown in figure 9, efficiency curves begin to deviate within the specific ratio numbers more than 0.5, which is an equivalent of about 13000 rpm for the simulated pressure ratio of 3.167. The deviation tends to increase with a rotational speed rising as a result of friction losses surge. The total-to-static efficiency exceeds 50% and reaches 0.51 in the specific speed ratio of 0.563 (about 14 500 rpm). Such a rotational speed is practically achievable in terms of strength and reliability.

For providing relevant data the simulation approach needs to be accurate. Since no direct comparison between test rig data CFD simulation was made, only tendencies may be evaluated. The results obtained via numerical simulation generally correspond to the experimental ones provided in figure 5 taking into account some inaccuracy of CFD simulation method. There are discrepancies observed for the value of efficiency (overprediction via CFD at about 7%) and for the location of maximum efficiency vs. specific velocity ratio (about 0.35 – 0.45 according to experimental testing and 0.563 via CFD).

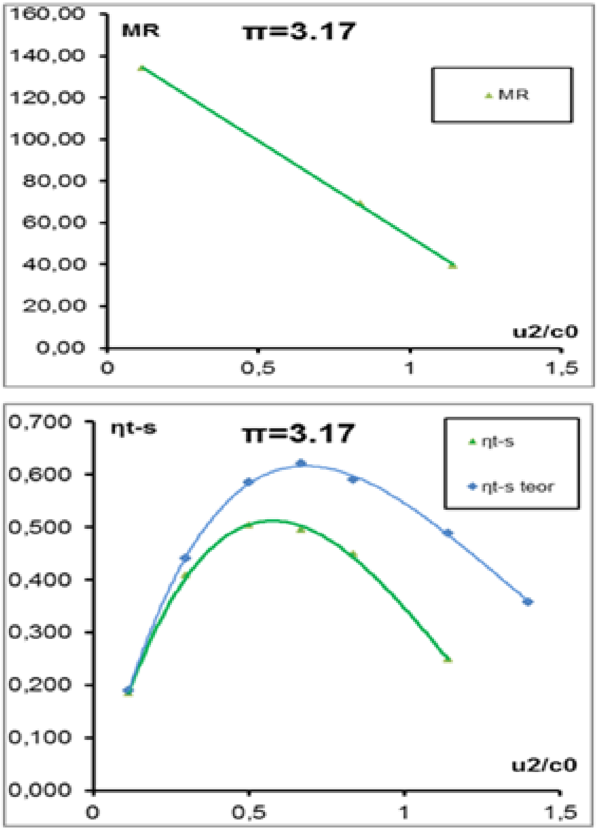


Figure 9: Total-to-static efficiency and turbine torque

CONCLUSIONS

The objective of the present research was to show the possibility of using the centrifugal reaction turbines for microturbomachinery applications. A brief overview of the related patents and publications was made.

The methodology of CFD simulation was tested and proved to be acceptable based on experimental studies of similar turbines. Intense disturbances of oscillating flow into exhaust duct and outlet mass flow oscillations were observed. On this account some activities aimed at stabilizing the flow are required, for instance, a diffuser seems to be helpful for enhancing the flow behavior and turbine operation.

It was shown that possible total-to-static efficiency exceeds 50% within feasible 14 500rpm rotational speed.

It could be summarized that efficiency of turbine is quite low but acceptable in case of waste and pressure utilization applications. Low cost and technological simplicity are advantages of such turbines contrary to their efficiency. Nonetheless, further investigations are still required in terms of using a diffuser, stabilization of flow and further improvement of efficiency.

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