УДК 621.165

L. Jedrzejewski, P. Lampart

Institute of Fluid-Flow Machinery, Polish Academy of Sciences, Gdansk, Poland

TESLA FRICTION-TYPE MICRO TURBINE FOR SMALL-SCALE COGENERATION

Tesla turbine is rarely used in professional power installations due to its specific work characteristics and relatively low internal efficiency. However, it has some potential to be applied in micro-power plants operating in distributed co-generation systems, also working in an organic Rankine cycle. This paper presents results from numerical calculations of flow in three models of Tesla disk turbines assuming nitrogen as a working fluid. It is shown that Tesla turbine models can yield reasonable flow efficiencies of 30-40% and therefore can be applied in micro-power installations.

Keywords: *Tesla micro turbine, organic Rankine cycle, computational fluid dynamics.*

Introduction

Tesla turbine is a very specific device. First time it was presented in 1913 by a Serbian engineer and inventor Nicola Tesla [1]. This unusual bladeless turbine, also known as a friction turbine, makes use of viscous effects occurring in the boundary layer flow. Opposite to classical bladed turbines, where viscous effects in flow are undesirable as a source of efficiency loss, these effects enable the rotational move of the rotor. The rotor consists of up to a few dozens of thin disks locked on a shaft perpendicular to its axis of revolution. In theory, the disks should be as thin as possible. The distances, or gaps, between the disks should be very small. According to [2], the highest value of efficiency appears when they are approximately equal to the double boundary layer thickness. Therefore, the gaps between the disks should depend on the occurring flow conditions and physical properties of the working fluid. An example of the multidisk rotor construction of a Tesla turbine found in the patent documentation [3] is shown in Fig. 1.

The supply of a Tesla turbine can be accomplished by one or several nozzles, discretely located along the circumference. The nozzles are tilted under a certain angle to the disk tangent. Working fluid flows between the disks from the outer to inner radius through spiral paths and transfers kinetic energy to the rotating disks. The medium flows out in the axial direction through a number of holes in the disks located near the turbine shaft.

There are many technical benefits and a quite big potential for the construction of Tesla microturbines, especially those working in an Organic Rankine Cycle, Fig. 2. Such a solution can be competitive to classical bladed turbines due to its operating at low parameters, relatively small pressure drop and low mass flow rate of the working fluid. There are much more advantages [4]:

- relatively high cycle and turbine efficiency (in theory),
- low mechanical stress in turbine, due to its small size and low peripheral speeds,
- no erosion of blades, due to the absence of moisture (if a working fluid other than water is used),
- long lifetime,
- simple start-stop operation,
- quiet operation,
- low level of maintenance requirements,
- good part load performance,
- no axial loads,
- easy to repair,
- cheaper than bladed turbines, due to the simple construction of the rotor.

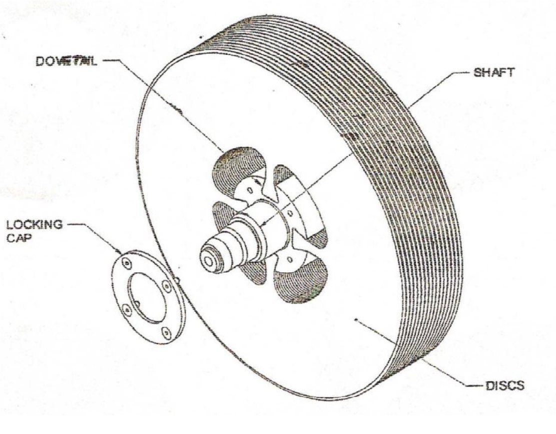

Fig. 1. Rotor of a multidisc Tesla bladeless turbine [3]

There are several design tips and instructions in several publications, books and patent documentations [2, 3, 5, 6] but many of them are contradictory. It is important to establish relationships between the turbine efficiency and working parameters such as:

– distance between the turbine disks,
– diameter and state of the disk surface,
– number and geometry of inlet nozzles,
– rotational speed of the rotor,
– boundary conditions: medium inlet pressure, temperature, velocity and angle,
– constructional materials (composites, ceramic materials, bronzes, aluminium alloys),
– kind of medium flowing through the micro-turbine (air, biogas, organic agents, exhaust gases, multi-phase media).

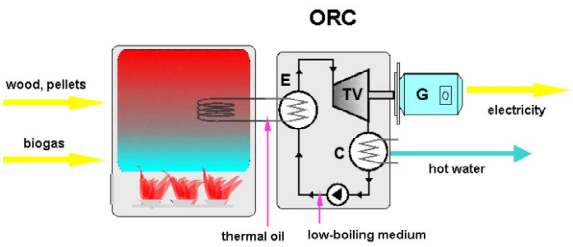

Fig. 2. A cogeneration unit with an ORC system:
E - evaporator, TV - steam turbine,
C - condenser, G - generator

Simulations of several disk turbine models were performed in [7]. Those models were designed to operate on low boiling medium SolkathermSES36 in an organic Rankine cycle system. In the present paper Tesla turbine models running on nitrogen as a working fluid are calculated. Nitrogen was chosen as a working fluid to compare CFD results with experimental data from a test stand where preliminary experiments will be performed on nitrogen.

1. Preparing the geometry

Calculation domains for the investigated models of Tesla microturbines were prepared in software Gambit. The investigations were made for the disk diameter 120 mm. The distance between the disks was established as 0.2 mm.The models are equipped with two, four or six supply nozzles discretely located along the circumference. The inlet angle relative to the disk tangent was assumed as 10°. In each configuration the nozzle geometry was the same to get similar expansion conditions (pressure drop along the nozzles and velocity at the outlet from the nozzles).

The group of models has several simplifications in the geometry. The computational domain contains one single interdisk space, one disk wall on each side and supply nozzles with symmetry plains in central plane perpendicular to the axis of rotation. The outlet area is realised here in a way to allow the medium outflow from the interdisk gaps along the entire circumference, in the radial direction. The tip clearance is also neglected.The calculations are carried out in a fixed (motionless) reference frame and the turbine disk surface is rotating. Periodic conditions are used to calculate only part of the disk-to-disk space, depending on the number of supplying nozzles, as shown in Fig. 3.

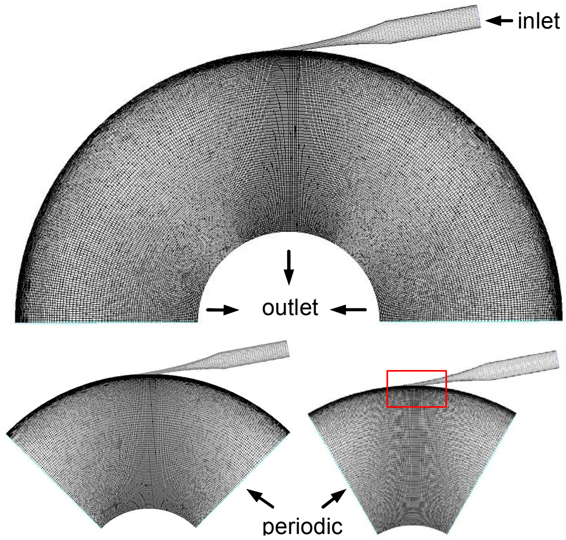

Fig. 3. The calculation domain for three different nozzle configurations

The calculation grid at the disk wall was refined in accordance with the assumed turbulence model so as to keep the y+ value in the range 1÷5. The mesh was also refined in the inlet and outlet regions and in the region near the outlet from the nozzles, where the highest magnitudes of velocity occur, Fig. 4. This is a structural mesh divided into blocks, which contains from 575 000 (six nozzle geometry) up to over 1 125 000 hexahedral cells (two nozzle model).

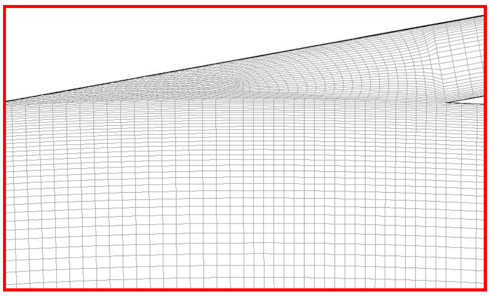

Fig. 4. Calculation domain near the nozzle outlet (symmetry plain view)

2. Boundary conditions and model

CFD calculations of various models of Tesla disk turbines were carried out on the basis of the RANS model [8] supplemented by the k-ω SST [9] turbulence model available in the computer program Ansys Fluent.

Numerical discretisation of the set of fundamental equations was performed using the finite volume method. The “segregated” solver with the sequential solving of the governing equations as well as the SIMPLE algorithm for correction of pressure and velocity were applied. Discretisation of convection fluxes was performed using an “upwind” scheme of the 2nd order accuracy. The time-domain discretisation was made by an ”implicit” scheme. At the beginning of calculation process all under-relaxation factors were lowered compared to their default values. The calculations were carried out until the stationary state is reached, lowering the residua of particular equations by 4 orders of magnitude (8 orders of magnitude for the energy residuum).

Thermodynamic parameters assumed for CFD calculations were found from preliminary 1D model calculations, making use of the data from literature sources. The nominal operating conditions are for the pressure drop from 13 bar to 1,1 bar. Nitrogen was assumed to be a working medium. The Redlich-Kwong-Aungier equation of state as a real gas model was chosen for the calculations, assuming polynomial dependence of the specific heats on the temperature. Pressure boundary conditions relevant for compressible flow were set. The calculations were carried out for a range of operating conditions (by changing the available pressure drop from 3 bar to the nominal value of 13 bar) and for one rotational speed of the rotor: 24000 rpm. Then it was able to make work characteristics of each microturbine model.

Flow efficiency of the Tesla turbine can be calculated from the following general formula:

$$\xi = \frac{P}{P_{is}} = \frac{P}{GH_{is}} = \frac{M\omega}{GH_{is}} \tag{1}$$

where ξ – isentropic efficiency,
P – power generated by the turbine disks,
P_{is} – theoretical power in isentropic conversion,
M – moment of force generated on disks surface,
ω – angular speed of the disk,
G – mass flow rate,
H_{is} – isentropic enthalpy drop.

Due to the fact that in each case we have the same nozzle geometry we must assume different number of rotating disks to get the same mass flow rate in the device. To fulfil the required operating conditions (mass flow rate 0.1 kg/s), 90 disks were assumed for two nozzle model, 45 for four nozzles and 30 for six nozzles. Moreover, if we set the number of disks as n than the number of interdisk channels will be equal to n+1 because we must include the outside channels located near the casing.

Then final formula for the efficiency can be rewritten as:

$$\xi = \frac{nM'\omega}{(n+1)G'H_{is}} \tag{2}$$

where G' - mass flow rate for one interdisk space,
M' - moment of force generated on one disk.

The isentropic drop of enthalpy can be found from the perfect gas approximation as:

$$H_{is} = c_P T_{in}\left[1-\left(\frac{p_{ex}}{p_{in}}\right)^{\frac{\kappa-1}{\kappa}}\right] \tag{3}$$

where c_p – specific heat,
T_{in} – inlet temperature,
p_{in} – inlet static pressure,
p_{ex} – outlet static pressure,
κ – ratio of specific heats.

3. Results of calculations

Contours of static pressure and velocity for Tesla turbine models with the outer disk diameter of 120 mm and 2,4 or 6 supplying nozzles are presented in Fig. 5. Fig. 6 presents velocity contours. The images show the view in the symmetry plane of the interdisk space for nominal operating conditions: rotational speed – 24000 rpm and pressure drop from 13 bar to 1 (absolute value), mass flow rate 0.1 kg/s.

It results from the pressure contours that most of the entire available pressure drop is accomplished in the nozzle, the other part takes place within the interdisk space. Expansion within the rotor leads to a pressure drop just behind the nozzles below the exit value of 1.1 down to 0.45 bar, 0.3 bar and 0.16 bar for two, four and six nozzle model, respectively. The velocities of the working medium at the outlet from the nozzles and at the inlet to the rotor reach 710 m/s, 680 m/s and 670 m/s respectively for 2, 4 and 6 supply nozzles.

Some distance downstream of the nozzles, high gradients of pressure and velocity occur and the configurations of isolines characteristic for the occurrence of shock waves. When a shock wave occurs, an increase of pressure and a decrease of flow velocity takes place. The compression within the shock wave is not desirable in bladed turbines [7].

Streamline patterns for the two, four and six-nozzle model, for the nominal load conditions are presented in Fig. 7. The streamlines are coloured by velocity magnitude. Fluid elements move along spiral pathlines from the outer to inner radius of the disk where outlet is located. The shape of the pathlines changes with operating conditions (pressure drop in the turbine) and number of supply nozzles. Sample pathlines presented for the three cases are slightly deformed around the nozzle outlet (rotor inlet) due to the influence of flow streams coming out from the nozzles. Fluid elements, depending on the nozzle configuration, make up to 2 rotations within the interdisk space before they reach the outlet section.

The longest path was observed for the two nozzle case. Paths are shorter for the case of four and six nozzles. According to the literature [5] when the turbine is spinning faster the flow path can be longer than in the case when the disks are rotating with lower velocity. The centrifugal forces increase then. Consequently, the fluid is forced to travel by longer paths and the transmission of energy can be done with a higher efficiency. Finally the knowledge of the shape of pathlines within the interdisk space can

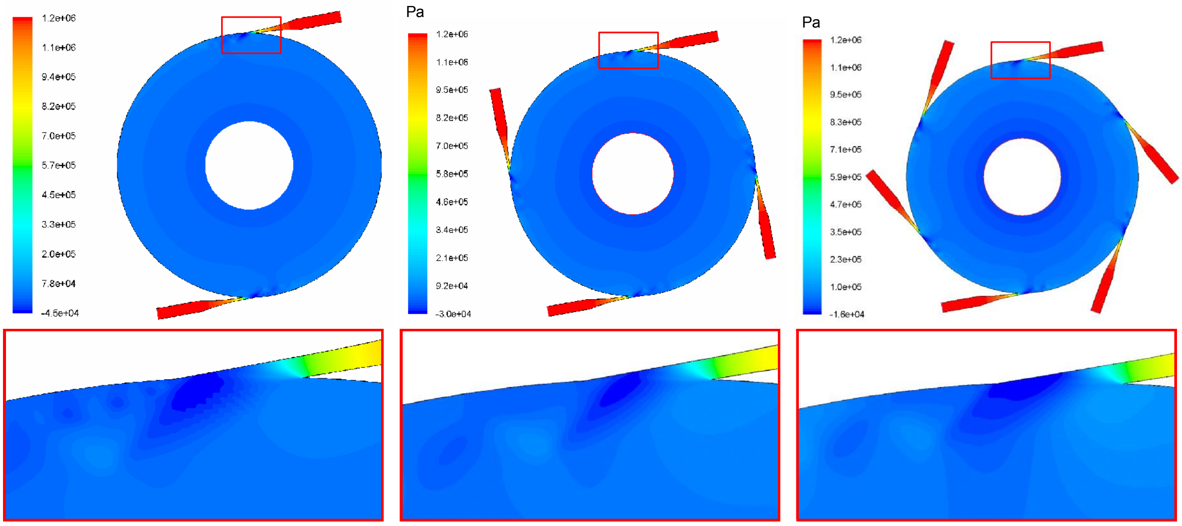

Fig. 5. Static pressure distribution in the symmetry plane, two, four and six nozzle supply

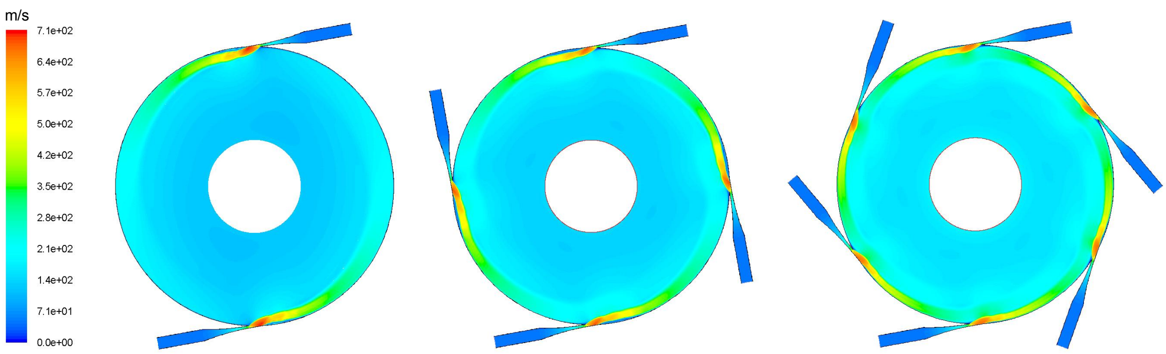

Fig. 6. Contours of velocity in the symmetry plane view for nominal load, two, four and six nozzles

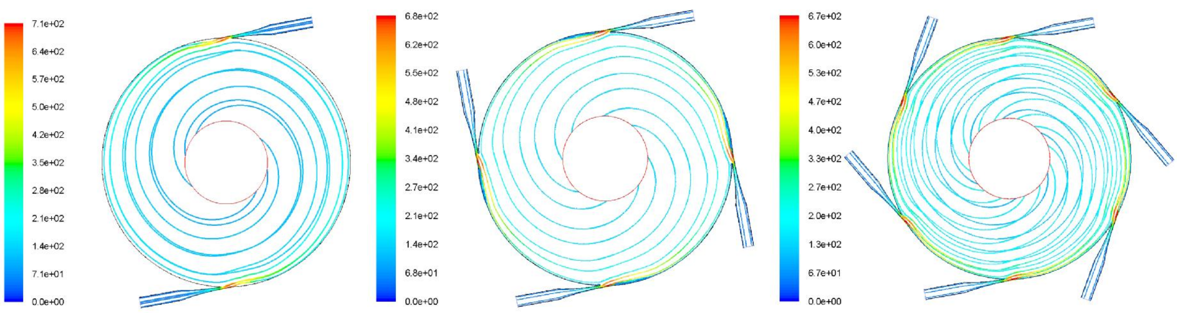

Fig. 7. Pathlines between the disks, coloured by velocity magnitude, nominal load conditions

be helpful in better designing the Tesla turbine disks.

The flow patterns observed within the interdisk space are decisive for the obtained output power and isentropic efficiency of the investigated Tesla turbine models presented in Figs. 8 and 9 as a function of mass flow rate of the multidisc configuration. The calculated output power tends to increase with the increasing of inlet pressure. For the nominal load the output power reaches about 6840 W for the case of two and four nozzles and 6770 W for six nozzle model. So the obtained power from the six-nozzle model is slightly lower than for the two and four nozzle models, which can be connected with the time of presence of fluid elements between the rotating disks (number of revolutions done by the streamline before the outlet). For the six nozzle case the paths were shorter than for the remaining models.

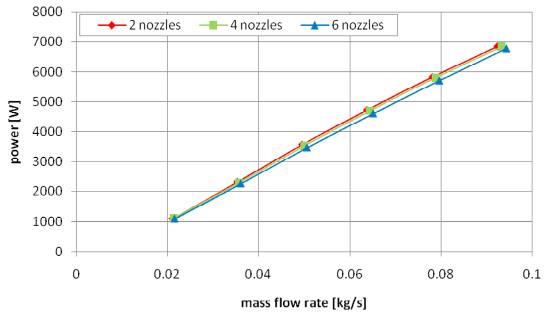

Fig. 8. Power for models of disk diameter 120 mm as a function of mass flow rate; the system supplied from two, four and six nozzles

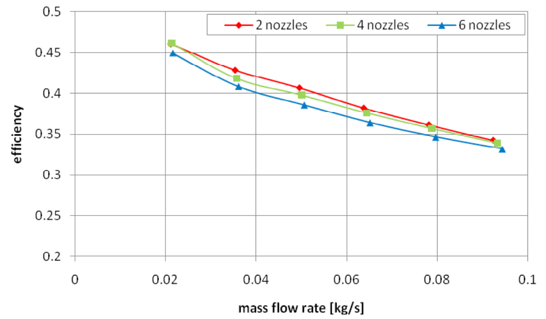

Fig. 9. Flow efficiency for models with of disk diameter 120 mm as a function of mass flow rate; the system supplied from two, four and six nozzles

A similar tendency can be observed at the efficiency diagram (Fig. 9), with a slightly lower efficiency for the six nozzle model. The isentropic efficiency for the nominal turbine load (13 bar for inlet pressure) is equal to about 34% for two and four nozzle model and 33% for six nozzle model. Higher values of efficiency are observed for smaller pressure drops but in order to obtain the required power from the device the number of installed disk should be largely increased then.

4. Conclusions

The presented results of numerical investigations show that Tesla turbine models can yield reasonable flow efficiencies of 30-40% and therefore can be applied in micro-power installations. The presented multi-disk models operating at a pressure drop from 13 to 1 bar, mass flow rate 0.1 kg/s and rotational velocity 24 000 rpm can give the total power up to almost 7 kW. The device supplied from more nozzles can have proportionally fewer rotating disks to achieve the same power.

More CFD simulations should be performed including more complex geometry without simplifications such as the lack of outlet holes and the clearance over the disk tips. Also using more complex turbulence model like the Reynolds Stress Model could bring us closer to better understanding of the flow phenomena, which occur between the rotating disks.

5. References

1. Patent no: 1,061,206, United States Patent Office, N.Y. Turbine / Tesla N., Patented May 6, 1913.
2. Rice W. Tesla Turbomachinery // Proc. 4th Int. Tesla Symp., Serbian Academy of Sciences and Arts, Belgrade, Yugoslavi. – 1991. – P. 117–125.
3. Patent no: 6,973,792 B2, United States Patent Office, N.Y. Method of and apparatus for a multi-stage boundary layer engine and process cell / Hicks K., Patented Dec. 13, 2005.
4. Kosowski K. Steam and Gas Turbines with examples of Alstom technology / 2-nd ed. – Alstom Power Sp. z o.o., 2007. – 376 p.
5. Praast T. D. Theory and Plans for the Tesla Turbine / TMMI 1315 Pepper Drive #7. – El Cajon CA 92021-1425. – 1993. – 11 p.
6. Rice W. An Analytical and Experimental Investigation of Multiple-Disk Turbines / J. Eng. Power, Trans. ASME. – 1965. – Vol. 87. – P. 29-37.
7. Lampart P. Investigations of aerodynamics of Tesla bladeless micro-turbines / P. Lampart, L. Jedrzejewski // J. Theor. and Appl. Mech. – 2011. – Vol. 49, No. 2. – P. 477-499.
8. Wilcox D.C. Turbulence modeling for CFD / 2nd ed. – Palm Drive: DCW Industries Inc., 2004. – 540 p.
9. Menter F.R. Ten Years of Industrial Experience with the SST Turbulence Model / F.R. Menter, M. Kuntz and R. Langtry // Turbulence, Heat and Mass Transfer 4, Begell House, Inc. – 2003. – P. 23-39.

Поступила в редакцию 18.07.2011

Л.Єнджеєвські, П.Лампарт. Мікротурбіна тесли фрикційного типу для маломасштабної когенерації

Турбіни Тесли рідко використовуються в комерційних енергетичних установках через їх своєрідні робочі характеристики та відносно низький внутрішній ККД. Проте вони мають деякий потенціал для їх застосування в розподілений когенераційних системах енергоустановок малої потужності, а також для роботи в органічному циклі Ренкіна. В даній статті представлені результати розрахунків течії в трьох моделях дискових турбін Тесли, що використовують азот у якості робочого тіла. Показано, що моделі турбін Тесли можуть забезпечити прийнятну гідродинамічну ефективність порядку 30-40% и тому можуть використовуватись в енергетичних установках малої потужності.

Ключові слова: *мікротурбіна Тесли, органічний цикл Ренкіна, обчислювальна аеродинаміка.*

Л.Енджеевски, П.Лампарт. Микротурбина теслы фрикционного типа для маломасштабной когенерации

Турбины Теслы редко используются в коммерческих энергетических установках из-за их своеобразных рабочих характеристик и относительно низкого внутреннего КПД. Однако они располагают некоторым потенциалом для их применения в распределенных когенерационных системах энергоустановок малой мощности, а также для работы в органическом цикле Ренкина. В данной статье представлены результаты расчетов течения в трех моделях дисковых турбин Теслы, использующих азот в качестве рабочего тела. Показано, что модели турбин Теслы могут обеспечить приемлемую гидродинамическую эффективность порядка 30-40% и поэтому могут использоваться в энергетических установках малой мощности.

Ключевые слова: *микротурбина Теслы, органический цикл Ренкина, вычислительная аэродинамика.*