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NUMERICAL ANALYSIS OF THE EXPANSION PROCESS IN A TWO-STAGE AXIAL TURBINE OPERATING WITH MDM SILOXANE

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ЧИСЛЕННЫЙ АНАЛИЗ ПРОЦЕССА РАСШИРЕНИЯ В ДВУХСТУПЕНЧАТОЙ ОСЕВОЙ ТУРБИНЕ, РАБОТАЮЩЕЙ С MDM СИЛОКСАНОМ

The problem of decreasing of fossil fuel consumption and energy efficiency is one of today's major conceptions in the field of energy economics. Waste heat recovery is one of the promising solutions for this problem. One of the ways to increase efficiency of the waste heat recovery process is using siloxanes as working fluids for organic Rankine cycles (ORC).

SPbPU scientists have analyzed peculiarities of the steady-state expansion process in the two-stage MDM siloxane turbine. The turbine is based on the principle of a classic velocity stage. First stage of the turbine was designed using the SPbPU high pitch-chord ratio supersonic design. The airfoils of the second stage are subsonic. A pressure ratio of the turbine is 37.2. Progressive steps of the initial temperature, pressure ratio and rotational velocity were used to obtain convergence of the solution process. The main factors, leading to the efficiency decreasing, were established and described.

The efficiency and power output of the investigated turbine stage were estimated as 67 – 68% and 544 kW respectively.

ORGANIC RANKINE CYCLE, AXIAL TURBINES, MDM SILOXANE, COMPUTATIONAL FLUID DYNAMIC

Необходимость снижения уровня потребления органического топлива и повышения энергоэффективности – в числе основных проблем в области современной энергетической экономики. Утилизация сбросной теплоты – одно из перспективных направлений в этой области. Использование силоксанов в качестве рабочих тел для органического цикла Ренкина (ORC) позволяет повысить эффективность утилизации сбросной теплоты.

Ученые СПбПУ проанализировали особенности стационарного процесса расширения MDM силоксана в силоксановой турбине. Турбина основана на принципе классической ступени скорости. Первая ступень турбины – высокоперепадная с большим относительным шагом лопаток рабочего колеса. Вторая ступень дозвуковая. Степень понижения давления 37.2. Для достижения сходимости процесса решения использовалось ступенчатое повышение начальных параметров. Установлены и описаны основные факторы, приводящие к уменьшению эффективности турбины. Эффективность и мощность турбины были оценены как 67-68% и 544 кВт соответственно.

ОРГАНИЧЕСКИЙ ЦИКЛ, АКСИАЛЬНЫЕ ТУРБИНЫ, MDM СИЛОКСАН, ВЫЧИСЛИТЕЛЬНАЯ ДИНАМИКА ЖИДКОСТИ

Introduction

The problem of waste heat recovery is one of up-to-date problems in the energy efficiency field (Larjola [1], Vescovo [2]). Analysis of the Key World Energy Statistics [3] shows, that the highest volumes of waste heat resources take place at different thermal power plants, cement, metallurgical and chemical productions. In Russia it is also the gas transport industry. The thermal power of waste heat at the all gas compressor stations of "Gazprom" is 87.9 GW by the estimation of Lykov et al. [4]. Rough estimations of waste heat thermal power at different productions in Russia, made on the base of Key World Energy Statistics [3], are:

3,9 GW in the cement industry;
2,8 GW in the metallurgical industry;
1,9 GW in the chemical industry.

The total waste heat thermal power in Russia is equal to 20 GW of electrical power by the most conservative estimate.

Nowadays in most cases the plants for waste heat recovery are based on Organic Rankine Cycle (ORC) because of higher cycle efficiency (Larjola [1], Hung et al. [5], Vescovo [2]). However, typically the efficiency of ORC recovery plants is less than 20% and strongly depends on the working fluid selection (Lecompte et al. [6]). Modern requirements for environment safety determine ozone depletion potential (ODP) and global warming potential (GWP) as main criteria for the selection process. It was shown that in this case the most promising alternatives to different hydrocarbons, freons and alcohols are zeotropic mixtures and siloxanes (Heberle et al. [7], Chys et al. [8], Weith et al. [9]) (fig. 1). Using of siloxanes in ORC allows increasing efficiency of the recovery units up to 23 – 25%. The aspects of siloxanes using in ORC were investigated by Lai et al. [10], Fernandez et al. [11], Uusitalo et al. [12].

The turbines for organic working fluids have essential differences in details of the expansion process in comparison with typical gas and steam turbines.

The special supersonic design is required for such turbines due to low speed of sound of different organic working fluids. Supersonic velocities in the turbine flow path and dense-gas effects have a significant influence on the turbine efficiency (Condego et al. [13], Guardone et al. [14]). The analysis of available experimental data shows that in case of axial turbines with mean diameter up to 500 mm the efficiency can dramatically drop down to 55% when the efficiency of traditional steam turbines is in the range 85-90% (see table 1).

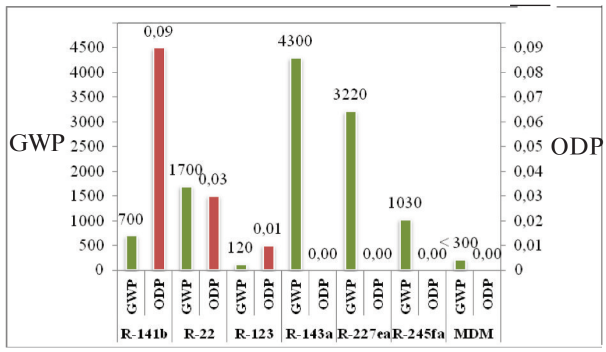

Fig. 1. Comparison of ODP and GWP of different working fluids

The design and performance of the siloxane axial turbines were investigated by Klonowicz et al. [22], Sebelev et al. [23]. However, relatively low level of efficiency of ORC axial turbines shows that details of the expansion process have to be investigated more clearly, especially in case of such a new class of working fluids as siloxanes. Thus, the scope of the present paper is to investigate the peculiarities of the MDM siloxane expansion process in the flow path of two-stage axial turbine.

Investigation object

Initial parameters of the expansion process. MDM siloxane was chosen as working fluid for the expansion process. The initial pressure p_0 was set as 0,75 MPa. The initial temperature T_0 was set as a vapor saturation temperature at chosen initial pressure. The turbine pressure ratio has been chosen as 37,2 to provide the required turbine enthalpy drop upon the condition of 500 kW power output of the turbine. Trans- and supercritical initial parameters were not considered. Positive slope of MDM vapor saturation curve provides inability of intersection between expansion process curve and two-phase region.

Table 1

Some available experimental data of ORC turbines efficiency

Authors	Working fluid	Turbine type	Turbine efficiency
Kang [15] Pei et al. [16]	R-245fa R-123	Radial-inflow turbine	0,822 0,625
Yamamoto et al. [17]	R-123	Centripetal turbine	0,500
Fu et al. [18] Klonowicz et al. [19] Li et al. [20] Ngyen et al. [21]	R-245fa R-227ea R-123 Pentane	Axial turbine	0,637 0,530 0,585 0,498

The turbine. A two-stage axial turbine was chosen as the investigation object. This turbine is based on the principle of a classic velocity stage, where the main part of the enthalpy drop falls at the first stage as shown in fig. 2.

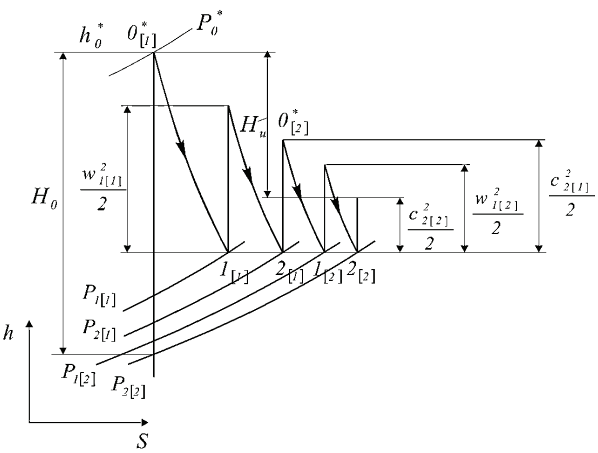

Fig. 2. Expansion process in a classic velocity stage (Lapshin [24])

The first stage was designed using the SPbPU high pitch-chord ratio supersonic design (Rassokhin [25]). The design of the 2nd stage is subsonic. The blade wheels of the turbine are shrouded. The nozzles and blade wheels design is shown in figure 3. The main geometric parameters of the turbine are presented in table 2.

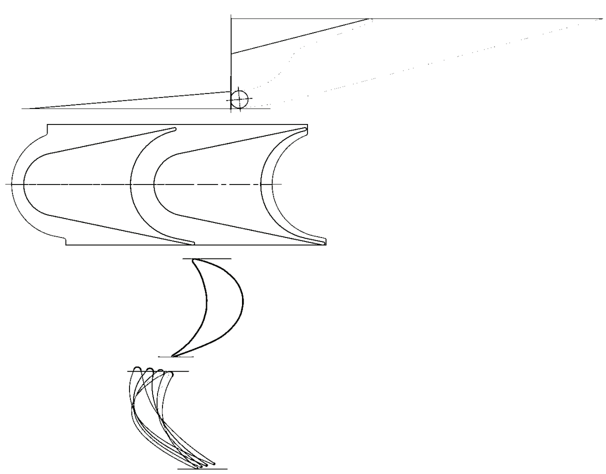

Fig. 3. The nozzles and blade wheels design

Table 2

The main geometric parameters of the turbine

Parameter	Dimensions	Value	
		1st stage	2nd stage
D_m	mm	550	
n	rev/min	3000	
H_0	kJ/kg	620,3	
u/C_{ax}	-	0,40	0,59
ε	-	0,89	1,00
Z_1	-	29	107
l_1	mm	30,4	82,5
α_1	grad.	5,0	20,0
ΔL_{ax}	mm	4,0	6,0
ΔL_{tc}	mm	1,0	1,0
β_1	grad.	12,0	56,0
Z_2	-	53	92
l_2	mm	60,4	88,5
β_2^*	grad.	12,0	30,0

Numerical simulation method

The SPbPU method for numerical simulation of processes in supersonic turbines, described by Zabelin et al. [26], was used. ANSYS CFX was used to provide the numerical simulation.

The original relation between the number of nozzles and number of working blades is 29/53 for the first stage and 107/92 for the second stage. The relation 1/2/4/3 and periodic boundary conditions were used in the computational model. This assumption is correct to be used with Frozen Rotor interface between the nozzles and blade wheels areas because the relations between connecting areas in this case are 1:1,094, 1:0,991 and 1:0,872 respectively. The modeling of blade wheel tip shroud was also considered in numerical model in assumption of rotating motion of tip shroud domain. The computational model of the investigated turbine is presented in fig. 4.

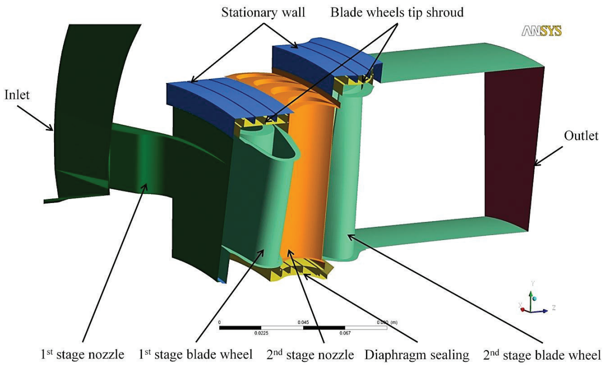

Fig. 4. Computational model of the investigated turbine

High-Reynolds version of the k-ω SST turbulence model was used. Steady-state Frozen rotor interface between the nozzles and blade wheels areas was used to model rotor-stator interactions. Flow parameters of the turbine were obtained by averaging of their values for 4 positions of the rotor relatively to the stator in the range of the 1st stage blade wheel pitch angle.

Aungier Redlich Kwong real gas equation of state was used to model thermodynamic properties of MDM during the expansion process. The main parameters have to be specified are: molar mass, critical temperature and pressure, acentric factor and boiling temperature. Zero pressure polynomial coefficients were obtained with using REFPROP databases to evaluate specific heat capacity of MDM. Kinetic Theory models were used to model transport properties of MDM. Rigid Non Interacting Sphere model was used to model MDM dynamic viscosity behavior.

Total parameters at the inlet (p = 0,75 MPa, T = 523,15 K) and static pressure at the outlet (p = 0.02 MPa) were specified as boundary conditions in the computational model. Progressive steps of the boundary conditions were used to obtain convergence of the solution process. The iteration steps between the changings of boundary conditions were different to decrease their negative influence on the convergence process. Monitoring of the RMS residuals, imbalances and turbine efficiency and power output were used to control convergence of the solution process. The criteria of the convergent solution in the present research were:

drop of the RMS residuals more than 10^2;

imbalances less than 0,5%;

fluctuation of the turbine efficiency and power output less than 5%.

The parameters of the computational domains discretization were chosen on the base of the grid independency study, presented by Sebelev et al. [23].

Discussion of the results

The values of the calculated thermodynamic and transport properties were compared with the values obtained with using REFPROP databases to estimate tolerance of the calculated results. Maximum deviation between CFX and REFPROP results was less than 5% for specific heat capacity and dynamic viscosity.

The turbine efficiency was estimated with consideration of the losses due to unsteady rotor-stator interaction with respect to equation:

$$\eta_{t-s} = \frac{M_{BW} \;\; \pi \; n}{30 \;\; G \; H_0} \, (1 - \zeta_{r-s}). \tag{1}$$

Te these losses were estimated as 0.13 (Natalevich [19]). Thurbine power output was calculated with respect to equation:

$$N = \frac{M_{BW} \;\; \pi \;\; n}{30} \, (1 - \zeta_{r-s}). \tag{2}$$

Calculated turbine parameters are presented in table 3.

Table 3

Calculated turbine parameters

Parameter	Dimensions	1st stage	2nd stage	Turbine
p_0^*	MPa	0,7725	0,0709	–
T_0^*	K	523,77	498,53	–
G	kg/s	13,915		–
c_1	m/s	219,75	127,68	–
α_1	deg.	18,37	-	–
p_1	MPa	0,1073	0,0367	–
w_1	m/s	147,03	61,06	–
w_2	m/s	206,53	153,23	–
c_2	m/s	128,75	97,09	–
p_2	MPa	0,0425	0,0199	–
T_2	K	-	488,87	–
H_N	kJ/kg	30,00	11,09	–
H_0	kJ/kg	45,58	21,38	58,34
π	-	18,16	3,57	38,91

Ending table 3

Parameter	Dimensions	1st stage	2nd stage	Turbine
$\overline{G_d}$	%	–	0,52	–
$\overline{G_{tsh}}$	%	3,90	1,05	–
u/C_{ax}	–	0,40	0,59	0,36
ρ_t	–	0,342	0,481	–
φ	–	0,897	0,857	–
M_{BW}	N·m	994,4	738,92	1733,33
F_{ax}	kN	10,52	2,39	12,92
$\eta_{t\text{-}s}$	–	–	–	0,671
$\eta_{t\text{-}t}$	–	–	–	0,730
N	kW	–	–	544,54

It is noteworthy that 1st stage has a significant value of reaction (0.34) despite its impulse design. Concurrently its u/C_{ax} value is 0,4 and velocity ratio φ is less than 0,9. These phenomena can take place due to the flow over-expanding in the 1st stage nozzle when the u/C_{ax} value is not optimal (Rassokhin [25]). It is important to emphasize that high value of the MDM density leads to the high value of the axial force acting to the rotor. More than 80% of the total axial force (10,52 kN) accrue to the 1st stage.

The 2nd stage fully performs its function. It decreases the flow velocity from 129 m/s to 97 m/s when the reaction value is 0,48 and pressure ratio is 3,57. It is important to emphasize that more than 42% of the total rotor torque accrue to the 2nd stage whereas its contribution to the total axial force is only 18,5%.

Analysis of the flow structure in the 1st stage shows negative influence of high values of hub and shroud overlaps on the flow characteristics due to spurious vortices formation. This process is illustrated in fig. 5. The influence of the hub and shroud overlaps on the efficiency of small-scaled turbines was investigated by Natalevich [27] and described in details by Zabelin et al. [26]. In case of the investigated turbine the negative influence of the hub and shroud overlaps is minimized by high speed of the MDM specific volume increasing after the nozzle. This leads to the stiff localization of the spurious vortices as shown in fig. 5.

It is also has to be emphasized that intensity of the oblique shock waves has its maximum at the nozzle hub and decreases towards to the shroud as illustrated in fig. 6. Another side of this phenomenon is that the mass-flow averaged nozzle outlet angle strongly differs from its geometrical value. These phenomena are the consequences of the flow linear motion in the axial clearance area as established by Kirillov [28] and Traupel [29].

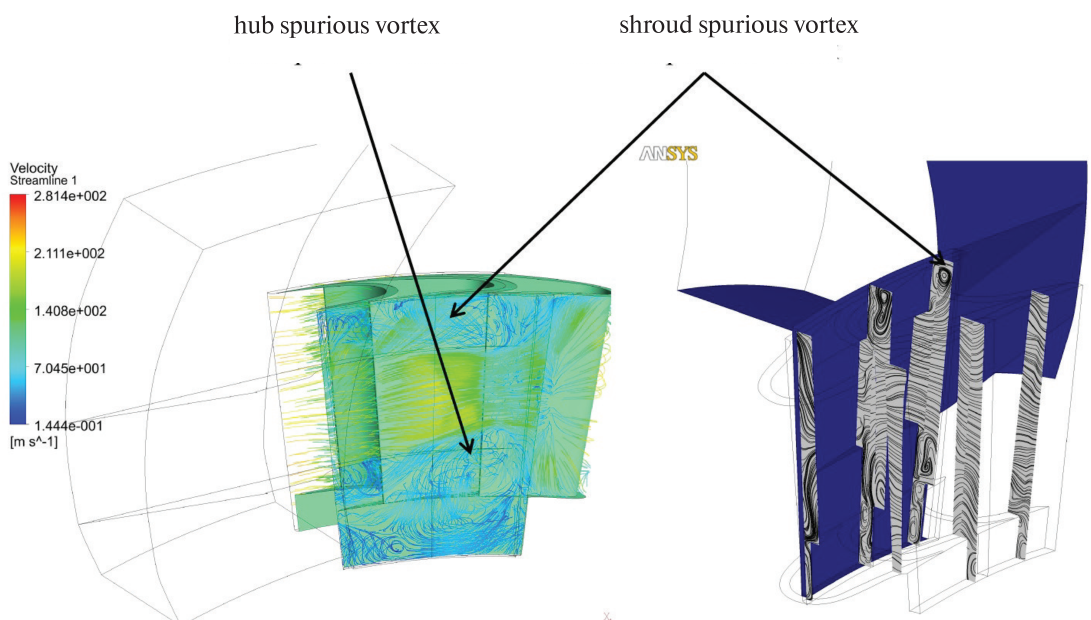

Fig. 5. Flow structure in the 1st stage

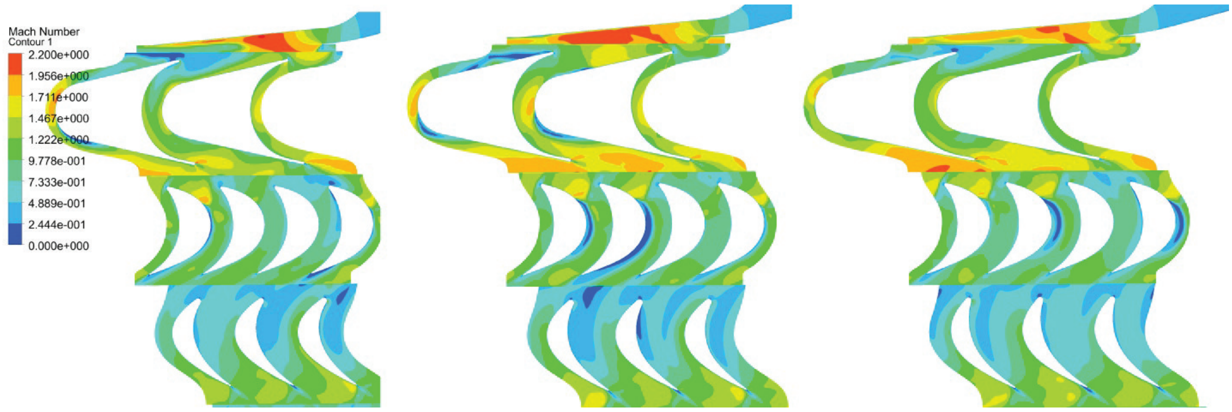

Fig. 6. Mach number fields on the cylindrical sections (left-to-right: $0{,}05l_1$, $0{,}5l_1$, $0{,}95l_1$)

Analysis of the flow structure in the axial cross section shows that flow tends to increase its “mean diameter” due to extreme increasing of its specific volume. This process is represented in fig. 7.

The occurrence of the normal shock wave after a turbine blade wheel was also described by Sebelev et al. [23]. Nonsufficient meridian fanning of the flow path leads to occurrence of additional losses on account of flow impingent on the tip shroud of the 2nd stage. This phenomenon leads to formation of the shroud vortex in 2nd stage nozzle. Hub overlap between 1st stage blade wheel and 2nd stage nozzle leads to formation of the hub vortex in the 2nd stage nozzle. These vortices occupy up to 50% of the nozzle cross sectional area. This situation is dramatized by the interaction of these vortices with secondary flows in the blade wheel. Filling of the cross sectional area by the passive working fluid leads to local increasing of flow velocities up to supersonic values. Conversely, this leads to increasing of the profile losses because of their subsonic design. Thus, it can be assumed that neglect of extreme radial expansion of the flow is the main source of losses in the investigated case.

To sum up, the described phenomena lead to decreasing of the turbine efficiency down to 67–68% when its power output is 544 kW. Taking into account that other authors have described the same phenomena for the ORC axial turbines it is reasonably safe to suggest that the main factor, which leads to the low efficiency of axial ORC turbines, is extreme radial expansion of the working fluid. In this case it becomes significant to take into account strong radial expansion of the organic working fluids in the turbine design process.

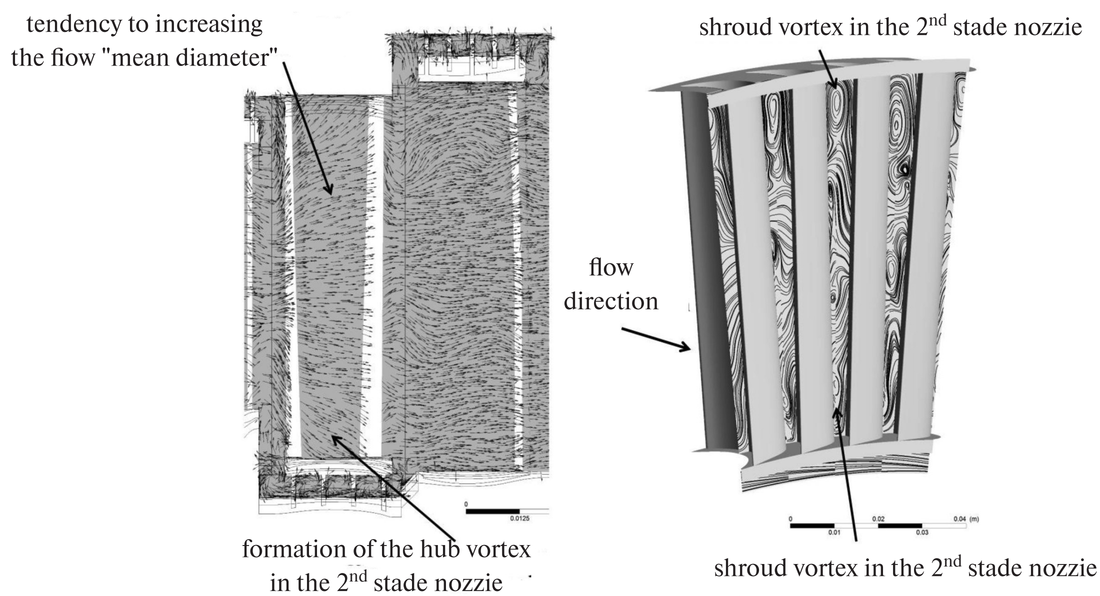

Fig. 7. Flow structure in the 2nd stage

Conclusions

The expansion process in the MDM siloxane turbine was modeled. The peculiarities of his process were outlined. Most of the outlined peculiarities are typical for the supersonic axial turbines because of subcritical initial parameters of the siloxane vapor. However, the strong relation of the siloxane properties to the thermodynamic parameters determines its nonconventional behavior during the expansion in the blade wheel. It was shown that in case of the axial turbines strong radial expansion of the siloxane is the main factor, which leads to dramatic decreasing of the turbine efficiency. As a result, calculated efficiency of the investigated turbine is 0,67–0,68 when its power output is 544 kW.

Nomenclature

GW – Gigawatt
GWP – Global Warming Potential
kW – Kilowatt
MDM – Octametyltrisiloxane
ODP – Ozone Depletion Potential
ORC – Organic Rankine Cycle
SPbPU – Peter the Great St. Petersburg Polytechnic University
c – velocity in stationary frame, m/s
D_m – mean diameter, m
F – force, N
G – mass flow rate, kg/s
$\overline{G}$ – relative leakage (leakage value divided by the mass flow rate)
H – enthalpy drop, kJ/kg
l – height, mm
M – torque, N·m
N – power output, W
n – rotational speed, rev/min
p – pressure, MPa
T – temperature, K
u/C_{ax} – stage load coefficient
w – velocity in relative frame, m/s
Z – number of nozzles (blades)
α_1 – outlet angle in stationary frame, deg.
β_1 – blade wheel inlet angle in relative frame, deg.
β_2^* – blade wheel outlet angle in relative frame, deg.
ΔL – clearance value, mm
ε – partial admission ratio
π – pressure ratio
η – efficiency
ρ_t – thermodynamic reaction
φ – nozzle velocity ratio

Subscript

BW – blade wheel
N – nozzle
ax – axial
d – diaphragm
r-s – rotor-stator
tc – tip clearance
tsh – tip shroud
t-s – total-to-static
t-t – total-to-total
0 – related to the turbine inlet
1 – related to the area after the nozzle
2 – related to the area after the blade wheel
[1] – related to the 1st stage
[2] – related to the 2nd stage
* – related to the total parameters

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