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Study on applicability of radial-outflow turbine type for 3 MW WHR organic Rankine cycle

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Abstract

The article presents the results of study on the reasonability of using radial-outflow turbines in ORC. Peculiarities of radial-outflow turbine design utilizing modern design technologies and application to ORC was considered in the first part of the paper. The second part of the paper describes the selection process of the best turbine type for a 3 MW WHR ORC power unit for an internal combustion engine. The selection was performed among different turbine types, like radial-inflow, axial and radial-outflow turbines which were designed with given boundary conditions. The advantages and disadvantages of their application were shown. Eventually, the recommendations regarding application of different turbine types for a 3 MW WHR Organic Rankine Cycle were given. For this particular cycle design, turbines of radial-outflow type were chosen. Their application enables the increase of mechanical output power by 11 % compared to original radial-inflow turbines.

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Keywords: Radial-outflow turbine; reasonability study; organic Rankine cycle; WHR unit

1. Introduction

The radial-outflow turbine (ROT) design was first invented by the Ljungström brothers in 1912, however it was rarely used due to a number of reasons. One of which was related to the decrease of turbine-specific work due to the increase of the peripheral velocity ($u_{in} < u_{out}$) while expanding the vapor. Another reason was the usage of steam as a

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working fluid. It is known from thermodynamics that the expansion of steam is characterized by high enthalpy drops, high volumetric flows and high volumetric ratios. Thus, a significant number of stages is needed to convert the enthalpy drop of the fluid into mechanical energy [1].

Nowadays, turbines for ORC are widely used. Organic fluids have high molecular weight which leads to significantly lower enthalpy drops. Moreover, application of a non-standard design approach which is based on non-equal enthalpy drop distribution between stages allows for the design of radial-outflow turbines with high efficiency - even higher than equal axial and radial-inflow turbines at times[2].

A schematic view of the radial-outflow turbine is shown in Fig. 1.

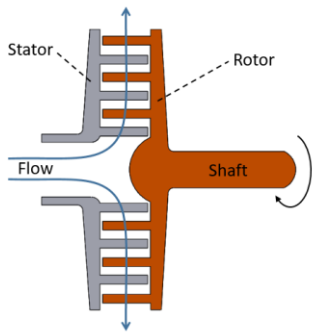

Fig. 1. Radial-outflow turbine (schematic view)

Obviously the possibility of applying ROT in organic Rankine cycle requires more detailed study. In paper [3] ORC using heat of the exhaust gases of the petrol engine CAT G3612 was proposed. It was shown that the usage of the ORC allows for achievement of an 18% power boost for the internal combustion engine without burning additional fuel. Wherein the source of generated additional power was two radial turbines.

This article presents the theoretical basis of the radial-outflow turbine development. The second part of the paper is dedicated to the problem of the most optimal turbine design selection (axial, radial-inflow, radial-outflow) for its usage in the WHR organic Rankine cycle [3].

Nomenclature

<u>*Symbols:*</u>

c	absolute velocity
c_0	spouting velocity
D	diameter
H_0	isentropic enthalpy drop
H_u	real enthalpy drop
K_u	circumferential velocity ratio
K_z	axial velocity component ratio
u	circumferential velocity
α	velocity flow angle
η	peripheral efficiency

<u>*Indexes:*</u>

0	stage inlet parameters
1	parameters after nozzle
2	parameters after blade
j	index
u	circumferential velocity component
z	axial velocity component

2. Design Technique

While designing a multistage turbine, it is important to determine the optimal number of stages and to distribute enthalpy drop between them.

To solve the ROT design problem we will use the method of the enthalpy drop distribution for the group of stages with a given axial and circumferential velocity components in all cross-sections [4,5,6].

Peripheral stage efficiency is determined by the formula:

$$\eta = \frac{H_u}{H_0} = \frac{u_1 c_{1u} - u_2 c_{2u}}{H_0}. \quad (1)$$

Let's assume that changing of axial and circumferential velocity components are determined as:

$$K_{jz} = \frac{c_{jz}}{c_{0z}}, \quad K_{ju} = \frac{u_j}{u_0} = \frac{D_{mean\,j}}{D_{mean0}}, \quad j = 1{,}2. \quad (2)$$

Taking into account (2) formula (1) can be transformed to the expression:

$$\eta = \upsilon_0^2 \cdot \overline{c_{0z}} \cdot \left(K_{1u} \cdot K_{1z} \cdot ctg\alpha_1 - K_{2u} \cdot K_{2z} \cdot ctg\alpha_2 \right), \quad (3)$$

Where:

$$\overline{c_{0z}} = \frac{c_{0z}}{u_0}, \quad \upsilon_0 = \frac{u_0}{c_0}, \quad c_0 = \sqrt{2H_0}. \quad (4)$$

For given values $K_{jz}, K_{ju}, \overline{c_{0z}}, \upsilon_0$ it is required to find such angles α_1 and α_2, where the peripheral efficiency (3) would be maximized.

Generally, the simplified algorithm of the design process can be represented as a sequence of operations:

a. Set parameters $K_{jz}, K_{ju}, \overline{c_{0z}}, \upsilon_0$;
b. Solve the optimization problem: find the angles α_1 and α_2 which provide maximum efficiency (3);
c. Set D_1;
d. Find D_{mean} for all stages (by known K_{ju});
e. Calculate all remaining geometry, flow parameters and the height of the blades.

In the case when a multistage turbine is designed, the equation (3) should be written as the sum of all stages peripheral efficiency.

The proposed algorithm has a number of unique features:

- In the turbine designing process, the optimization task is solved. This enables obtainment of the highest possible turbine efficiency for the given boundary conditions at the stage of preliminary design.
- Enthalpy drop for each stage is determined automatically in terms of turbine maximum efficiency. This results in an uneven enthalpy drop distribution between stages. Fig. 2 presents the process of the working fluid expansion in the axial turbine when it is designed using a standard design approach (a) and the same process in the radial-outflow turbine designed according to the proposed technique (b). The figure shows that in the ROT work output increases at each stage. Obviously, this is due to a significant increase in the rotor's peripheral velocity stage by stage with diameter increasing.
- Due to the uneven work distribution between the stages, blade profiles obtained by the proposed method are significantly different from each other. Thus, it is not possible to make a uniform profile for its application in several stages. This limitation can be critical for designing a multi-stage axial turbine. And vice versa, the

standard approach of the enthalpy drop distribution between stages obviously can't be applied for designing ROT, since the geometrical parameters of turbine stages have significant differences from the first stage to the last one.

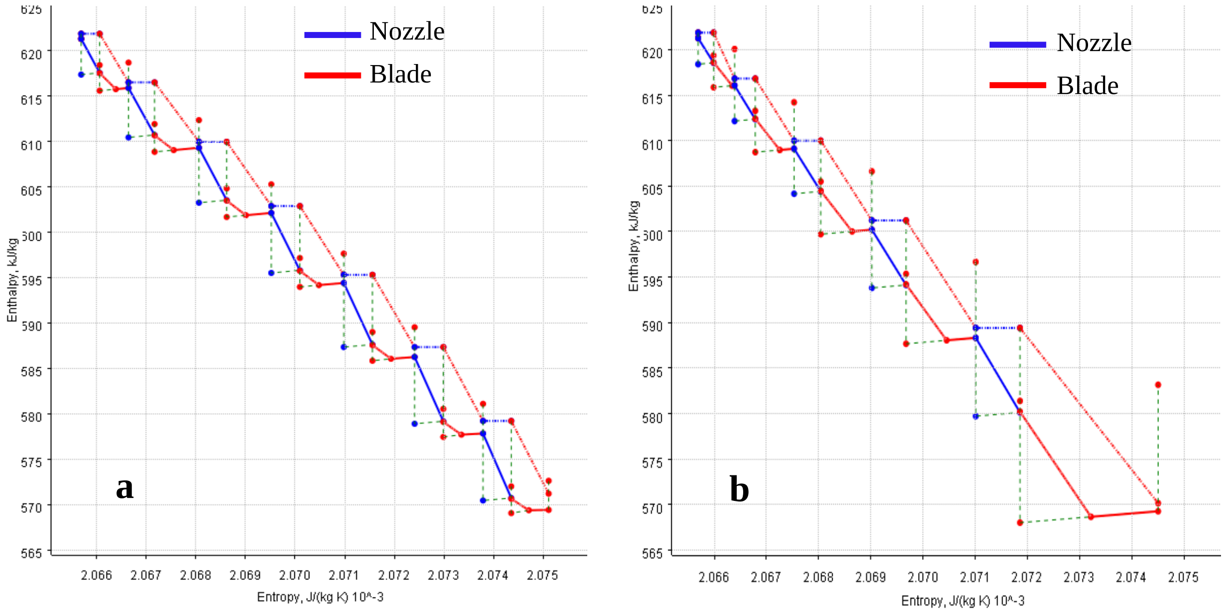

Fig. 2. h-s diagram of the working fluid expansion when turbine is designed using standard design approach (a) and using proposed design technique (b).

3. Preliminary Design and Selection Process of the Different Turbine Types

To perform a comparative analysis of a different turbine designs proposed in [3] 3 MW WHR ORC for the combustion engine CAT G3612 [7] has been chosen.

The cycle consists of 6 heat exchangers; 2 turbines: high pressure (HPT) and low pressure (LPT); 2 pumps: high pressure (HPP) and low pressure (LPP); and the condenser (fig. 3). Working fluid expands in the HPT up to the pressure of the LP loop (7 bars). After that the flows from the different loops mix and expand in the shared LPT. This means that the mass flow through the LPT is equal to the sum of the HP and LP flows. Flows are again split at the condenser outlet.

Boundary conditions for the turbines design are presented in table 1. Operational range of the HPT is in supercritical region, while the LPT would be operating under subcritical conditions.

Table 1. Boundary conditions for turbine designs

Parameter	Unit	HPT	LPT
Inlet total pressure	bar	45.00	7.000
Inlet total enthalpy	kJ/kg	621.87	515.63
Static pressure at outlet	bar	7.000	1.300
Static enthalpy at outlet	kJ/kg	569.26	480.18
Mass flow rate	kg/s	5.249	11.77

In proposed cycle [3] R245fa (pentafluoropropane) organic fluid was used as the working fluid. Fluid properties were calculated based on the NIST REFPROP library [8].

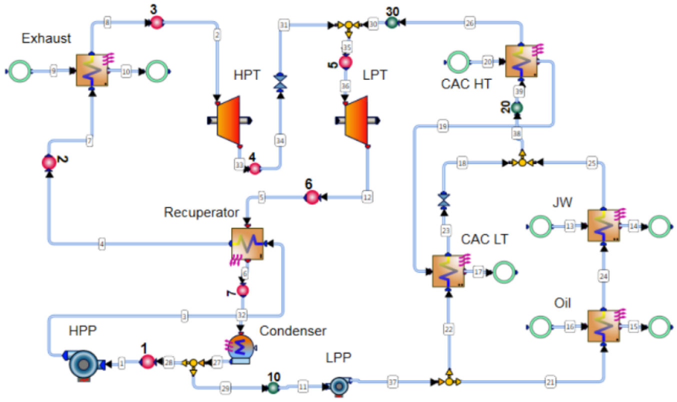

Fig. 3. The flow diagram of the cycle under consideration

(where LP and HP – low pressure and high pressure; LT and HT – low temperature and high temperature; CAC – charge air cooler; JW – jacket water; Oil – oil from engine lubrication system)

3.1. Radial-outflow Turbine Design

The aforementioned design approach has been implemented in a SoftInWay turbomachinery design / analysis tool AxSTREAM [9,10]. This software was used to design turbines.

To obtain the most efficient turbine, the design process was divided in two stages:

1) Preliminary design for the feasibility study which shows the influence of the number of stages on the turbine efficiency

2) Detailed turbine design with the optimal number of stages

Generally, enlargement of the turbine stage number will increase efficiency of the entire turbine. However, since in radial-outflow turbine stages are arranged in radial direction, the circumferential velocity of the blades will increase with radius increasing. This, in turn, can lead to an unwanted supersonic velocity and additional total pressure losses. Furthermore, a large circumferential velocity leads to significant increase in the bending stresses on the last stages of the turbine. Obviously for ROT design there exists an optimal number of stages for a given enthalpy drop.

Thousands of turbine designs were automatically generated using AxSTREAM's preliminary design tool. At this step an optimum number of stages was selected. Fig. 4 shows how number of stages affects a turbine's total-to-static efficiency for considered LPT and HPT.

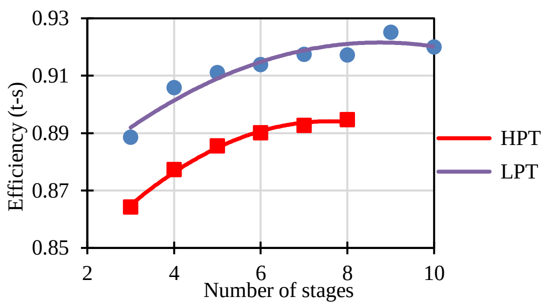

Fig. 4. Dependence of the efficiency on the turbine stage number

The maximum of HPT efficiency corresponds to an eight-staged design. While detailed designing it was found that, in this case, the obtained chord profiles are very small (approximately 2 mm). Such small blades are structurally difficult to implement. To enlarge the blade chord turbine diameter can be increased, however, this will reduce the efficiency. Based on these considerations, it was decided that HPT would have a five-staged design (preliminary design of five-staged HPT have shown that all blade chords are larger than 4 mm).

Maximum of LPT efficiency was achieved when turbine had 9 stages. Moreover, it was observed that in this case the rotor rotational speed is close to 3000 rpm. Therefore, for LPT detailed designing shaft rotational speed was fixed at 3000 rpm.

After the above-mentioned steps have taken place, detailed design of HPT and LPT was performed. For detailed designing, AxSTREAM's preliminary design tool was also used. Turbines with a higher efficiency than in preliminary feasibility study were obtained due to increasing the number of calculation points and narrowing the ranges of parameter variation. Meridional view of the designed turbines are shown in Fig. 5.

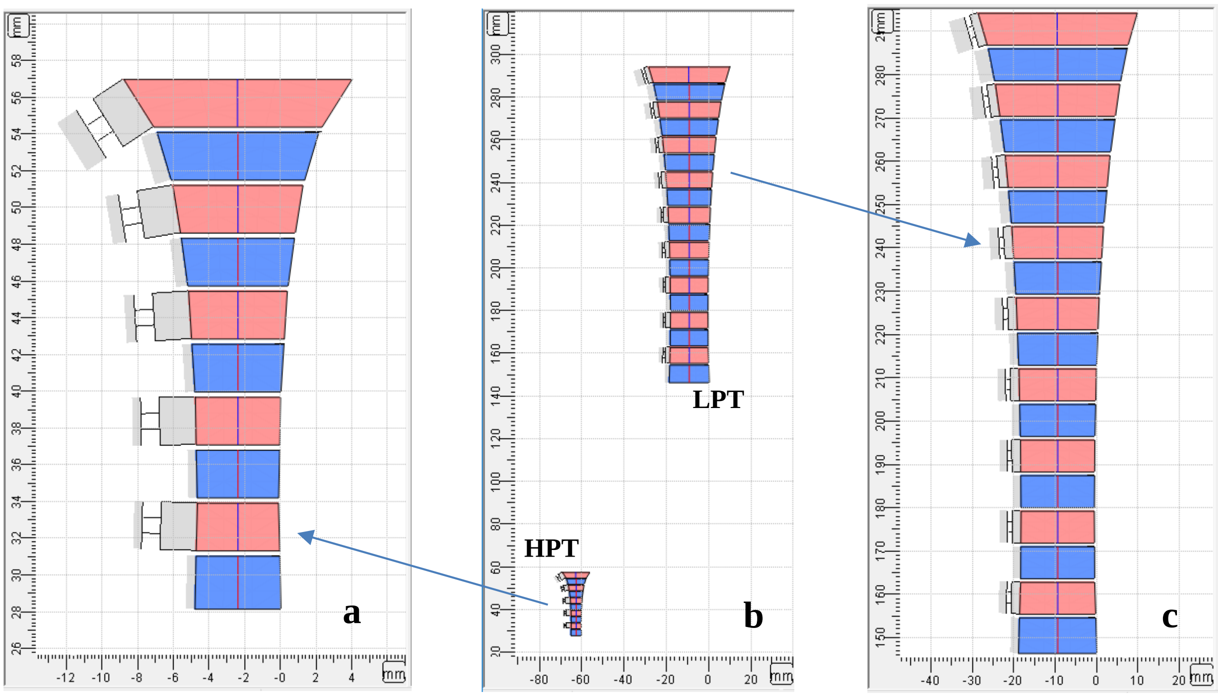

Fig. 5 HPT and LPT meridional view comparison

It should be noted that turbines on Fig. 5 (b) are presented in one space using equal scale. Figures 5 (a) and (c) shows closed-up meridional view of HPT and LPT using different scale. Real dimensions can be seen on the vertical and horizontal rulers on the left and the bottom sides of the pictures, respectively.

The obtained turbines have high performance indexes. The HTP gives 266.5 kW of mechanical power with efficiency of 0.8953; the LPT provides 406.5 kW of mechanical power with efficiency of 0.9249. It should be noted that replacement of turbines proposed in paper [3] with designed radial-outflow turbines would increase generated mechanical power of the ORC by 11.06%. Wherein, thermal efficiency of the cycle calculated as a relation between Net Power Production and Transferred Heat is equal to 0.1907.

All performance data and crucial dimensions are collected in Tables 2, 3.

3.2. Comparison of different turbine types

After the preliminary design phase, two radial-outflow turbines with high performance indexes were obtained. However, it is interesting to compare designed ROT with radial-inflow turbine (RIT) and axial one, which are more common nowadays.

For this purpose, axial high-pressure (HPT) and low-pressure (LPT) turbines were designed with the boundary conditions specified in Table 1. RIT for aforementioned boundary conditions were designed in paper [3]. In this article they are also presented and are turbines regarding to which performance comparison was carried out. Main characteristics of all aforementioned turbine types are collected in Tables 2, 3.

It should be noted that in common practice, axial turbines for ORC are limited to 3 stages due to mechanical or compactness restrictions. The same conclusion can be drawn regarding ROT. Despite this it should be noted that aerodynamic efficiency changes unequally for axial turbines and ROT with the changing of number of stages. In Fig.6 are comparative charts of how number of stages affects a turbine's total-to-static efficiency for axial turbine and ROT (both turbines were designed using HPT boundary conditions).

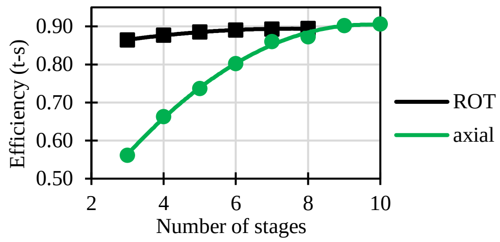

Fig. 6. Dependence of the efficiency on the turbine stage number for axial turbine and ROT

Obviously, the maximum efficiency for axial turbine's and ROT are almost the same. However, in order to achieve maximum efficiency, the axial turbine should be performed with the high number of stages. The ROT efficiency slightly changes with decreasing of number of stages. For example, for considered ROT, efficiency of the seven-staged design is 2.8% lower than of the three-staged design, while for axial turbine this difference achieves 30%. Such efficiency benefit is a big advantage of ROT, especially when the maximum number of stages is limited. In present paper we do not consider any particular restrictions: mechanical or compactness limitations. Therefore, further analysis is performed using only one design criterion – turbine aerodynamic efficiency.

Table 2. Performance data and crucial dimensions of the HPTs

Parameter	Unit	Axial	RIT	ROT
Internal t-s efficiency	–	0.9047	0.8518	0.8953
Power	kW	269.5	211.4	266.5
Shaft rotational speed	rpm	18305	40 000	27 129
Number of stages	–	10	1	5
Max Mach number	–	0.529	1.142	0.949
Min machine diameter	mm	62.48	45.2	56.3
Max machine diameter[1]	mm	114.7	106.1	114
Axial length	mm	86.1	37.7	12.8
Machine volume[2]	dm^3 (liter)	0.89	0.333	0.131

Table 2 shows that different HPT types have significant differences in efficiency. The highest efficiency (and power respectively) has the 10 staged axial turbine. Radial-outflow turbine efficiency is slightly lower because of the stage number limitation. Wherein, efficiency of both the axial and the radial-outflow turbines are superior far from RIT performance values. Due to reduced shaft rotational speed in ROT and axial turbine, maximum Mach number is also reduced, that is obviously beneficial to the flow in the flowpath.

[1] without volute
[2] approximate value calculated as multiplication of the circle area of maximal radial dimension to the axial length

Table 3. Performance data and crucial dimensions of the LPTs

Parameter	Unit	Axial	RIT	ROT
Internal t-s efficiency	–	0.9288	0.9133	0.9249
Power	kW	408.7	394.6	406.5
Shaft rotational speed	rpm	3000	7 914	3 000
Number of stages	–	13	1	9
Max Mach number	–	0.411	1.02	0.636
Min machine diameter	mm	195.9	137.8	292.9
Max machine diameter[1]	mm	405.1	459.2	588.6
Axial length	mm	630.5	184.2	38.52
Machine volume[2]	dm^3 (liter)	81.264	30.506	10.481

For the LPT, all three turbine types have substantially the same performance values (difference in efficiency is less than 1.5 %). However, the ROT and the axial turbine have a significant advantage – their rotational speed is 3000 rpm. This allows integration directly to a generator and avoids additional mechanical losses in the gearbox which now is not needed. Furthermore, ROT type has the smallest sizes for both the HPT and the LPT units. This fact reduces the required space for installation and material costs for manufacturing such turbine.

This analysis of the designed turbines has been made only with regard to the considered organic Rankine cycle and it may be not applicable to other cycles. Furthermore, as previously mentioned, ROT efficiency strongly depends on the used working fluid. Despite this, the data obtained in this research highlights the most important advantages and disadvantages of each turbine type:

a. RIT
- Easy to manufacture
- Wide application range
- Low efficiency

b. Axial
- The highest efficiency
- Low rotational speed
- The largest size

c. ROT
- High efficiency
- Reduced rotational speed (may be equal to generator's rotational speed)
- The smallest size
- Not applicable to all working fluids

4. Conclusions

- The proposed design technique can solve the optimization task at the stage of preliminary design. The unique feature of this technique is the uneven enthalpy drop distribution between turbine stages which allows to design radial-outflow turbine with highest possible efficiency.
- Two new turbines were designed for considered 3 MW WHR ORC. Their application will result in more than 11 % of mechanical power boost.

- For comparative analysis, three turbines of different types (axial, radial-inflow, radial-outflow) were designed with the same boundary conditions. Turbine performance comparison enables discovery of the advantages and disadvantages of each turbine type and as a result, recommendations can be made on their applicability.

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