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## Article

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Energy Reports

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*Suggested Citation:* Giovannelli, A.; Archilei, E. M.; Salvini, C. (2020) : Full-admission radial turbine for waste heat recovery Organic Rankine Cycles, Energy Reports, ISSN 2352-4847, Elsevier, Amsterdam, Vol. 6, Iss. 1, pp. 646-651,  
<https://doi.org/10.1016/j.egy.2019.09.043>

This Version is available at:

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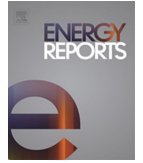
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6th International Conference on Energy and Environment Research, ICEER 2019, 22–25 July  
2019, Aveiro, Portugal

# Full-admission radial turbine for waste heat recovery organic Rankine cycles

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Received 29 July 2019; accepted 16 September 2019

Available online 9 October 2019

## Abstract

Organic Rankine Cycle (ORC) plants are interesting systems for power production by Waste Heat Recovery (WHR), although their application, especially for small/medium plants, could be not economically convenient. This is due to the inherent low thermodynamic cycle efficiency connected to the temperature of the heat source. Therefore, to avoid further penalizations, plant components (especially the turbine) should reach high performance both at nominal and off-design conditions. The paper deals with the design (from a 1-D to a fully 3-D level) of a full-admission radial-inflow turbine for a WHR ORC plant with a power output less than 50 kW. A parametric study was carried out to improve the turbine performance varying the most relevant geometric parameters, and the most promising geometry was analyzed in off-design conditions.

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Peer-review under responsibility of the scientific committee of the 6th International Conference on Energy and Environment Research, ICEER 2019.

*Keywords:* Organic Rankine Cycle (ORC); Radial turbine; Turbine design; CFD; Waste Heat Recovery (WHR)

## 1. Introduction

The application of energy saving techniques to every sector (e.g., building, industrial, transport) is considered a priority to reduce the effects of the human impact on the environment as reported by IEA [1], and both industrialized and emerging countries have fixed their targets (in some cases very challenging) at medium and long terms. Among such energy-saving techniques, Waste Heat Recovery (WHR) is one of the most interesting. Usually, waste heat can be utilized for power production or process heating. In the first instance, recovery plants based on Organic Rankine Cycles (ORCs) are a common solution. Even though ORC systems are considered a consolidated technology, their application is generally far to be economically convenient, especially for small/medium scale plants. It is due to the relatively low temperature of the heat source made available to the ORC system, which necessarily implies a low thermodynamic cycle efficiency. Therefore, to avoid further penalizations of the system, plant components (especially the expander) has to reach high performance both at nominal and off-design conditions. From the

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<https://doi.org/10.1016/j.egy.2019.09.043>

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thermodynamic point of view, ORC plants are usually equipped with a recuperator to allow an internal heat recovery and enhance the overall plant efficiency. Therefore, for the present study, a reference plant layout with a recovery system has been taken into consideration, as shown in Fig. 1. The main plant components are:

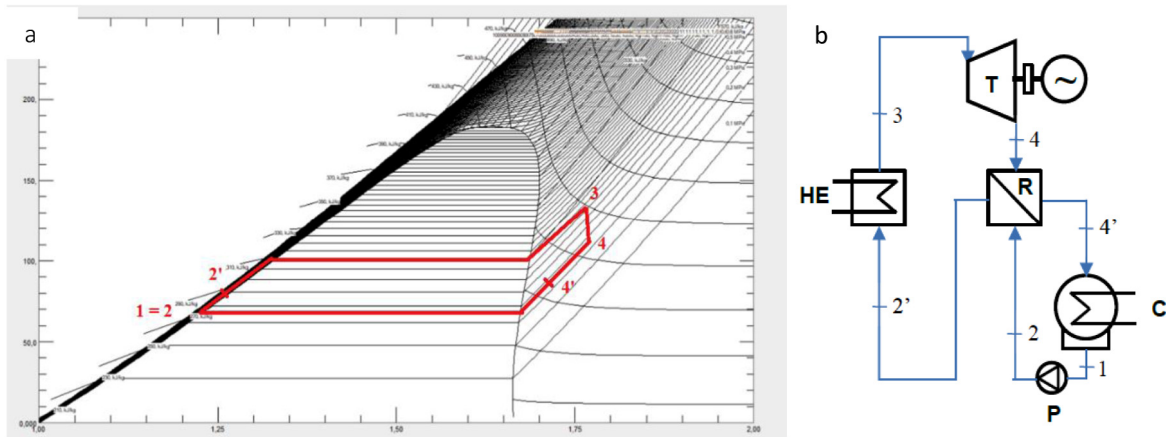


Fig. 1. (a) Thermodynamic reference cycle: T-S diagram; (b) reference plant scheme.

- A Heat Exchangers (HE), which inputs the thermal power coming from the WHR in the ORC plant. The working fluid evaporates completely during this transformation, and it can be also overheated;
- An Expander (T), which extracts the mechanical power from the plant and is connected with the electric generator;
- A Condenser (C), for the cooling and the condensation of the working fluid using the rejection of thermal power to an external heat sink;
- A Pump (P), which delivers the working fluid to the recuperator, increasing its pressure;
- The Recuperator (R), which pre-heats the WF before it enters in the evaporator using part of the heat rejected by the WF at the turbine exit.

For WHR applications, a maximum nominal temperature of 145 °C has been considered according to Quoilin et al. [2]. Based on the suggestions given in [3,4], and [5], R123 was selected as WF. Such fluid is a dry, pure substance developed in the past for refrigeration systems. Detailed thermodynamic properties are given by DuPont Suva refrigerants [6]. Moreover, it shows properties suitable for direct Rankine cycles at medium/low temperature. Once the WF was selected, the maximum cycle pressure is consequently established. For the present study, it is set at 17,56 bar. The condensation pressure is a compromise between the cycle optimization and the expander selection. For a 50 kWe plant, both radial turbines and volumetric machines like screw expanders could be adopted as reported by Rahbar et al. [7]. For the present study, a one-stage radial inflow turbine has been taken into consideration because of its robustness, cheapness, and its potential good performance at nominal and part-load conditions. The WF sound speed is quite low in the turbine reference operating range (around 115–130 m/s): this concern limits the nominal pressure at the machine exit in order to avoid transonic/supersonic flow conditions which would reduce both turbine fluid dynamic performance and operational flexibility. Therefore, the outlet pressure of 6 bar (corresponding to a condensation temperature of 88 °C) has been established. Consequently, fixing the electric power, a nominal mass flow rate of 3,37 kg/s has been evaluated. Under such thermodynamic boundaries, the ideal thermodynamic efficiency of the recuperated cycle is almost 10,5%. Thereby, it is clear that the turbine has to be designed carefully to reach good performance at nominal and off-design conditions. In the following Chapters, the turbine design steps from a 1-D to a 3-D level and the results of the sensitivity analysis on the most relevant geometric parameters are described.

## 2. Radial turbine designs

The development of a turbomachine for unconventional WFs, shows some concerns and challenging aspects, as reported in [8]. In this case, the R123 variation in density (and consequently in the volumetric flow) in the

expansion is relevant (about 3.2) if compared with small radial inflow turbines operated with conventional gases (e.g., air, exhausted gases). Moreover, sound speed and viscosity of reference are out of the common range. It means that the preliminary layout cannot be established completely by means of global correlations or charts based on adimensional parameters as in [9]. Therefore, to address the fluid dynamic machinery design, a preliminary layout can be achieved using 1-D and 2-D models (at the mean line) and then, a more detailed layout can be verified using 3-D Computational Fluid Dynamic (CFD) analyses. Some examples of the application of the design procedure applied to small radial turbomachines for refrigerant fluids are given by Dong et al. [10], Zheng et al. [11], Giovannelli and Archilei [12], and Giovannelli et al. [13].

Firstly, the preliminary design addressed the rotor, which is the most relevant component. Then the others (nozzle and exhaust diffuser) were designed. According to the simplified turbine scheme reported in Fig. 2, calculations involved averaged aerodynamic and geometric parameters at the mean-line of the stations located at inlet and exit of the main components. In particular, for the rotor sizing, the nominal speed (23 000 rpm) was set according to the expected enthalpy drop, the discharge spouting velocity, reasonable absolute and relative angles at inlet and exit sections and the coupling with a suitable electric generator. Reference values for such parameters are taken according to Aungier [14].

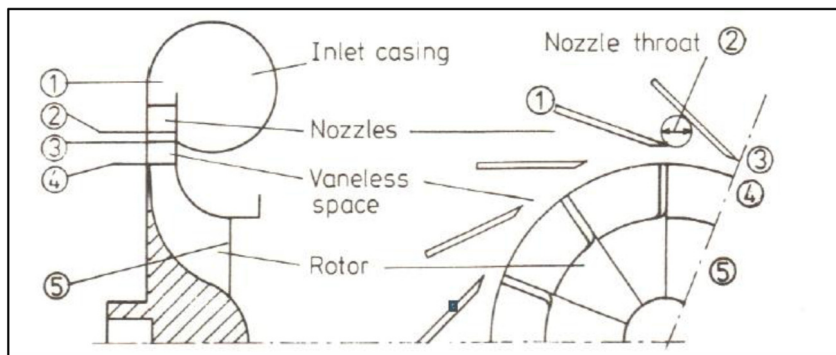


Fig. 2. Layout for an inward-flow radial turbine and main sections.

After that, a preliminary 3-D shape of the shrouded rotor was sized, considering, as a first attempt, an isentropic expansion. Such a preliminary rotor geometry is depicted in Fig. 3a. To analyzed in detail the behavior under nominal conditions, and achieved more reliable information on the component efficiency, a 3-D CFD quasi-steady-state analysis has been carried out.

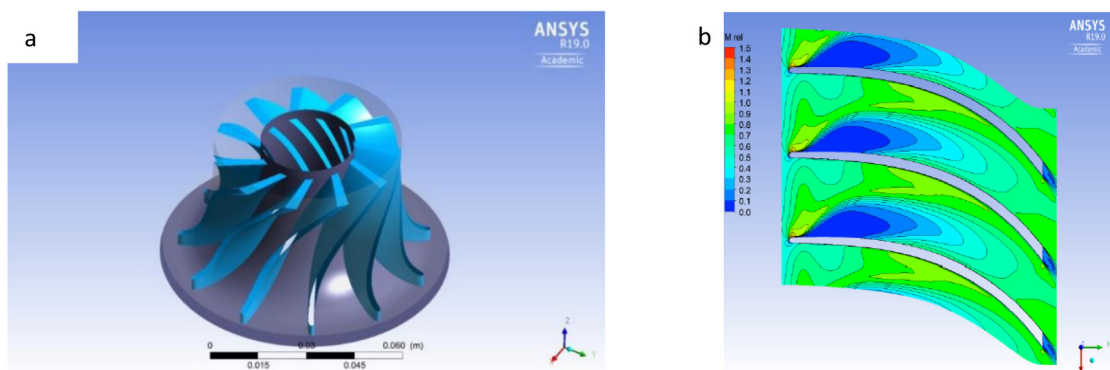


Fig. 3. (a) Preliminary rotor layout; (b) Relative Mach number at the mean-line for the nominal conditions.

Taking advantage of the axial symmetry, an H–O–H mesh for two rotor channels has been generated. A structured mesh of about 290 000 nodes for each channel has been set after a grid independency analysis. Overall mesh quality parameters (maximum and minimum face angles, edge length ratio, element volume size) have been checked to provide good accuracy. The commercial software ANSYS-CFX has been used to perform the simulations. The standard  $k-\varepsilon$  model with scalable wall function was selected and, to describe R123 properties, the real gas

cubic equation Aungier–Redlich–Kwong Model by Aungier [15] was chosen. Such a model provides a reasonable prediction of the real fluid behavior in the cases of interest. As reported in Fig. 3b, the preliminary design achieved using conventional simplified models did not lead to acceptable results: a large flow separation region can be observed on the suction side due to a high incidence angle of the flow on the blade leading edge combined with an incorrect blade shape. Such phenomena have been attributed to the rough evaluation of the local flow density under simplified conditions. Therefore, the rotor was re-designed in detail by means of an iterative process between the gas-path design and the 3-D flow field analysis. Several geometric parameters have been taken into account: hub and shroud shapes, number, thickness and height of blades, absolute angle and blade angle at the rotor inlet section, relative blade angle distribution along the path and leading edge geometry.

The iterative procedure led to an enhanced rotor geometry reported in Fig. 4a. As depicted in Fig. 4b, the stall on the suction side has been contained in a restricted area corresponding to the elbow. The internal rotor efficiency at nominal conditions raised from 68% (preliminary layout) to almost 90% (final layout).

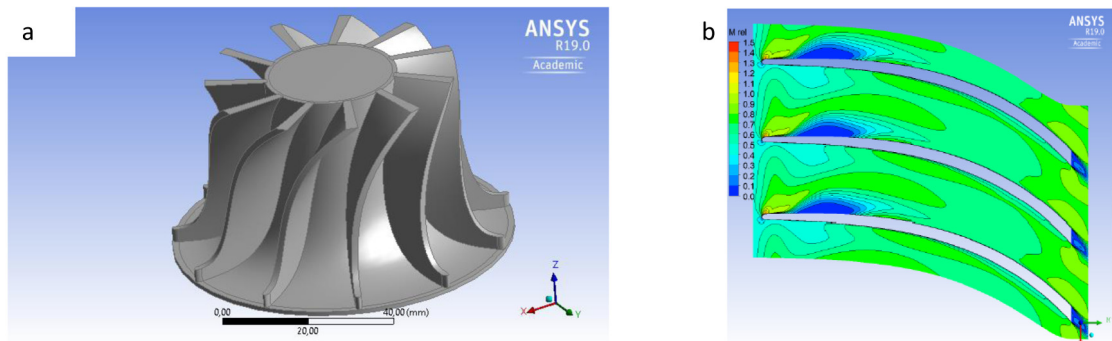


Fig. 4. (a) Enhanced rotor layout; (b) Relative Mach number at the mean-line for the nominal conditions.

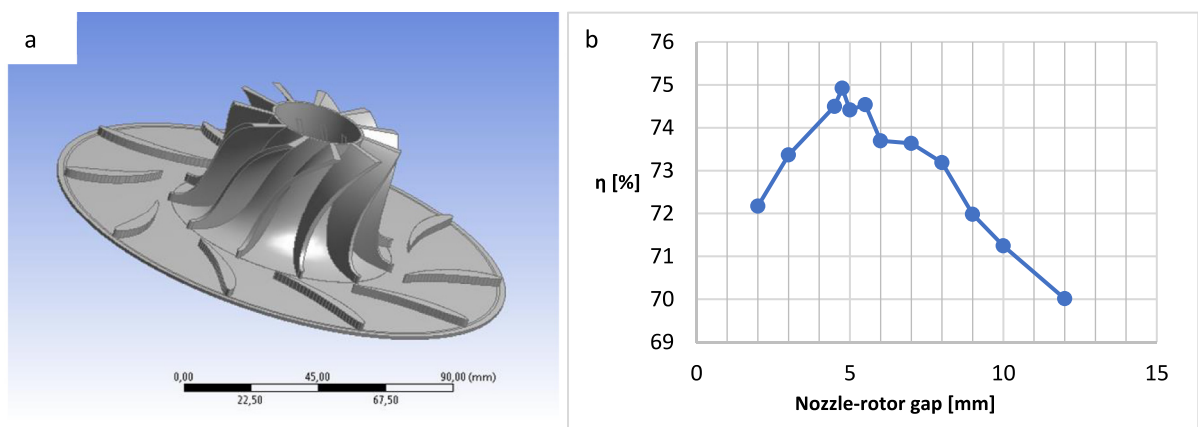


Fig. 5. (a) Enhanced final complete turbine geometry; (b) Nominal turbine internal efficiency varying the gap between nozzles and rotor.

After having completed the design of the rotor, the nozzle has been assessed accordingly. It needs to accelerate the flow in the proper direction before entering the rotor. In this study, the designed nozzle has been considered fixed to reduce the complexity and costs of the machine. In Fig. 5, the complete geometry (nozzles and rotor) is depicted. The designed rotor is shrouded to reduce volumetric losses; nonetheless, in Fig. 4, it is represented without the shroud to highlight the rotor channels. A sensitivity analysis has been carried out varying the gap between the nozzles exit section and the rotor inlet section. As reported in Fig. 5b at nominal conditions, the turbine shows a maximum internal efficiency (about 15%) for a gap of 4.5 mm. Performance drop at lower gaps because of the flow distortion provoked by the nozzle trailing edge wakes. On the other hand, at higher gaps, despite the more uniform flow conditions at the rotor inlet, the efficiency decreases because the flow is no longer well guided by the bladed

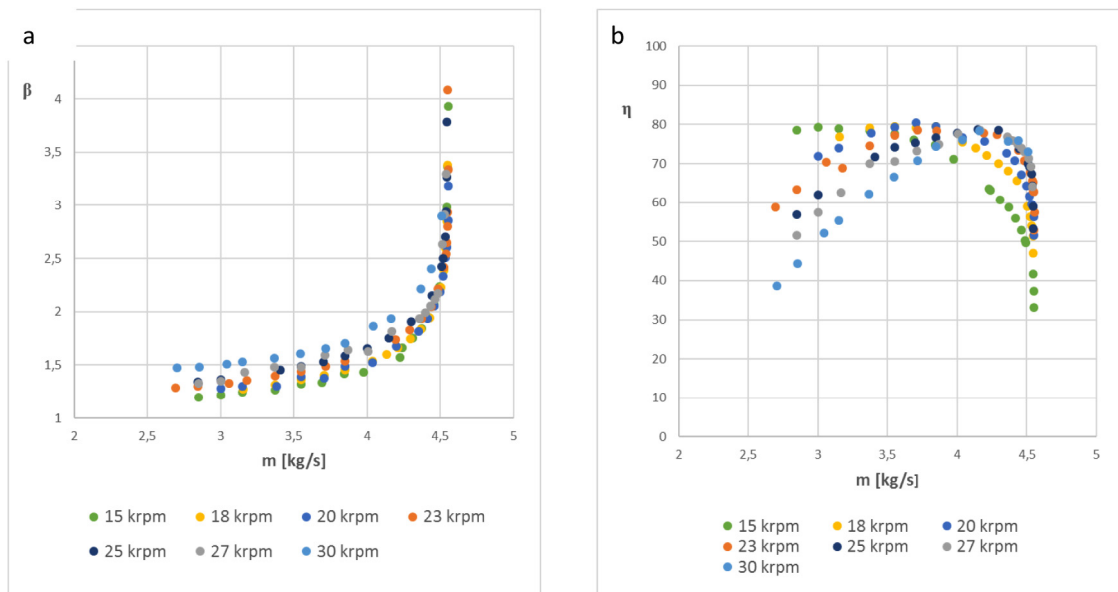
nozzle. To perform such CFD simulations, the tool ANSYS-CFX has been used. Two nozzle vanes and three rotor blade channels have been considered taking advantage of the geometry symmetry. The “Frozen Rotor” model has been adopted at the interface between nozzle and rotor. For each nozzle vane, a grid of about 190 000 nodes has been set.

3-D CFD simulations have been performed using the overall turbine geometry at off-design conditions for several rotational speeds (from 15 000 to 30 000 rpm). Results achieved for internal power and total-to-static efficiency are presented in Fig. 6, while the final geometry details are given in Table 1.

**Table 1.** Main geometric results for the enhanced rotor.

An example of a column heading	Inlet section	Exit section
Blade height [mm]	4	14.5
Diameter [mm]	80	45
Blade thickness [mm]		1.5
Number of blades		11
Blade angle (mean-line) [°] <sup>a</sup>	90	50

<sup>a</sup>Taken from the peripheral direction.



**Fig. 6.** Turbine maps (a) pressure ratio vs. mass flow rate; (b) Total-to-static efficiency vs. mass flow rate.

### 3. Conclusions

The paper presents the design of a radial-inflow turbine to be applied in a 50 kWe ORC plant for WHR. Main fluid dynamic concerns related to the development of an unconventional radial turbine have been mentioned and taken into consideration during the design process. The machinery layout has been modified iteratively by means of detailed information given by 3-D quasi-steady-state CFD viscous simulations. Results varying the rotor most relevant geometric parameter and the gap between the vaned nozzle and the rotor have been presented and discussed. Moreover, off-design turbine maps have been carried out. It can be noted that the turbine shows good overall performance in a wide operational range without highlighting the requirement of a vaned nozzle with variable stagger angle.

### Acknowledgments

Authors acknowledge HiRef S.p.A, Italy and University of Roma Tre, Italy for their kind support.

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