COMBINED THERMODYNAMIC AND TURBINE DESIGN ANALYSIS OF SMALL CAPACITY WASTE HEAT RECOVERY ORC

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ABSTRACT

In this paper, the design of small-capacity ORC turbines for a waste heat recovery application is studied and discussed. A turbine design tool was coupled with a thermodynamic analysis tool in order to evaluate the effect of different working fluids and process parameters, not only by taking into account the thermodynamic aspects of the process design, but also evaluating the availability to design turbines with a relatively high efficiency and feasible geometry. The studied turbine type is a radial inflow turbine since radial turbines represent relatively simple geometries when compared to multistage configurations, can have a high expansion ratio over a single stage, and have a better efficiency at a low power capacity than the axial counterparts. The results indicate that the main difficulties in the design of small capacity ORC turbines are related to high rotational speeds, small dimensions, and large blade height ratios. In addition, the use of single stage turbines leads to highly supersonic flow in the stator even when adopting low or moderate flow velocities. The results of this study highlight the importance of combining both the thermodynamic process design and the turbine design when evaluating suitable working fluids and operational parameters.

1. INTRODUCTION

Organic Rankine Cycles (ORC) are a widely implemented technology in geothermal, biomass, and in different waste heat recovery applications. The use of organic working fluids allows to design small-scale power systems that are capable of utilizing efficiently low temperature heat sources. The current ORC applications and the most important design aspects of ORCs are presented and discussed e.g. in (Colonna et al., 2015). In small power output ORCs (about 10-50 kW) a volumetric expander, namely screw or scroll expanders, are typically preferred instead of a turbine whereas a turbine is typically used as an expansion machine in larger scale ORC applications (Colonna et al., 2015). The limitations of using volumetric expanders are related to the lower achievable expansion ratios over an expander when compared to the use of turbines and thus, the use of volumetric expanders disables the use of high molecular weight and high critical temperature fluids characterized by high expansion ratios and high thermodynamic efficiencies. Therefore, the use of turbines could enable the design of high efficiency small-scale ORCs, especially for high temperature applications.

The turbine types used in the ORCs are typically axial or radial turbines (Colonna et al., 2015). Studies on using multistage radial outflow turbines have been carried out in the recent years (Pini et al., 2013). The expansion ratio over the turbine can be very high, especially in high-temperature ORCs, which can result in high Mach numbers and significantly large rotor blade height ratio between the rotor outlet and inlet (Macchi, 1977; van Buijtenen et al., 2003; Uusitalo et al., 2013). In addition, the real gas effects in the expansion become significant, especially when fluids having a high molecular complexity are used and when the expansion occurs near the critical point of the fluid (Harinck et al., 2009; Guardone et al., 2013). In many ORC turbines, the flow is supersonic since the organic fluids have a low speed of sound. This causes losses related to the occurrence of shock waves in the flow passages. The design and flow analysis of supersonic ORC turbine stators have been presented for radial turbines e.g. in (Harinck et al., 2010, Uusitalo et al., 2014) and for axial turbine stator e.g. in (Colonna et al., 2006). Geometry optimization methods by using automated CFD-design have been developed in the recent years in order to further improve the performance of supersonic ORC turbines (Harinck et al. 2013). The results by Harinck et al. (2013) indicated that the losses caused by the occurrence of shock waves can be reduced significantly by bending and turning the stator flow channels instead of using straight nozzles.

Few experimental works on small-scale ORC turbines are available in the literature. Verneau (1987) presented the experimental results for two high pressure ratio ORC turbines, having a power output of 50 kW and using FC 75 as the working fluid. The first turbine design was a supersonic two-stage axial turbine and the second design was a single-stage axial turbine. The pressure ratios of the designed turbines were about 150. The experiments showed that a relatively high turbine efficiency in a range of 70 to 80 % was achieved for both the two-stage and single-stage turbine designs. An experimental study on a small-scale ORC having a high speed radial turbine was presented by Kang (2012). The turbine and generator were assembled to a single shaft having a rotational speed of 63 000 rpm. The system used R245fa as the working fluid and the evaporation temperature was in a range from 70 ^{o}C to 90 ^{o}C. The measured power output was 32.7 kW and the turbine efficiency of 78.7 % was achieved. Additionally, Klonowicz et al. (2014) presented experimental results of a low-temperature ORC using R227ea. The system had a 10 kW hermetic turbogenerator and the designed turbine was a partial admission axial turbine. The electric efficiency of the turbogenerator of less than 60 % was measured including the losses in the turbine and in the generator.

Despite several studies on high-expansion ratio ORC turbines, guidelines to design such expanders, especially for small-scale systems are lacking and only few studies have presented experimental results on the performance of small ORC turbines. In addition, a large number of publications have discussed the working fluid selection from the thermodynamic point of view, but totally neglecting the effect of the working fluid on the turbine design. Maraver et al. (2014) included a preliminary turbine size evaluation for different working fluids in a thermodynamic study by using a size parameter (SP), but no further assessment on the turbine rotational speeds, geometry, or Mach numbers were included. This paper presents a method of combining the thermodynamic process analysis and preliminary turbine design tool to evaluate suitable working fluids and cycle design parameters for a small-scale ORC from the point of view of the turbine design. In addition, the most critical aspects of designing radial turbines for small-scale ORCs are highlighted and discussed. The studied application is exhaust heat recovery of small-scale gas turbines.

2. DESIGN METHODS

The design of an ORC process utilizing exhaust gas heat is studied by using exhaust gas values typical for small-scale gas turbines(Invernizzi et al., 2007). The exhaust gas temperature at the ORC evaporator inlet is 300 ^{o}C, and the mass flow rate of the exhaust gas was set to 1 kg/s. The exhaust gas temperature at the evaporator outlet was varied in the analysis in order to study the effect of the exhauts gas outlet temperature on the cycle power output and turbine design. A commercial thermodynamic library Refprop was used for calculating the fluid properties. The studied fluids are hydrocarbons toluene, cyclohexane, and pentane; siloxanes MDM and MM; and fluorocarbons R245fa and R365mfc. The main thermodynamic properties of the studied fluids are presented in Table 1.

In the thermodynamic analysis of the process, the evaporation pressure of the working fluid was optimized to reach a minimum pinch-point temperature difference of 15 ^{o}C in the evaporator with the given heat source values. If the temperature difference in the evaporator remained sufficient, a limit of $p_{\mathrm{ev}}/p_{\mathrm{crit}}$ = 0.95 was used for the evaporation pressure. A condensing temperature of 60 ^{o}C was used, the cycle included a recuperator and all the studied fluids were superheated by 10 ^{o}C, since a flow through type of evaporator was considered. Turbine efficiency of 80 %, generator efficiency of 95 %, efficiency

Table 1: Studied working fluids and main thermodynamic properties.

fluid	T_{crit}, $[^oC]$	p_{crit}, [bar]	MW, [kg/kmol]	p_{cond} (@ 60 ^oC), [bar]
toluene	318.6	41.3	92.1	0.2
cyclohexane	280.5	40.75	84.2	0.5
pentane	196.6	33.7	72.1	2.1
MDM	290.9	14.2	236.5	0.04
MM	245.6	19.4	162.4	0.3
R245fa	154.0	36.5	134.0	4.6
R365mfc	186.9	32.7	148.1	2.0

of the frequency converter of 97 %, and the efficiency of feed pump of 60 % were used in the analysis.

The design tool of radial turbines calculates the main dimensions and velocity triangles of the turbine. The turbine inlet and the outlet conditions were obtained as a result from the process thermodynamic design. The evaluation of the turbine rotational speed, n, was carried out by using the non-dimensional parameter, turbine specific speed N_{s} defined as,

$$N_s = \frac{\omega {q_v}^{0.5}}{{\Delta h_s}^{0.75}}. \tag{1}$$

According to the performance diagram for radial turbines operating with compressible fluids, the highest efficiencies are obtained with specific speeds in a range from 0.4 to 0.8 (Balje, 1981). The intermediate static pressure between the stator and rotor was selected by using the degree of reaction which is defined by dividing the enthalpy change in the turbine rotor by the total enthalpy change in the turbine stator and rotor

$$r = \frac{\Delta h_{rot}}{\Delta h_{tot}}. \tag{2}$$

The rotor outlet blade tip to rotor inlet diameter ratio D_{2t}/D_1, the blade hub-to-tip diameter ratio D_{2h}/D_{2t} at the outlet as well as the stator exit flow angle α_1 were evaluated by following the guidelines of (Rohlik, 1972). The velocity calculations were based on the continuity equation and on the Euler turbomachine equation. The velocity vectors were solved at the rotor inlet and at the rotor outlet. An example of the turbine velocity triangle at the rotor inlet is presented in Fig. 1. The tangential component of the absolute velocity, c_u, was used in monitoring the shape of the velocity triangle at the rotor inlet by using a velocity ratio c_u/u while the radial component of the absolute velocity, c_r, was used in the continuity equation to calculate the blade height at the rotor inlet. The stator efficiency of 82 % was used in the calculations to estimate the static enthalpy at the stator outlet. The turbines were designed to have the rotor discharge in axial direction, and thus, the tangential component of the absolute velocity at the rotor outlet c_{u2} = 0. The calculations of the turbine geometry were based on the assumption that the flow velocity at the rotor discharge is relatively low and thus, an assumption of $h_{2,\mathrm{st}} \approx h_{2,\mathrm{tot}}$ was used and no analysis for the turbine diffuser was included.

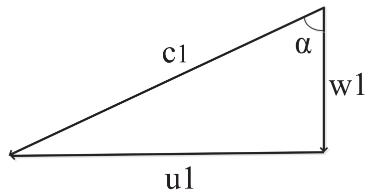


Figure 1: An example of a velocity triangle at the rotor inlet, having a radial relative velocity (w_{u1} = 0).

It should be noted that the results of the process calculations were based on the 80 % turbine efficiency with all the studied fluids and process design parameters. This kind of approach can be well justified in the preliminary evaluation of turbine dimensions and rotational speeds, but it should be noted that the turbine efficiency has an effect on the size of the rotor wheel and on the optimal turbine rotational speed. Thus, a more accurate turbine design would be achieved in a case if the turbine efficiency used in the process design were iteratively changed based on the obtained turbine dimensions and Mach number at the stator outlet. However, this would require detailed numerical simulations of the turbine since there is very limited data available about the losses and applicable loss correlations for radial ORC turbines.

3. RESULTS

3.1 Comparison to radial turbine designs available in the literature

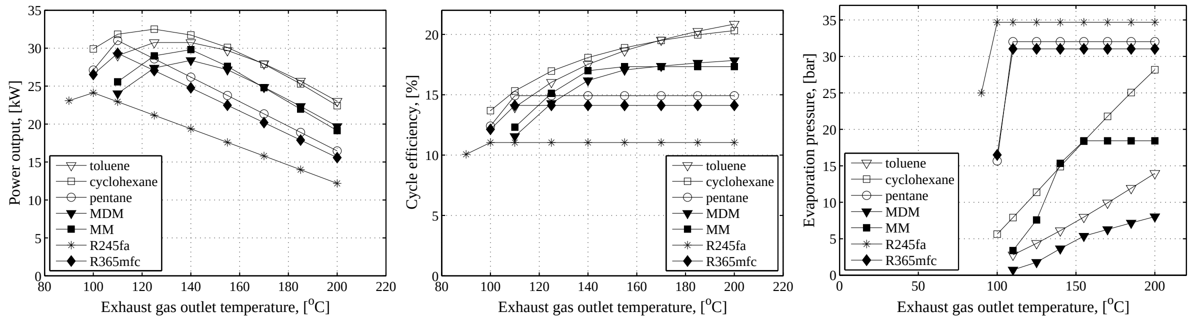
A comparison was made between the turbine geometries designed with the code used in this study and ORC radial turbine designs available in the literature to verify the design method. The working fluid mass flow rate, the turbine inlet state, the turbine efficiency and the turbine outlet pressure, as well as the degree of reaction and the specific speed if available, were set to the same values as presented in (Kang, 2012; van Buijtenen et al, 2003). The turbine isentropic efficiency of 75 % was used in the comparison of (Kang, 2012) and 80 % in the comparison of (van Buijtenen et al., 2003). The absolute flow angle at the stator outlet of 75 ^{o} was used for the both turbines designs. The results of the comparison are presented in Table 2. In general, based on the results of the comparison the developed turbine design code has a good agreement in the turbine diameter and rotational speed with the radial ORC turbine designs selected for the comparison. The found agreement shows the reliability of the design method and gives a good foundation for the analysis in the following sections.

Table 2: Comparison of the turbine rotor diameter and rotational speed between Turbine 1 (Kang, 2012), Turbine 2 (van Buijtenen et al., 2003), and the used turbine design code.

	fluid	P_t, kW	q_m, kg/s	N_s, -	r, -	$p_{t,in}$, bar	$T_{t,in}$, ^oC	$p_{t,out}$, bar	D_{rot}, mm	n_{rot}, rpm
Turbine 1	R245fa	$\approx$ 30.0	1.58	na	na	7.3	80	1.78	125	20 000
comparison	R245fa	31.0	1.58	0.45	0.47	7.3	80	1.78	124.1	21 597
Turbine 2	toluene	$\approx$ 200	1.24	0.44	0.26	32.3	325	0.27	224	28 300
comparison	toluene	191.6	1.24	0.44	0.26	32.3	325	0.27	227.5	28 081

3.2 Results with different working fluids

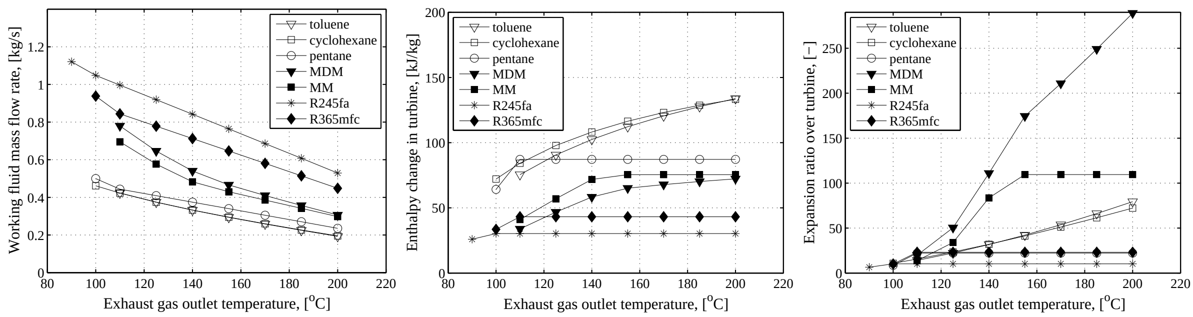
The process and turbine design results with different working fluids are presented in this section. The results presented in this section were calculated for turbines having the specific speed of 0.5 and the degree of reaction of about 0.5. The results of the cycle power output, the cycle efficiency and the evaporation pressure as a function of the exhaust gas temperature at the evaporator outlet are presented in Fig. 2a, Fig. 2b and Fig. 2c. The results for the power output show that the studied hydrocarbons represent the highest simulated cycle power output, above 30 kW, and the fluorocarbon R245fa represents the lowest power outputs, especially with the high exhaust gas temperatures at the evaporator outlet. The fluids with the highest critical temperatures represent the highest values for the power output with the higher exhaust gas temperatures at the evaporator outlet when compared to the fluids with the lowest critical temperatures. The highest cycle efficiencies are reached with hydrocarbons toluene and cyclohexane, and the lowest efficiencies with R245fa and R365mfc. It should be noted that these power output and efficiency results are based on the use of constant turbine isentropic efficiency with all the studied fluids and cycle parameters. Are more detailed analysis would require the turbine efficiency to be iteratively changed according to the obtained turbine geometries. The evaporation pressure limit of $p_{\mathrm{ev}}/p_{\mathrm{crit}}$ = 0.95 is reached with fluids having the lowest critical temperatures, namely R245fa, R365mfc, pentane, and MM, since the temperature difference between the working fluid and the exhaust gas remains sufficient. Evaporation pressures well below the critical pressures are adopted with each fluid when the lowest exhaust gas outlet temperatures are considered in order to maintain a sufficient temperature difference of 15 ^{o}C in the evaporator.



(a) **(b)** **(c)**

Figure 2: Results of a) power output, b) cycle efficiency, and c) evaporation pressure as a function of exhaust gas temperature at the evaporator outlet.

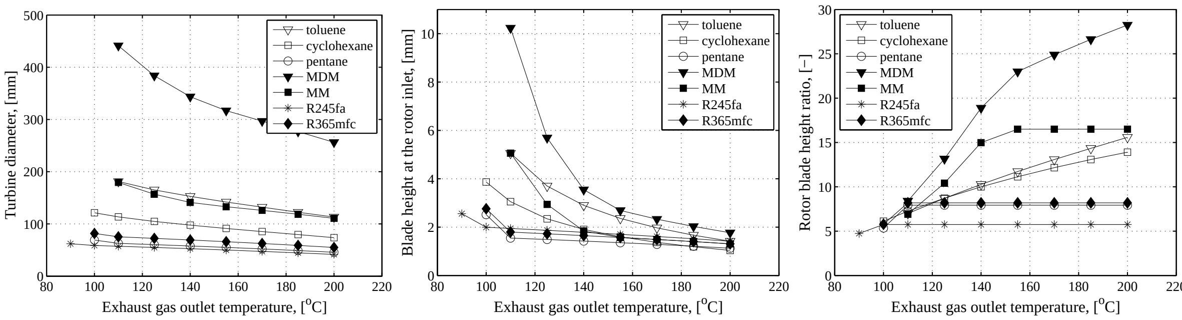
The working fluid mass flow rate is presented in Fig. 3a, the enthalpy change in the turbine in Fig. 3b, and the expansion ratio over the turbine in Fig. 3c as a function of the exhaust gas outlet temperature. The studied fluorocarbons, R245fa and R365mfc, represent the highest mass flow rates in the process, while the studied hydrocarbons, toluene, cyclohexane, and pentane represent the lowest mass flow rates. The studied hydrocarbons represent the highest enthalpy drop in the turbine and the studied fluorocarbons represent the lowest enthalpy drop. The studied siloxanes and high critical temperature hydrocarbons represent significantly higher expansion ratio over the turbine than the other fluids, especially when the exhaust gas temperature at the evaporator outlet is high. This can be explained by the low condensing pressure of these fluids and if the exhaust gas temperature is high at the evaporator outlet a high evaporation pressure was obtained, resulting in a large expansion ratio over the turbine. The studied fluorocarbons and pentane, which are the fluids having the lowest critical temperatures, represent the lowest expansion ratios over the turbine.



(a) **(b)** **(c)**

Figure 3: Results of a) working fluid mass flow rate, b) enthalpy change over the turbine, and c) expansion ratio over the turbine as a function of exhaust gas temperature at the evaporator outlet.

The calculated turbine diameters are presented in Fig. 4a, the blade height at the rotor inlet in Fig. 4b, and the rotor outlet-to-inlet blade height ratio in Fig. 4c. Based on the results, the fluids having the lowest critical temperatures presents the smallest turbine wheels and small blade heights at the turbine rotor inlet. The studied siloxanes and the hydrocarbons with the highest critical temperatures, toluene and cyclohexane, represent the largest rotor wheels and highest blade heights at the rotor inlet, especially when a low exhaust gas outlet temperature is used. These fluids also represent significantly large rotor blade height ratios, especially with the high exhaust gas outlet temperatures. This can be mainly explained by the high expansion ratio over the turbine with these fluids, which requires a large change in the flow area over the turbine wheel. The blade heights at the rotor inlet are significantly small ranging from 1 mm to 2 mm when a high exhaust gas outlet temperature is adopted. The small blade height is estimated to cause turbine efficiency reductions due to the high tip clearance losses and relatively thick boundary layers when compared to the height of the flow channel, as well as difficulties in manufacturing the turbine wheel. On the other hand, the effect of the tip clearance at the rotor inlet on the radial turbine efficiency has been reported to be less significant when compared to significance of tip clearance loss in axial turbines (Dambach et al., 1999). The large rotor blade height ratio might cause flow separation in the rotor and thus, reducing the turbine efficiency.



(a) **(b)** **(c)**

Figure 4: Results of turbine design values as a function of exhaust gas temperature at the evaporator outlet: a) rotor diameter, b) blade height at the rotor inlet, and c) rotor blade height ratio.

The Mach number at the stator outlet is presented in Fig. 5a, the peripheral speed at the turbine rotor inlet in Fig. 5b, and the turbine rotational speed in Fig. 5c. Fig 5a shows that all the studied fluids have a supersonic flow at the turbine stator outlet. The studied siloxanes and the hydrocarbons, toluene and cyclohexane, represent the highest Mach numbers at the stator outlet while the fluids with the lowest critical temperatures, namely R245fa, R365mfc, and pentane represent the lowest Mach numbers. The highly supersonic flow requires the use of accurate design methods for the stator flow channel in order to reduce the losses. In addition, the losses related to the stator-rotor interaction and to the reflection of shock waves from the rotor blades have been identified as a significant source of losses in this type of turbines (Rinaldi et al., 2013). The studied hydrocarbons represent the highest peripheral speed at the rotor inlet and the studied fluorocarbons represent the lowest values of peripheral speed. The high peripheral speed causes higher stresses for the turbine wheel, which should be taken into account in the mechanical design of the turbine wheel. The highest turbine rotational speed is achieved with pentane and the lowest rotational speed is achieved with the siloxane MDM. The high rotational speed increases the turbine shaft mechanical and windage losses and sets demands for the bearing design.

In general, the results indicate that by designing the cycle for low exhaust gas outlet temperatures and thus, resulting in a low evaporation pressure, several benefits can be achieved from the point of view of the turbine design. The working fluid mass flow rate and the blade height at the rotor inlet are higher and the turbine wheel is larger. In addition, the rotor blade height ratio, expansion ratio over the turbine, rotational speed and the Mach number at the stator outlet are lower when compared to a cycle designed with a higher evaporation pressure.

3.3 Effect of specific speed on turbine design

The effect of the turbine specific speed on the turbine design is presented and discussed in the following. The working fluid is toluene and the degree of reaction is about 0.5 in the calculated cases. The effect of the turbine specific speed on the rotor diameter is presented in Fig. 6a and on the turbine rotational

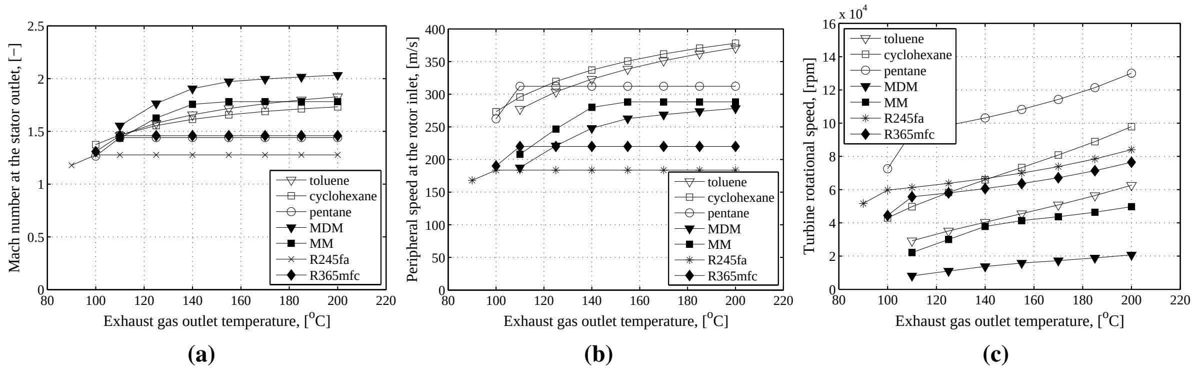


Figure 5: Results of a) Mach number at the stator outlet, b) peripheral speed at the turbine rotor inlet, and c) turbine rotational speed.

speed in Fig. 6b. The effect of the turbine specific speed on the blade height at the rotor inlet is presented Fig. 6c, and on the turbine rotor blade height ratio in Fig. 6d.The results presented in Figs. 6a-d, show

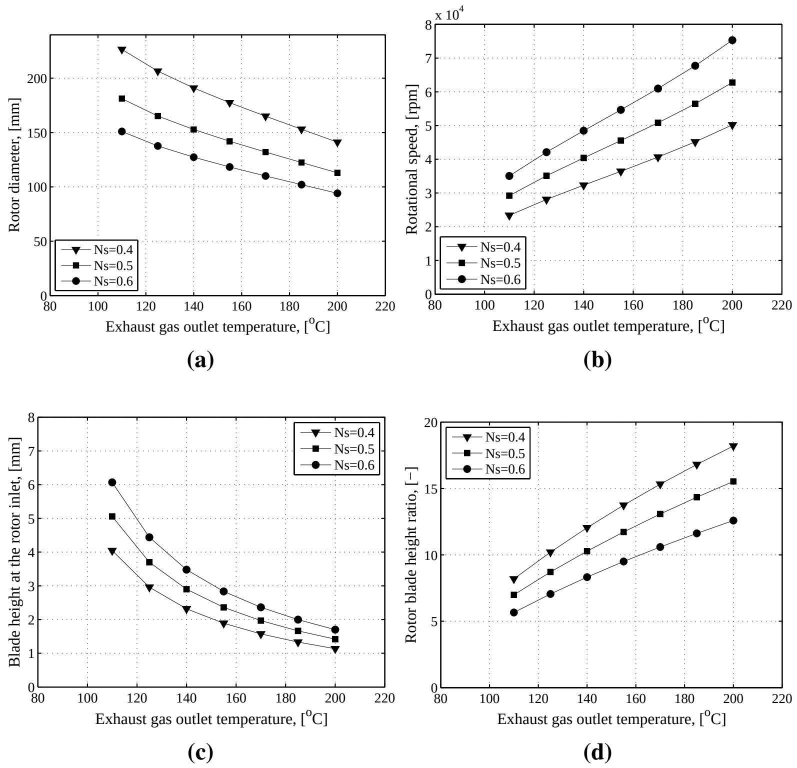


Figure 6: The effect of turbine specific speed on a) rotor diameter, b) rotational speed, c) blade height at the rotor inlet, and d) rotor blade height ratio. The working fluid is toluene.

that by selecting a low specific speed the turbine wheel is larger and the rotational speed is lower, when compared to a turbine design with a high specific speed. Thus, a turbine with a low specific speed can be considered if the turbine tends to be fast rotating and having a small diameter with a high specific speed. On the other hand, the results indicate that the blade height at the rotor inlet is smaller and the rotor blade height ratio is higher, possibly resulting to lower turbine efficiency when a low specific speed is selected, when compared to a turbine designed for higher specific speed.

3.4 Effect of degree of reaction on turbine design

The effect of the turbine's degree of reaction on the turbine design is presented and discussed in this section. The working fluid is toluene and the specific speed is 0.5 in the calculated cases. The effect of the degree of reaction on the rotor diameter, on the blade height at the rotor inlet, on the turbine rotor blade height ratio and on the Mach number at the stator outlet is presented in Fig. 7a, 7b, in Fig. 7c and 7d, respectively.The results presented in Figs. 7a-d, show that by selecting a low degree of reaction

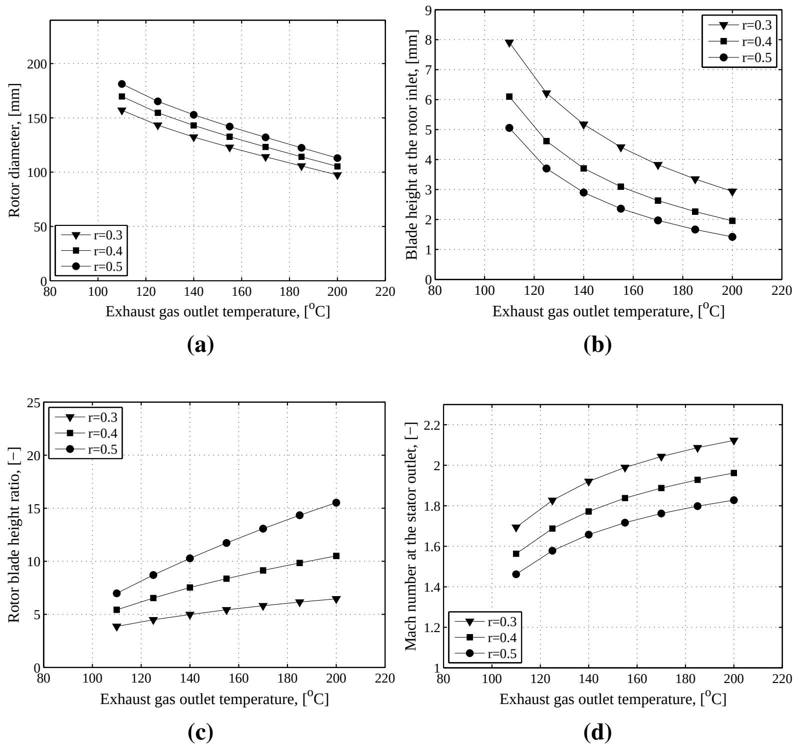


Figure 7: The effect of the degree of reaction on a) rotor diameter, b) blade height at the rotor inlet, c) rotor blade height ratio, and d) Mach number at the stator outlet. The working fluid is toluene.

the rotor wheel is smaller and the blade height at rotor inlet is higher when compared to a higher value of degree of reaction. In addition, the rotor blade height ratio is significantly reduced by selecting a low degree of reaction. However, the Mach number at the stator outlet is higher when a low degree of reaction is adopted. This can be explained by the fact that the pressure ratio over the turbine stator is high when a low degree of reaction is selected resulting in a high stator outlet velocity. In addition, the selection of low degree of reaction increases the velocity ratio c_{u1}/u_1 and there is a risk of a flow separation at the rotor inlet caused by a large incidence angle of the relative velocity if radial blades at the rotor inlet are considered. Thus, the use of bent blades at the rotor inlet, as were used e.g. in the turbine presented in (van Buijtenen et al. 2003), could be considered if a low degree of reaction is selected.

4. CONCLUSIONS

In this study, a simplified design tool of a radial turbine was coupled with a thermodynamic analysis tool to evaluate and compare different working fluids, not only from the thermodynamic point of view, but taking into account the turbine design considerations as well. Based on the results the selection of working fluid highly influences not only the thermodynamic performance of the cycle, but has a significant impact on the turbine dimensions and rotational speed as well. In general, the design of small-scale ORC turbines is difficult because the turbine wheels tend to be small and fast rotating, and represent small blade heights at the rotor inlet. The largest turbine wheels and the lowest rotational speeds were obtained with fluids having the highest critical temperatures. However, the results indicate that the use of a fluid with a high critical temperature leads to a high expansion ratio over the turbine and thus, represents highly supersonic flow at the stator outlet and large rotor blade height ratio, which is estimated to reduce the achievable turbine efficiency. The results indicate that by designing the cycle for a low evaporation pressure several benefits can be achieved from the turbine design point of view. The working fluid mass flow rate and the blade height at the rotor inlet is higher and the turbine wheel is larger. In addition, the rotor blade height ratio, expansion ratio over the turbine, the rotational speed and the Mach number at the stator outlet are lower when compared to a cycle designed for a higher evaporation pressure. Thus, if a small-scale ORC adopting a single-stage turbine is considered, it might be beneficial to design the cycle for a low evaporation pressure despite the reduction in the cycle efficiency. It was also observed that the turbine dimensions are very sensitive on the choice of turbine specific speed and degree of reaction. Loss correlations for high expansion ratio radial turbines for ORC applications should be created and impelemented in the future. The authors suggest that more experimental work on ORC turbines should be carried out to provide knowledge on the performance and feasibility of small-scale ORC systems based on turbine technology.

NOMENCLATURE

c	absolute velocity	(m/s)
h	specific enthalpy	(kJ/kg)
P	power	(kW)
p	pressure	(bar)
q_m	mass flow rate	(kg/s)
q_v	volumetric flow rate	(m^3/s)
T	temperature	(°C)
c	absolute flow velocity	(m/s)
N_s	specific speed	(-)
n	rotational speed	(rpm,1/s)
w	relative velocity	(m/s)
u	tangential/peripheral velocity	(m/s)
D	diameter	(m)
r	degree of reaction	(-)
α	absolute flow angle	(deg)
ω	angular speed	(rad/s)
Subscript		
ev	evaporation	
s	isentropic	
st	static	
rot	rotor	
t	turbine/blade tip	
h	blade hub	
0	turbine inlet	
1	rotor inlet/stator outlet	
2	rotor outlet	
u	tangential component	
r	radial component	

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