

Design of Radial Turbo-Expanders for Small Organic Rankine Cycle System

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Abstract. This paper discusses the design of radial turbo-expanders for ORC systems. Firstly, the rotor blades were design and the geometry and the perfromance were calculated using several working fluid such as R134a, R143a, R245fa, n-Pentane, and R123. Then, a numerical study was carried out in the fluid flow area with R134a and R123 as the working fluid. Analyses were performed using Computational Fluid Dynamics (CFD) ANSYS CFX on two real gas models, with the k-epsilon and SST (*shear stress transport*) turbulence models. The results analysis shows the distribution of Mach number, pressure, velocity and temperature along the rotor blade of the radial turbo-expanders and estimation of performance at various operating conditions. CFD analysis show that if the flow area divided into 250,000 grid mesh, and using real gas model SST at steady state condition, 0.4 kg/s of mass flow rate, 15,000 rpm rotor speed, 5 bar inlet pressure, and 373K inlet temperature, the turbo expander produces 6.7 kW, and 5.5 kW of power when using R134a and R123 respectively.

1. Introduction

Climate change concerns coupled with high oil prices are driving research and development on renewable energies such as solar and geothermal energy. The Organic rankine Cycle (ORC) uses organic fluid as the working fluid to provide higher thermal cycle efficiency compared to the conventional steam Rankine cycle at resource temperatures below 300°C. ORC has been studied as the utilization of waste heat recovery [1][2], solar energy [3], the combination of heat and power (CHP) [4], geothermal [5], and heat recovery from the exhaust gases from the engine [6]. The results of experimental studies show that the small-scale units ORC has a promising performance for power generation especially in remote areas. The ORC could provide a wide output power range, but consist

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of less components, so that the ORC could be more compact and smaller in size compared to conventional power plants.

Most studies were focused on the thermodynamic analysis of the ORC, and the selection of the working fluid, with particular attention to obtain the best efficiency of power generation. On the other hand, only few published papers discuss on the design and geometry optimization of turbo expander. Basically, for output power ranges from 5 to 5000 kW) two types of turbines are proposed, they are axial or radial turbines. The latter turbine type is considered more attractive, because it has better performance at small output capacity. Therefore, it is important to develop knowledge on geometry design and its relation to the performance of radial turbo-expanders in ORC systems. The objectives of this study is to calculate the geometry of a radial inflow Turbo Expander for small ORC system and predict its performance for various working fluids and operating conditions.

2. Rotor blade geometry design

Baines' method [7] is normally used for preliminary geometry design of the radial turbine. Load coefficient can be calculated based on the U_4 and can be calculated by using Euler's turbomachinery equation:

$$\psi = \frac{\Delta h_0}{U_4^2} = \frac{C_{\theta 4}}{U_4} - \varepsilon \frac{C_{\theta 6}}{U_4} \quad (1)$$

Where $\varepsilon = r_6/r_4$ is ratio radius of rotor.

Figure 1 shows important parameters those are used for geometry design of the radial turbo-expander rotor. Meanwhile Figure 2 shows the meridional-plane geometry on the radial turbo-expanders. In this model the rotor shroud contour is assumed to be a circle, while the rotor hub to the contours of which are described as elliptical geometry assumptions made by Glassman [8]. [8]

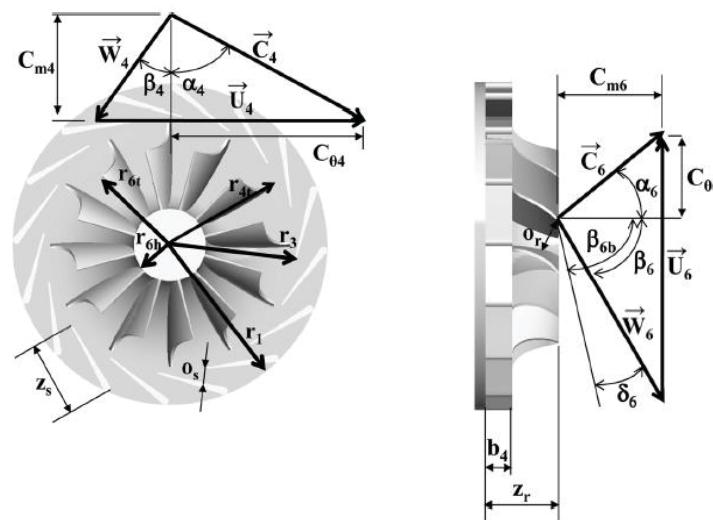


Figure 1 Geometry parameters of radial turbo-eksponder [9].

Magnitude of exit swirl very small, load coefficient can be predicted as:

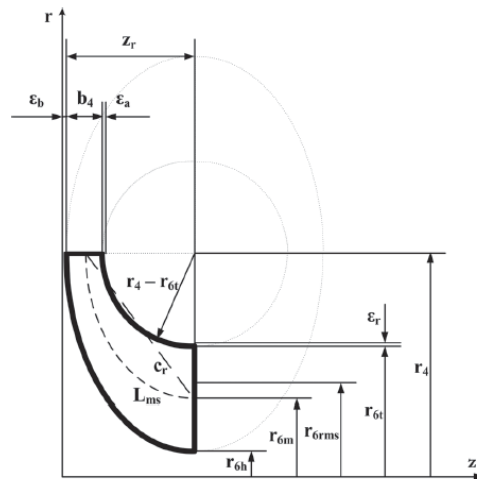
$$\psi = C_{\theta 4}/U_4 \quad (2)$$

Hence the rotor inlet velocity triangle (station 4) is defined as:

$$C_{m4} = \xi \phi U_4 \quad (3)$$

$$C_4 = (C_{m4}^2 + C_{\theta 4}^2)^{1/2} \quad (4)$$

$$\alpha_d = \tan^{-1}(C_{\theta_d}/C_{m_d}) \quad (5)$$



$$\beta_4 = \tan^{-1}[(C_{\theta 4} - U_4)/C_{m4}] \quad (6)$$

Figure 2 The meridional-plane geometry on the radial turbo-expanders [8].

The static temperature and pressure at the inlet to the rotor are :

$$T_4 = T_{04} - C_4^2/2C_y \quad (7)$$

$$p_4 = p_{04} (T_4/T_{04})^{k/(k-1)} \quad (8)$$

Where $T_{04}=T_{01}$ dan $p_{04}=p_{01}-\Delta p_0$. Here Δp_0 is the total pressure loss in the stator.

$$A_4 = mRT_4/p_4C_{m4} \quad (9)$$

At exit, the total and static temperatures are :

$$T_{06} = T_{04} - \Delta h_0 / C_p \quad (10)$$

$$T_6 = T_{06} - C_6^2/2C_p \quad (11)$$

3. Flow simulation on the rotor of radial turbo-expander

Table 1 Operational condition of the radial turbo-expander

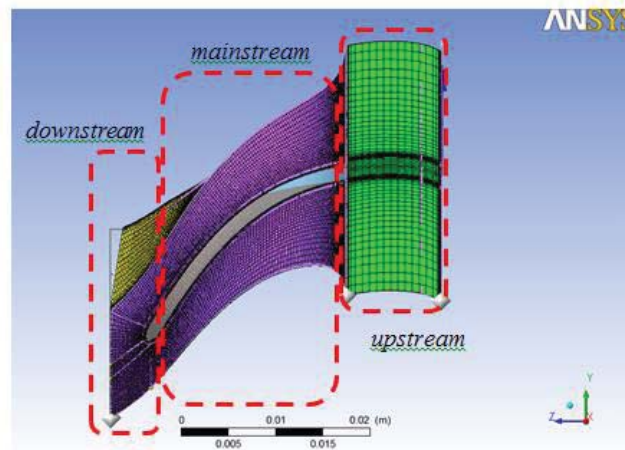
Parameter	Value
Inlet temperature (K)	353-423
Mass flow (kg/s)	0.1-1.0
Outlet pressure (atm)	1.0
Outlet temperature (K)	313-373K

3.1 Number of grid

CFD simulation were done with various number of grids, to know the influence of number of grid on the simulation speed and accuracy. Figure 2 show grid model and Table 2 show the grid number used in the simulation.

Table 2 Grid number for analysis rotor turbo-expander

Model	Grid number
Model 1 (Coarse Grid)	20,000
Model 2 (Medium Grid)	100,000
Model 3 (Fine Grid)	250,000

**Figure 3** Grid model for CFD analysis for downstream, mainstream, and upstream area

3.2 Turbulence of real gas model

ANSYS CFX turbomachinery software provide for 2 types of turbulence models for real gas, the k-epsilon, and SST (Shear Stress Transport). The influence of these model on the simulation results were investigated. In this paper, simulations were performed at steady state condition at various rotor speeds 15,000 rpm, 20,000 rpm and 30,000 rpm.

4. Result and discussion

4.1 Blade geometry design

Results of geometry design are shown in Table 3. Figure 4 shows model for radial rotor turbo-expander with geometry obtained from the design process.

4.2 CFD analysis on the rotor of radial turbo-expander

Numerical simulations using the finite volume method were done to find the fluid flow characteristics, that able to predict the resulted torque, power and efficiency of the designed turbo-expander. The simulation were done at various of number of grids, rotational speeds and working fluids. As an example. Figure 5 shows three model with different number of grids using R123 as working fluid at 20,000 rpm. Model 3 with high number of grid (250,000 grid), gives better results. With high number of grid, a strong vortex can be seen. This vortex form a higher pressure area in the region after the leading edge, separating flow from the hub surface and moving it up the blade toward the tip. Other calculation results obtained from the simulation for R-123 and R134 working fluids can be in Table 4.

Table 3 Result for geometry parameter rotor radial turbo-expander

Parameter	Unit	R134a	R123	R245fa	R143a	nPentane
Absolute meridional velocity (C_{m4})(inlet)	m/s	50	44	43	48	49
Blade speed (U_4)	m/s	167	146	143	160	163
Absolute tangential velocity ($C_{\theta 4}$)	m/s	150	132	129	144	147
Absolute flow angle (inlet) (α_4)	Degree	71.57 ⁰	71.57 ⁰	71.57 ⁰	71.57 ⁰	71.57 ⁰
Relative flow angle (inlet) (β_4)	Degree	-18.43 ⁰	-18.43 ⁰	-18.43 ⁰	-18.43 ⁰	-18.43 ⁰
Absolute velocity (C_4)	m/s	158	139	136	151	155
Relative absolute inlet (W_4)	m/s	52.7	46.38	45.33	50.60	51.60
Inlet Temperature (T_4)	K	360.47	361.15	364.37	362.60	367.37

Table 3 (continue)

Parameter	Unit	R134a	R123	R245fa	R143a	nPentane
Inlet Pressure (P_4)	bar	3.60	3.75	3.93	3.85	4.15
Inlet Area (A_4)	m ²	6.5x10 ⁻⁴	4.7x10 ⁻⁴	5.3x10 ⁻⁴	7.7x10 ⁻⁴	8.3x10 ⁻⁴
Radius rotor (r_4)	m	0.100	0.093	0.091	0.101	0.104
Inlet Blade height (b_4)	m	0.010	0.008	0.009	0.012	0.012
Inlet Density (ρ_4)	Kg/m ³	12.238	19.121	17.384	10.737	9.813
Inlet Mach number (M_4)	-	0.87	0.93	0.85	0.75	0.72

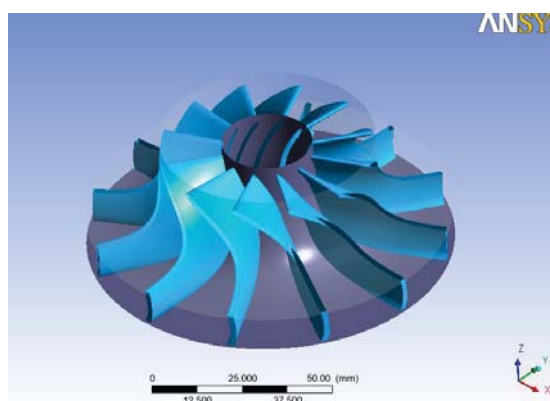
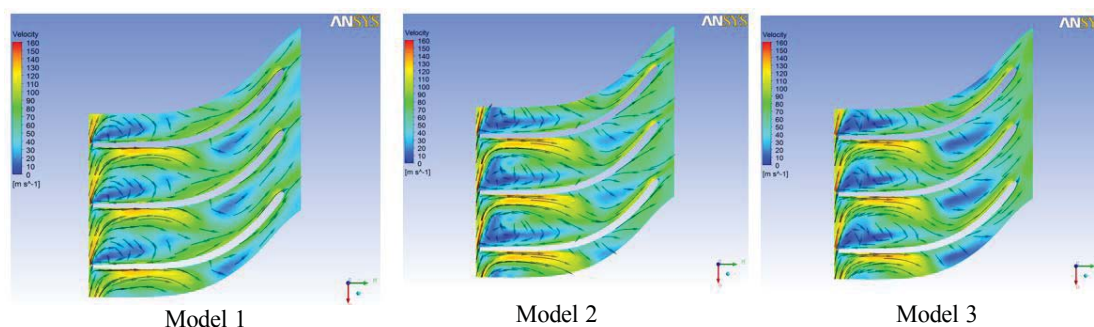
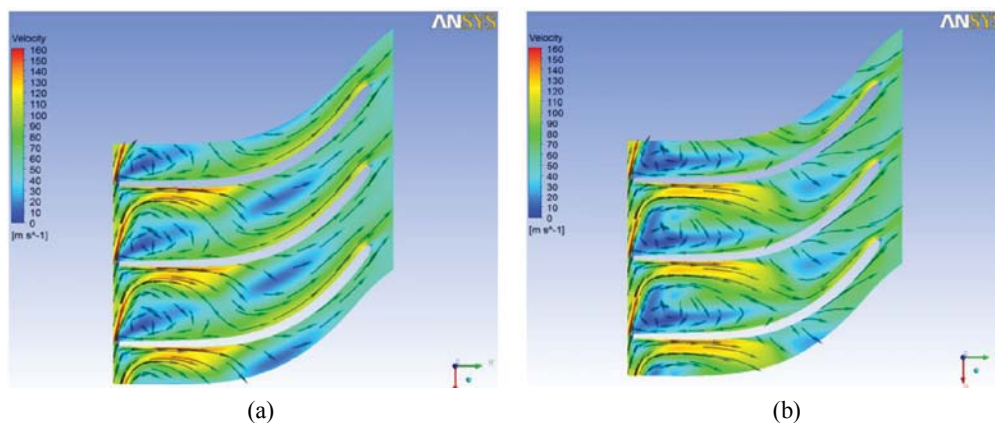
**Figure 4** Result geometry radial rotor turbo-ekspander for small organic rankine cycle system**Figure 5** Three model grid flow field predictions for radial turbine using R123 as working fluid with rotational speed 20,000 rpm

Table 4 Power and efficiency for rotor radial turbo-expander with different grid number

Number Grid	Model 1 20,000	Model 2 100,000	Model 3 250,000
Fluid: R123			
20,000 rpm			
Torque(Nm)	0.32	0.35	0.36
Power (kW)	6.56	7.08	7.21
Efficiency	0.63	0.64	0.65
Time consuming	6 minutes 30 seconds	11 minutes 56 seconds	28 minutes 5 seconds
Fluid R134a			
20,000 rpm			
Torque(Nm)	0.44	0.47	0.48
Power (kW)	7.94	8.60	8.53
Efficiency	0.63	0.64	0.65
Time consuming	6 minutes 21 seconds	11 minutes 54 seconds	27 minutes 30 seconds

**Figure 6** Velocity contour for turbulence models (a) k-epsilon dan (b) SST**Table 5** Time computation, iteration number on turbulence models for radial turbine small ORC with R123, 20,000 rpm

Model	Time consuming	Iteration	Power (kW)
k-epsilon (k-ε)	28 minutes	200	6.1
SST	28 minutes	200	7.2

Table 5 shows comparison of the computation time and the number of iterations for the k-epsilon and SST turbulence models. From Figure 5 and Figure 6, it can be seen that SST model has good swirl flow prediction on blade passage and results higher power output. For k-epsilon turbulence model, the power output is 6.1 kW while for the SST models the power output is 7.2 kW, which is 18% higher. A better model between those two cannot be verified in this study, experimental results on the expander model may clarify this discrepancy.

The result of an aerodynamic evaluation of rotor radial turbo-expander using SST turbulence model are shown in Figure 7 and Figure 8. This rotor is designed for a low specific speed of under 1.0. The lower specific speed means that the exducer area is considerably smaller, so that the exit tip to inlet radius ratio is much smaller and the blade height at the exit is smaller fraction of the inlet radius [7].

The results are different with conventional characteristic of rotor radial turbo-expander of this type. At rotor with low specific speed using air as working fluid, the fluid flow is more uniform. Meanwhile, at rotor radial turbo-expander using R123 and R134a as working fluid, a strong vortex can be seen forming on the pressure surface soon after the leading edge, separating flow from the hub surface and moving it up the blade toward the tip. It is interesting to note that the pressure gradient introduced in inlet region of the blade passage affect even the flow upstream of the leading edge quite strongly. The large variations in flow field that are created in the passage cause very significant changes in the exit flow (Figure 9). Contours of both relative and absolute flow angle just downstream of the trailing edge.

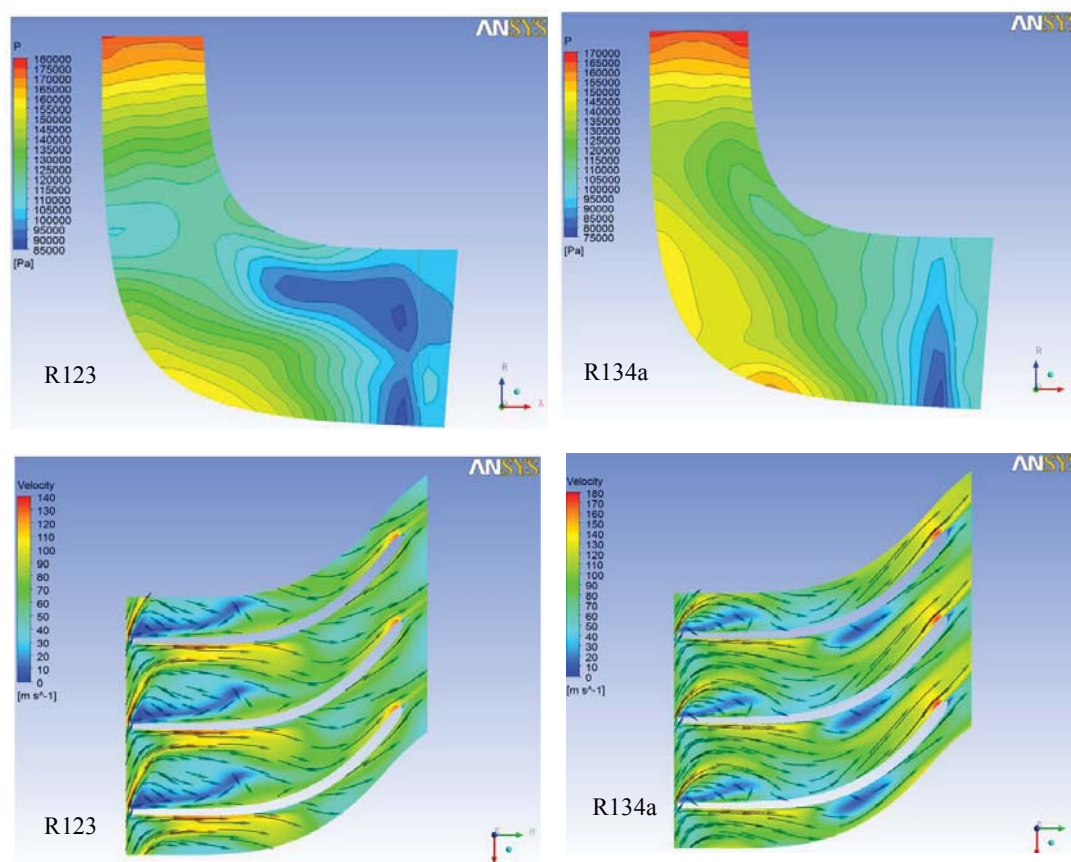


Figure 7 Flow prediction at meridional plane and blade-to-blade view for rotor radial turbo-expander using R123 and R134a at 15000 rpm

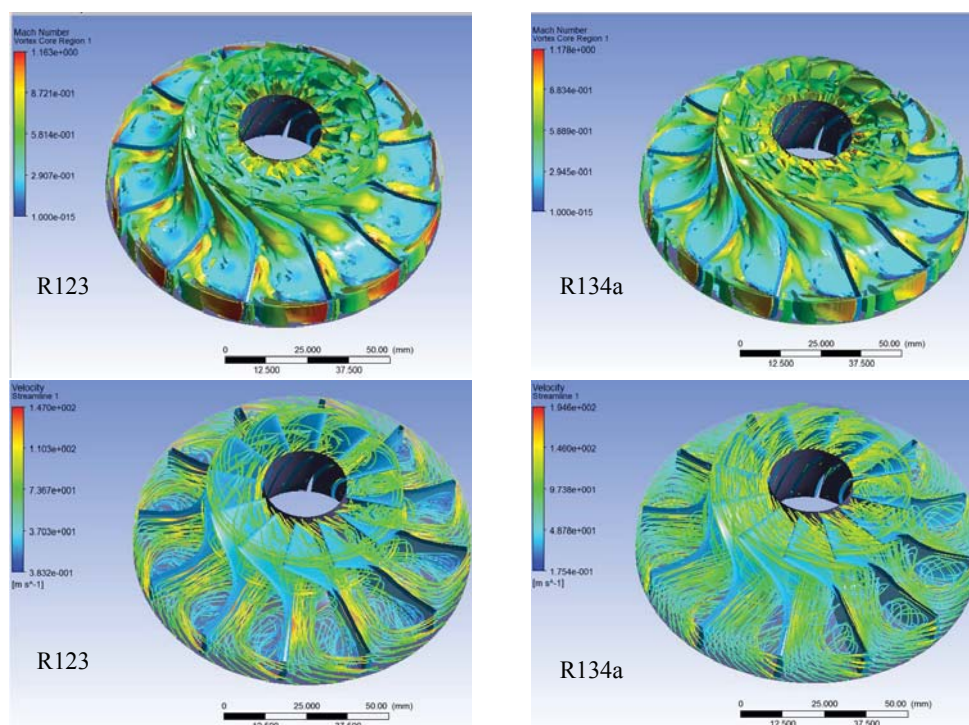


Figure 8 Flow prediction for rotor radial turbo-expander using R123 and R134a at 15000 rpm

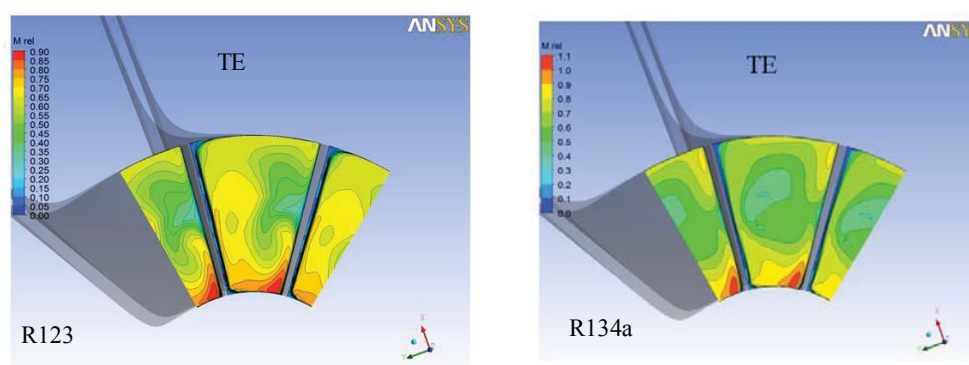


Figure 9 Flow prediction at trailing edge (TE) for rotor radial turbo-expander using R123 and R134a at 15000 rpm

5. Conclusion

This paper presents the geometry calculation of a small radial turbo-expander for small organic rankine cycle system. The performances of the expander have been evaluated using 3D numerical analysis method. From the simulation it can be concluded that the higher the number of grid will give better and accurate results. The study also suggested to use SST models for the numerical analysis because it gives higher power output. However this results has to be verified in the experimental study. From the two working fluids used in the analysis, R134a gives better performance. At mass flow rate 0.4 kg/s, 15,000 rpm, inlet pressure 5 bar, and inlet temperature 373K, the expander with R134a produces 6.7 kW power output, with total efficiency-to-static (η_{ts}) 0.71. On the other hand R123 only produces 5.5 kW, with total efficiency-to-static (η_{ts}) 0.66.

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