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 designed and the geometry and the performance 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 \( \eta_4 \) and can be calculated by using Euler’s turbomachinery equation:

\[
\psi = \frac{\Delta h_4}{\eta_4^2} = \frac{C_{\theta_4}}{\eta_4} - \frac{\theta_4}{\eta_4}
\]

where \( \psi = \frac{r_6}{r_4} \) 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].

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

\[
\psi = \frac{C_{\theta_4}}{U_4}
\]

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

\[
C_{in4} = \frac{K \beta_4 U_4}{\eta_4}
\]

\[
C_4 = \sqrt{\left( C_{in4}^2 + C_{\theta_4}^2 \right)}
\]
The static temperature and pressure at the inlet to the rotor are:

\[ T_4 = T_{04} - \frac{C_p^2}{2C_p} \]
\[ p_4 = p_{04}(T_4/T_{04})^{\gamma/(\gamma-1)} \]

Where \( T_{04} \) dan \( p_{04} \) and \( \Delta p_0 \) is the total pressure loss in the stator.

At exit, the total and static temperatures are:

\[ T_{08} = T_{04} - \frac{\Delta p_0}{C_p} \]
\[ T_8 = T_{04} - \frac{C_p^2}{2C_p} \]

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 charateristics, 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 (Cm)     | (inlet)| m/s   | 50   | 44     | 43    | 48       |
| Blade speed (U4)                       | m/s    | 167   | 146  | 143    | 160   | 163      |
| Absolute tangential velocity (Cθ)     | m/s    | 150   | 132  | 129    | 144   | 147      |
| Absolute flow angle (inlet) (α4)      | Degree | 71.57°| 71.57°| 71.57° | 71.57°| 71.57°   |
| Relative flow angle (inlet) (β4)      | Degree | -18.43°| -18.43°| -18.43°| -18.43°| -18.43°  |
| Absolute velocity (C4)                | m/s    | 158   | 139  | 136    | 151   | 155      |
| Relative absolute inlet (W4)          | m/s    | 52.7  | 46.38| 45.33  | 50.60 | 51.60    |
| Inlet Temperature (T4)                | K      | 360.47| 361.15| 364.37 | 362.60| 367.37   |

Table 3 (continue)

| Parameter            | Unit  | R134a | R123 | R245fa | R143a | nPentane |
|----------------------|-------|-------|------|--------|-------|----------|
| Inlet Pressure (P4)  | bar   | 3.60  | 3.75 | 3.93   | 3.85  | 4.15     |
| Inlet Area (A4)      | m²    | 6.5x10⁻⁴| 4.7x10⁻⁴ | 5.3x10⁻⁴ | 7.7x10⁻⁴ | 8.3x10⁻⁴ |
| Radius rotor (r4)    | m     | 0.100 | 0.093| 0.091  | 0.101 | 0.104    |
| Inlet Blade height (b4)| m   | 0.010 | 0.008| 0.009  | 0.012 | 0.012    |
| Inlet Density (ρ4)   | Kg/m³ | 12.238| 19.121| 17.384 | 10.737| 9.813    |
| Inlet Mach number (M4)| -    | 0.87  | 0.93 | 0.85   | 0.75  | 0.72     |

Figure 4 Result geometry radial rotor turbo-expander 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  | 0.32           | 0.35            | 0.36            |
| Torque (Nm) | 6.56           | 7.08            | 7.21            |
| Power (kW)  | 0.63           | 0.64            | 0.65            |
| Efficiency  | 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 verify 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 present 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 gives 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 verify in the experimental study. From the two working fluids use in the analysisi, 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 ($\eta_{ts}$) 0.71. On the other hands R123 only produces 5.5 kW, with total efficiency-to-static ($\eta_{ts}$) 0.66.

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