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Radial Turbine Preliminary Design and Modelling

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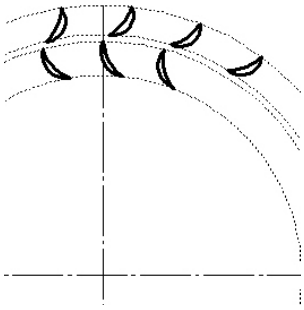
Abstract— **The turbine is the single most critical component in a thermal conversion cycle. A review of available options of organic Rankine cycle expanders suitable for use in low temperature and small to medium level power cycles has been undertaken and findings do not show sufficient evidence of significant research and development work in this regard; a turbine design suitable for small scale and low temperature operation based on the organic Rankine cycle (ORC) thermodynamic cycle is required because the operating conditions such as speeds, flow rates, pressure ratios, etc. are quite different from those of large scale, higher temperature, conventional steam and gas turbines; also the properties of the organic fluids used as working fluids in low temperature cycles are different from those of the conventional steam or fossil-fuel-gas mixtures. This paper presents the preliminary design and modelling of a radial turbine suitable for use in a small to medium level low temperature solar thermal conversion cycle. For micro operations, the radial turbine option seems more attractive as it allows a better performance in the lower size range which is also of special interest for distributed combined heat and power (CHP) units. The ideal solution should be characterised by maximum efficiency, small footprint, and minimum shaft speed. The turbine design process can be broken down into three stages: Preliminary Design (PD); Meanline/Streamline (1D/2D) Analysis and Optimization; and Profiling, 3D Blade Design, 3D Modelling and Analysis. Empirical loss correlations are used to account for the different kinds of losses. The engineering equation solver (EES) is used to perform the thermodynamic analysis. 2D and 3D computational fluid dynamics (CFD) simulation and the aerofoil design are not done at this stage as they require specialised CFD software such as SoftInWay Inc.'s AxSTREAM, AutoDesk's CFD Simulation or SolidWorks Flow Simulation.**

Keywords— ORC thermodynamic cycle, preliminary design and modelling, radial turbine, mean-line analysis, EE



I. INTRODUCTION

A turbine is a turbomachine that transfers energy from a working fluid to a rotating shaft, thereby producing work.

In a radial turbine the flow of the working fluid is substantially in the radial direction towards the turbine shaft axis. This configuration allows a radial turbine to be simpler, more robust, and more efficient especially for lower power ranges [1] when compared to an axial machine. The flow is generally inward, although there are newer designs with an outward flow such as the Euler turbine [2]. There are two types of inward flow radial turbines: cantilever turbines and ninety degree in-flow radial (90ºIFR) turbines.

The cantilever radial turbine is similar aerodynamically to the axial impulse type turbine and can be designed in a similar manner to axial turbines. Figure 1 shows a cantilever turbine rotor and Figure 2 shows the turbine blade arrangement and the corresponding rotor entry and exit velocity triangles.

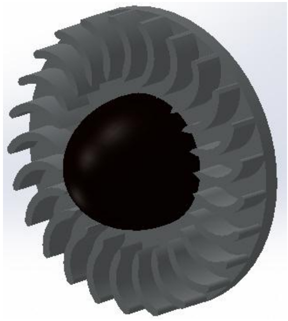
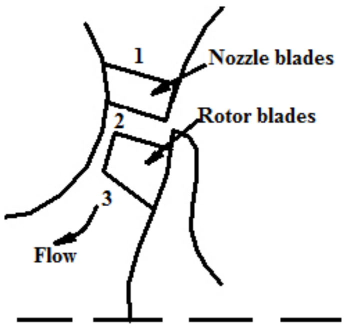


Figure 1: Cantilever radial turbine rotor (N=25)



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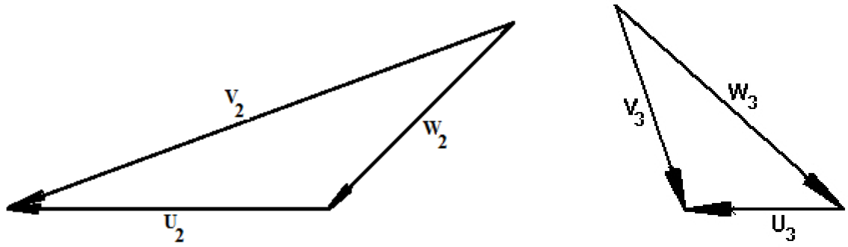

Figure 2: Cantilever turbine arrangement and velocity triangles
(where: V is absolute fluid velocity, U is blade velocity, and W is relative velocity of fluid flow to moving blades)

The 90ºIFR turbine has a striking similarity with a centrifugal compressor with the flow direction and blade motion reversed. The flow enters the turbine radially and exits the turbine axially. Straight radial blades are generally preferred as curved blades would incur additional stresses. The rotor or impeller ends with an exducer. Usually the flow exiting the rotor passes through a diffuser to recover kinetic energy which would otherwise be wasted. The 90ºIFR turbine rotor is shown in Figure 3 and the turbine blade arrangement and the corresponding rotor entry and exit velocity triangles are shown in Figure 4.

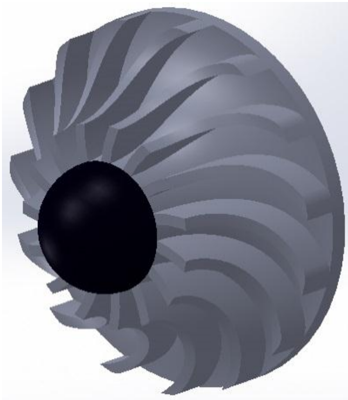

Figure 3: 90º IFR turbine rotor (N=15)

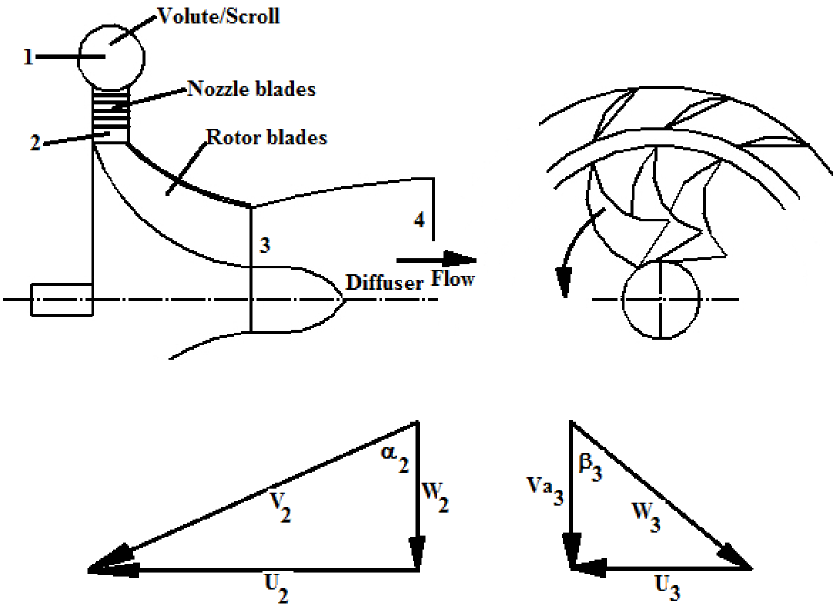

Figure 4: 90º IFR turbine arrangement and velocity triangles

II. MATHEMATICAL MODEL

The following figure 5 is an enthalpy-entropy diagram (Mollier diagram) [3] of the expansion process for a radial turbine with a diffuser where 1, 2, 3 and 4 represent the entry to the stator, entry to the rotor, exit from the rotor and exit from the diffuser respectively.

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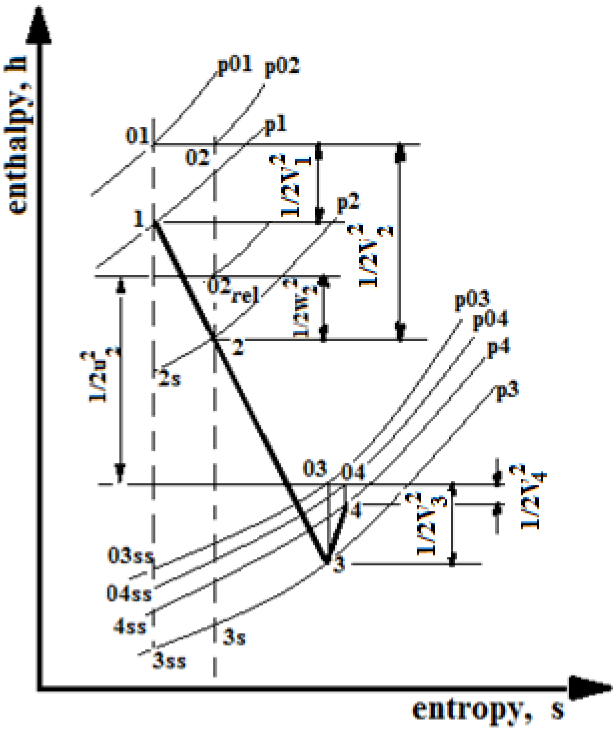

Figure 5: 90° IFR turbine enthalpy-entropy representation of the expansion process

Parameters designated as p01, p02, p03, and p04 are stagnation pressures. For states 1 and 2 which represent the inlet and outlet to the nozzle (stator), the following can be said:

$$h_{01} = h_{02}; \quad h_{01} = h_1 + \frac{V_1^2}{2}; \quad h_{02} = h_2 + \frac{V_2^2}{2} \qquad (11)$$

Similarly, the same can be said about states 3 and 4 which represent the inlet and outlet to the diffuser:

$$h_{03} = h_{04}; \quad h_{03} = h_3 + \frac{V_3^2}{2}; \quad h_{04} = h_4 + \frac{V_4^2}{2} \qquad (12)$$

Since states 2 and 3 represent the inlet and outlet to the rotor, encompassing a process that involves shaft work, the equation above is not adequate, instead that of rotational enthalpy (Rothalpy is conserved across the rotor) is used [4]:

$$I_2 = I_3; \quad I_2 = h_2 + \frac{W_2^2}{2} - \frac{U_2^2}{2}; \quad I_3 = h_3 + \frac{W_3^2}{2} - \frac{U_3^2}{2} \qquad (13)$$

Rate of Work done or Power output is given by:

$$\dot{W} = \dot{m} \cdot (U_2 \cdot V_{u2} - U_3 \cdot V_{u3}) \qquad (14)$$

Total-to-Static expander efficiency is given by:

$$\eta_{ts} = \frac{h_{01} - h_{03}}{h_{01} - h_{3ss}} \qquad (15)$$

Total-to-Total expander efficiency is given by:

$$\eta_{tt} = \frac{h_{01} - h_{03}}{h_{01} - h_{03ss}} \qquad (16)$$

Blade loading coefficient:

$$\Psi = \frac{\Delta h_{tt}}{U_2^2} = \frac{h_{01} - h_{03}}{U_2^2} \qquad (17)$$

Flow coefficient:

$$\phi = \frac{V_{m2}}{U_2} \qquad (18)$$

The effective degree of reaction is given by:

$$R_r = \frac{\Delta h_{RR}}{\Delta h_{Rstage}} \qquad (19)$$

where: $\Delta h_{RR} = h_2 - h_3$, $\Delta h_{Rstage} = h_1 - h_3$

and the specific speed:

$$N_s = \frac{rpm}{60} \cdot \frac{Q_3^{0.5}}{\Delta h_{0stage}^{3/4}} \qquad (20)$$

where: $\Delta h_{0stage} = h_{01} - h_{03ss}$

III. COMPUTER SIMULATIONS

The following parameters as represented in Figure 6, together with the inlet conditions shown in Table 1, are used in the computer simulations. The simulations are performed on the EES (engineering equation solver) platform [5]:

L_d = axial flow length of turbine;

d1 = nozzle inlet diameter;

d2 = rotor inlet diameter;

d3 = rotor mean outlet diameter;

d4 = diffuser outlet diameter;

b1 = nozzle inlet blade height;

b2 = rotor inlet blade height;

b3 = rotor exit blade height;

b4 = diffuser gap height;

α1 = nozzle inlet (absolute) flow angle;

α2 = rotor inlet absolute flow angle;

α3 = rotor exit absolute flow angle;

β2 = rotor inlet relative flow angle;

β3 = rotor exit relative flow angle; and

α4 = diffuser exit (absolute) flow angle.

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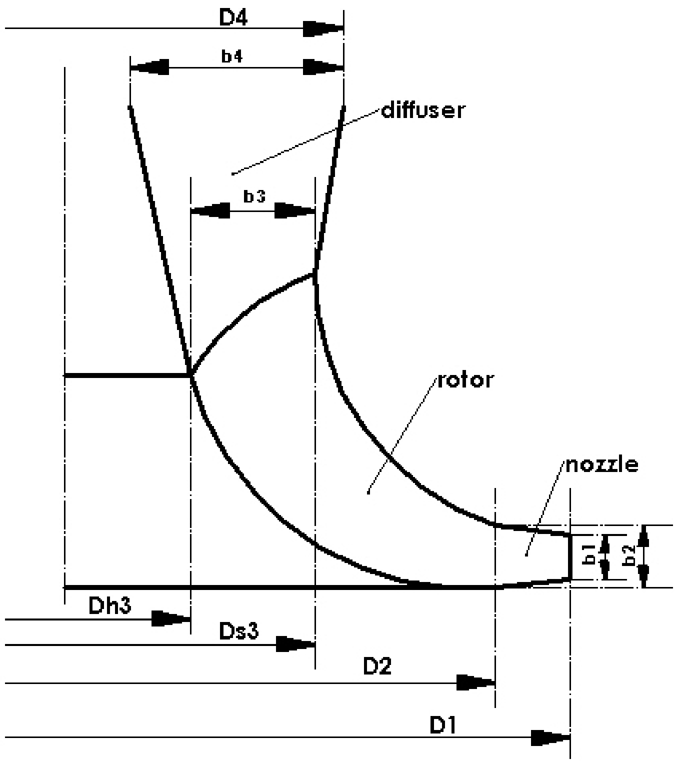

Figure 6: Radial turbine blade geometry

Table 1: Inlet conditions [6]

Working Fluid	Mass flow rate [kg/s]	Inlet Pressure [Pa]	Inlet Temperature [ºC]
R245fa	0.396	810600	80.99
R134a	0.396	810600	80.99
n-butane	0.207	1010000	80.03
isobutene	0.241	1010000	66.82

In the simulations that follow, the outlet pressure was varied from 130 kPa to 180 kPa and the rotor inlet diameter varied from 20 mm to 150 mm. R134a was included at a later stage, although it had previously not been considered favourable during the development stage of the evaporator model; the reason for this change of heart was mainly because it had now been established that since this was the working fluid that we were able to procure, this was the liquid we were going to use for the experimental model.

Three sets of simulations were conducted:

Simulation 1: rotor exit static pressure (P_3) was varied from 130 kPa to 180 kPa; the results are shown in plots of Figures 7 and 8.

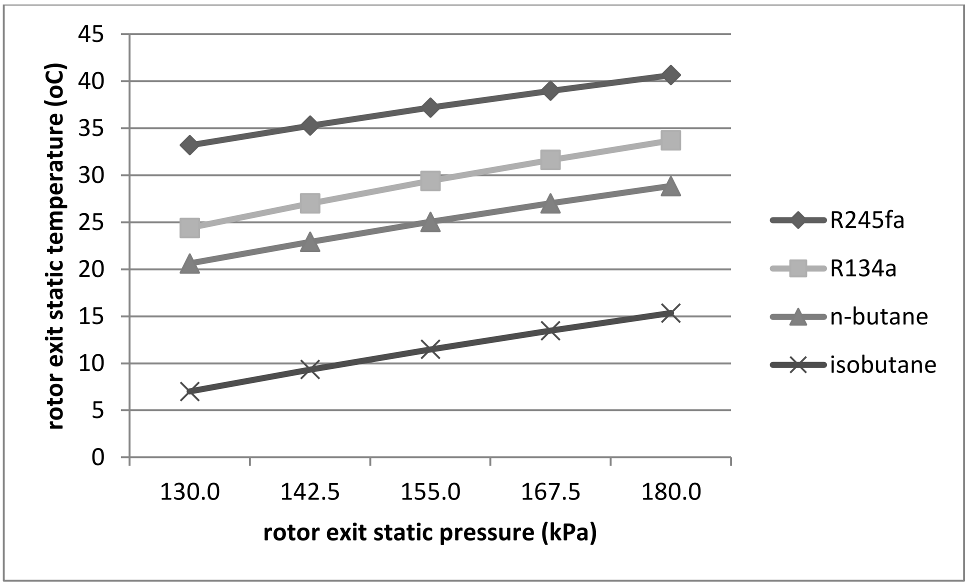

Figure 7: radial turbine model rotor exit – temperature versus pressure

The turbine (rotor) exit temperature is important in that it needs to be higher than the sink (usually ambient) temperature for the thermal cycle to meet requirements for thermodynamic functionality; from these simulations it is evident that given the current operating conditions only R245fa (and to a lesser extent R134a) satisfy this cycle temperature constraint; for n-butane and isobutene the cycle requires to operate at higher pressures for them to satisfy the temperature limits.

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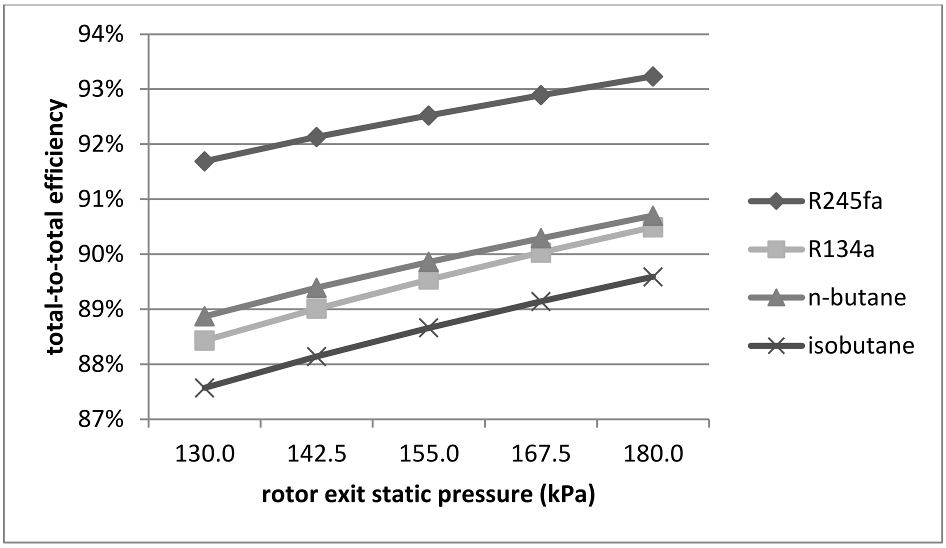

Figure 1: Radial turbine model total-to-total efficiency versus rotor exit pressure

In terms of the efficiency it is seen that all the four working fluids perform satisfactorily; R245fa has the highest efficiency followed by n-butane, R134a is third and isobutene has the lowest efficiency.

Simulation 2: rotor inlet diameter (D_2) was varied from 20 mm to 150 mm while keeping rotor exit pressure constant at 180 kPa by adjusting the machine rotational speed to achieve the desired power output. Results are shown in Figure 9.

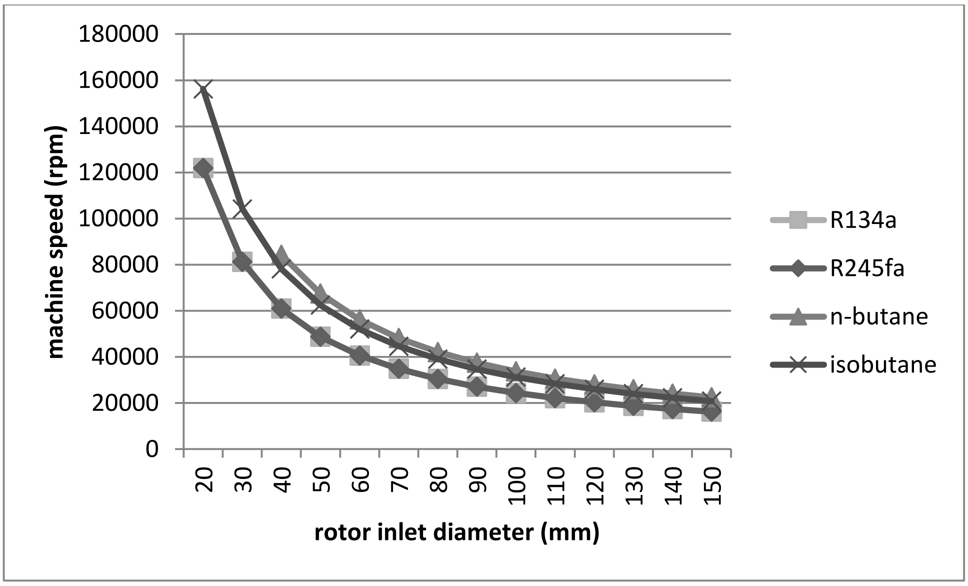

Figure 9: Radial turbine model machine speed versus rotor diameter

Simulation 3: machine speed was set constant at 20000 RPM. With the rotor exit pressure and machine speed set constant at 180 kPa and 20000 rpm, the final simulation was performed for the optimized results; the rotor outlet velocity triangle for R134a had to be reset to the 'NO–swirl' state thus giving a β_3 angle of 77.09°.

With the first simulations, the machine speed was not constrained and attained too high levels of 61000 rpm for both R245fa and R134a, 84500 rpm for n-butane and 78000 rpm for isobutene. Since it was desirable to limit the machine speed to lower values, another set of simulations was performed whereby the rotor exit pressure was kept at the higher optimum value of 180 kPa while the machine speed was varied by incrementally changing the rotor inlet diameter; all other dimensional characteristics of the turbomachine model were geometrically linked to the rotor inlet diameter. The objective of these simulations was to size the turbomachine such that it had an acceptable speed. From Figure 9 it can be seen that 20000 rpm is an acceptable optimal speed. The corresponding preliminary design parameters for the 10 kWe radial ORC turbine model are tabulated and plotted in the Table 2 and velocity triangles in Figures 10 to 12 respectively.

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Table 2: Radial turbine model simulation results

Fluid$	m_dot	P_1	P_2	P_3	PR	T_1	T_2	T_3	T_4	eta_tt
	[kg/s]	[Pa]	[Pa]	[Pa]		[C]	[C]	[C]	[C]	[-]
R245fa	0.396	810600	250080	180000	4.5	80.99	48.67	40.64	40.67	0.9324
R134a	0.396	810600	350969	180000	4.5	80.99	53.84	33.69	34.55	0.9051
n-butane	0.207	1010000	372911	180000	5.6	80.03	48.65	28.86	29.61	0.907
isobutene	0.241	1010000	412422	180000	5.6	66.82	38.13	15.33	16.29	0.896

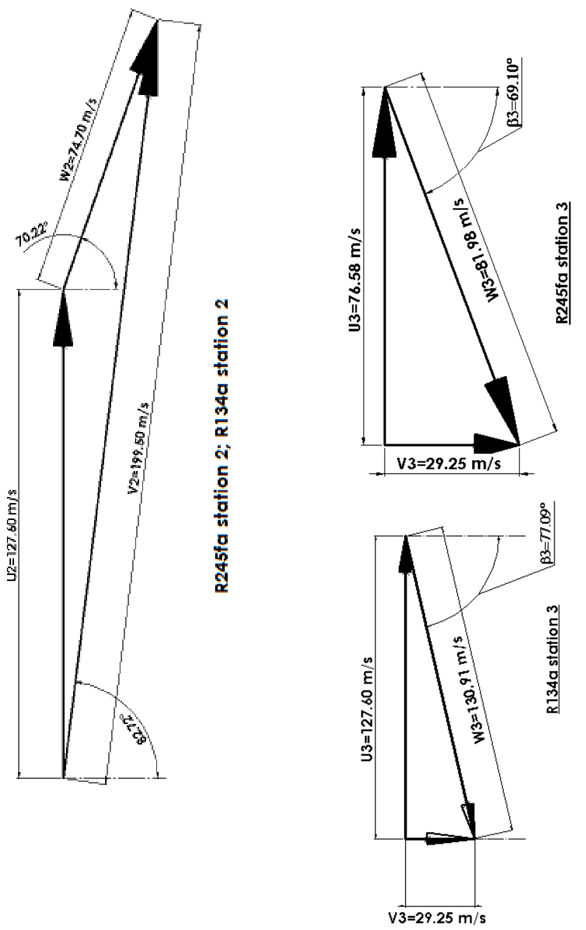

Figure 2: Radial turbine model velocity triangles for R245fa and R134a

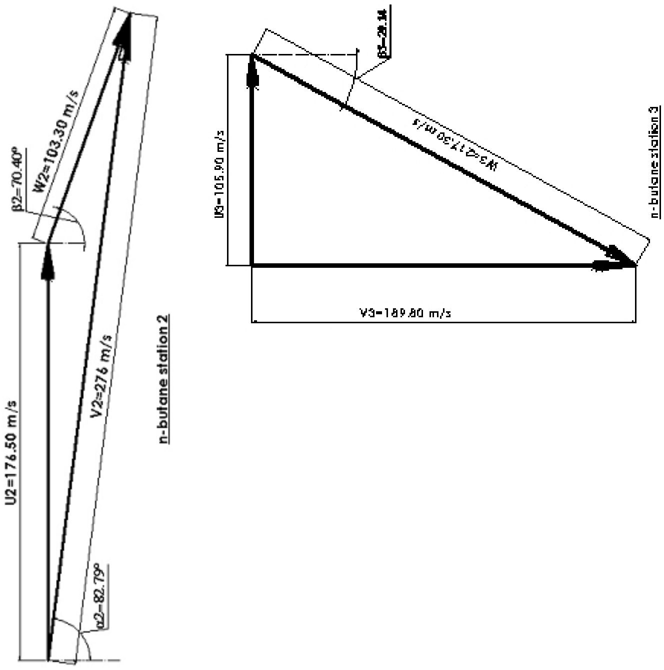

Figure 3: Radial turbine model velocity triangles for n-butane

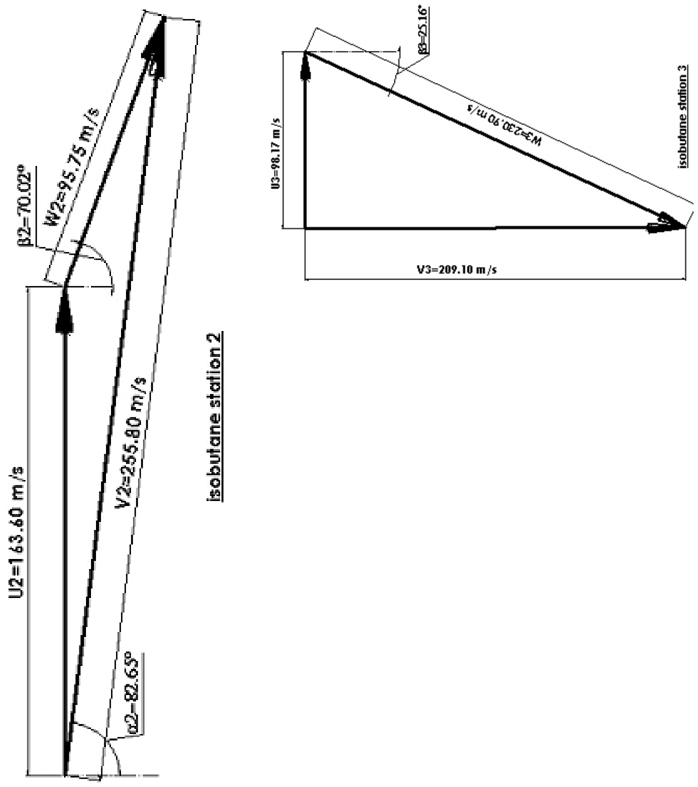

Figure 4: Radial turbine model velocity triangles for isobutene

IV. DISCUSSION AND CONCLUSIONS

This paper has presented preliminary design models for the radial turbine suitable for a 10 kWe low temperature ORC. The preliminary design has been presented in terms of geometric parameters of flow angles, blade diameters and heights. The preliminary design included thermodynamic parameters of stagnation and static pressures, temperatures and enthalpy's; the thermodynamic analyses were conducted within the cycle temperature ranges of the evaporator and condenser. Although the presented design models are not complete, this work has shown that small radial turbines for low temperature cycles are a feasible design option. The turbine preliminary design parameters for the 10 kWe turbine model after parametric optimization are listed in the Tables 1 and 2.

V. WAY FORWARD

To fully complete this task it is necessary to employ CFD and FEA analyses and modelling of the detailed blade and nozzle geometry and flow profile design. This would be followed by providing material and manufacturing specifications for prototype construction and testing; 3D printed physical prototypes could be produced and laboratory tested. AxSTREAM software suite by SoftInWay Inc. is a good package for turbine CFD modelling.

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