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# Modelling and parametric analysis of small-scale axial and radial-outflow turbines for Organic Rankine Cycle applications

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## 8 Abstract

9 The existing literature pays limited attention to the design and 3D analysis of small-scale axial and 10 radial-outflow turbines that can be utilised in Organic Rankine Cycles (ORC) for power generation with a 11 low-temperature (<100°C) heat source and low mass flow rate. Turbine efficiency significantly affects an ORC's 12 efficiency because the turbine is considered a key component of the ORC. Therefore, obtaining high cycle thermal 13 efficiency requires high turbine efficiency and power output. This work presents an integrated mathematical model 14 for developing efficient axial and radial-outflow (centrifugal) turbines using a range of organic working fluids 15 (R141b, R245fa, R365mfc, isobutane and n-pentane). This mathematical approach integrates mean-line design and 3D CFD analysis with ORC modelling. The ANSYS<sup>R17</sup>-CFX is used to predict 3D viscous flow and turbine 16 17 performance. To achieve accurate prediction, the ORC/turbines model uses real gas formulations based on the 18 REFPROP database. The results showed that the axial turbine performed better, with efficiency of 82.5% and power 19 output of 15.15 kW, compared with 79.05% and 13.625 kW from the radial-outflow turbine, with n-pentane as the 20 working fluid in both cases. The maximum cycle thermal efficiency was 11.74% and 10.25 % for axial and radial-21 outflow turbines respectively with n-pentane as the working fluid and a heat source temperature of 87 °C. The large 22 tip diameter of the axial turbine was 73.82 mm compared with 108.72 mm for the radial-outflow turbine. The 23 predicted results are better than others in the literature and highlight the advantages of the integrated approach for 24 accurate prediction of ORC performance based on small-scale axial and radial-outflow turbines.

25 Keywords: Mean-line design; Organic Rankine Cycle; CFD; small-scale; axial and radial-outflow turbines.

#### 26 1. Introduction

27 Recently, exploiting low-temperature heat sources such as solar and geothermal energy has increasingly focused on 28 decreasing reliance on fossil fuels. Consequently, ORC systems are a promising technology that can be used to 29 convert low-temperature heat sources into useful energy. Small-scale ORC systems based on axial and 30 radial-outflow turbines are suitable for many electricity generation applications, such as domestic and rural 31 situations, isolated installations and off-grid zones.

| Nomenc          | clature  |                 |                                  |
|-----------------|--|-----------------|----------------------------------|
| А               | area $(m^2)$   | AS              | aspect ratio                     |
| В               | constant of tip clearance loss (-)                                   | blds            | blade                            |
| b               | axial chord (m)  | cr              | critical                         |
| С               | absolute velocity (m $s^{-1}$ )                                      | e               | evaporator                       |
| с               | chord length (m)   | sec eff         | second law efficiency            |
| CL              | lift coefficient (-)   | ex              | exergy                           |
| D, d            | diameter (m)   | Н               | high                             |
| f               | correction/friction coefficient's (-)                                | hyd             | hydraulic                        |
| h               | specific enthalpy (kJ/kg)  | L               | low                              |
| Н               | Blade height (m)   | nbp             | normal boiling point             |
| K               | losses coefficient (-)   | Р               | profile/pump                     |
| k               | specific turbulence kinetic energy (m <sup>2</sup> s <sup>-2</sup> ) | R               | rotor                            |
| ṁ               | mass flow rate (kg $s^{-1}$ )  | Re              | Reynolds number                  |
| Ν               | Number of blade (-)  | Rec             | recuperator                      |
| 0               | throat (m)   | S               | stator                           |
| р               | pressure (bar)   | Sec             | secondary                        |
| Ż               | heat (kW)  | sh              | shock                            |
| R <sub>n</sub>  | reaction (-)   | t               | turbine                          |
| S               | blade space (pitch) (m)  | Т               | total                            |
| S               | entropy (kJ kg <sup>-1</sup> .K <sup>-1</sup> )                      | TC              | tip clearance                    |
| Т               | temperature (K)  | TE              | trailing edge                    |
| t               | time (s)/ Blade thickness (m)  | ts              | total-to-static                  |
| U               | blade velocity $(m/s)/$ mean flow velocity $(m s^{-1})$              | tt              | total-to-total                   |
| V               | Velocity (m/s)   | *               | uncorrected                      |
| W               | relative velocity (m $s^{-1}$ )                                      | <u>Acronyms</u> |                                  |
| W               | specific work (kJ kg <sup>-1</sup> )                                 | 1D, 3D          | one and three dimensional        |
| Ŵ               | power (kW)   | CFD             | computational fluid dynamics     |
| Greek           | symbols  | GWP             | global warming potential         |
| a               | absolute flow angle (degree)   | ODP             | ozone depletion potential        |
| ß               | relative flow angle (degree)   | ORC             | organic Rankine cycle            |
| P<br>n          | efficiency (%)   | PD              | preliminary mean-line design     |
| ч<br>ф          | flow coefficient (-)   | RANS            | Revnolds-Averaged Navier-Stockes |
| Ψ               | loading coefficient (-)  | SST             | shear stress transport           |
| Ψ<br>ω          | specific turbulence dissipation rate $(m^2 \text{ sec}^{-3})$        | 551             | shour substituitsport            |
| 0<br>0          | Angular velocity (rad $s^{-1}$ )                                     |                 |                                  |
| τ               | Tip clearance (m)  |                 |                                  |
| r               | enthalpy loss coefficient (-)  |                 |                                  |
| د               | ( )  |                 |                                  |
| <u>Subscrip</u> | pt/superscript   |                 |                                  |
| 1-6             | station within the turbine and cycle respectively.                   |                 |                                  |
| accel           | accelerating   |                 |                                  |

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In an ORC system, organic fluids such as refrigerants and hydrocarbons are used as working fluids instead of steam. High-density organic fluids offer unique advantages through their small plant size compared with steam turbine/Rankine cycle systems. Using dry organic fluids produces a dry vapour phase after expansion through the turbine, preventing the presence of liquid droplets in the flow path and reducing the cost of maintenance. The low-temperature heat source leads to a low level of operating pressure, reducing the system's complexity and alleviating safety concerns. Selecting a practical turbine type depends on the maximum efficiency of the turbine type in specific operating conditions; the turbine is considered a key component of the ORC system and its performance (efficiency and power output) significantly affects the ORC's thermal efficiency. Mean-line design of a radial-inflow turbine for ORC applications was performed for low power output levels in [1,2,3,4,5,6,7,8,9,10,11] with maximum turbine isentropic efficiency of 84% as reported in [10].

44 For the axial turbine stage, the preliminary mean-line design was proposed and performed by [12,13] with 45 different maps of isentropic efficiency for different working fluids. Martins et al. [14] designed and optimised a 46 partial-admission axial turbine used in an ORC system for heat recovery below 140 °C. R245fa was used as the 47 working fluid with the Redlich-Kwong-Soave equation of state as a real gas model. Maximum efficiency of around 48 81% was achieved with a convergent nozzle. Pei et al. [15] carried out an experimental test of a kW-scale ORC 49 system operating with R123. The test was performed using a radial-inflow turbine with a maximum temperature 50 difference of 70 °C between the hot and cold sides; isentropic and cycle efficiencies were 65% and 6.5%. Wang et 51 al. [16] conducted an experimental investigation of low-temperature solar recuperative ORC with R245fa as the 52 working fluid and a flat-plate solar collector. Their results showed that with constant mass flow rate of R245fa, the 53 experimental system's cycle efficiency remained steady at 3.67%. Kang et al. [17] conducted an experimental study 54 of an ORC based on a radial-inflow turbine using a low-temperature heat source, with R245fa as the working fluid. 55 The maximum turbine efficiency, power output and cycle thermal efficiency were found to be 78.7%, 32.7 kW and 56 5.22% respectively. Ssebabi et al. [18] replaced the rotor of the radial turbine kit with one manufactured in-house 57 and designed for low-grade waste heat recovery. The test was carried out using air as the working fluid, then used to 58 scale the turbine for R123. The predicted performance was very similar for both rotors, with low isentropic 59 efficiency (6-10%). Clemente et al. [19] evaluated the performance of different expanders, including axial turbine, 60 radial turbine, scroll and positive displacement expanders to design a bottoming cycle to recover heat from the 61 exhaust gases of a 100 kWe gas turbine. The highest power achieved was 26 kWe with 8% cycle efficiency. Pu et al. 62 [20] presented an experimental study on a small-scale ORC system based on a single-stage axial turbine, which 63 reported the influence of mass flow rate and evaporation pressure on ORC performance. Maximum power output 64 from the ORC system was 1979 W and 1027 W for R245fa and HFE7100 respectively. Kang [21] designed and 65 tested an ORC, with R245fa as the working fluid, based on a two-stage radial-inflow turbine to enhance the system's 66 performance. The results showed power, turbine isentropic and cycle efficiencies of around 39 kW, 58.4% and 9.8%

67 respectively with an evaporation temperature of 116 °C. Hu et al. [22] presented off-design analysis of ORC system 68 performance for a geothermal application, with variable mass flow rates and temperature, using a radial-inflow 69 turbine. Turbine efficiency, power output and ORC system efficiency at the design point were 82.3%, 66.9 kW and 70 5.5% respectively with R245fa as the working fluid and mass flow rate of 5.85 kg/s. Chang et al. [23] undertook an 71 experimental investigation of a low-temperature organic Rankine cycle based on a scroll expander with R245fa as 72 the working fluid. For heat source temperatures below 100 °C, the results showed expander efficiency, power output 73 and cycle thermal efficiency of 73.1%, 2.3 kW and 9.44% respectively. Eyerer et al. [24] performed an experimental 74 and analytical study of ORC for low-temperature applications by replacing R245fa as the working fluid with the 75 low-global-warming-potential fluid R1233zd. The experimental investigation was conducted using a scroll-expander 76 with different mass flow rates, rotational speeds and condensing temperatures. The results showed that R1233zd 77 outperformed R245fa by 6.92% in terms of cycle efficiency. Ziviani et al. [25] conducted an experimental and 78 numerical study of a single-screw expander for an ORC application with a heat source temperature of 125 °C. Two 79 different working fluids (R245fa and SES36) were used. The results showed that R245fa generated 10% higher 80 power output than SES36.

81 Fiaschi et al. [26] proposed a 3D design and analysis of a 5 kW micro radial-inflow turbine with R134a as the 82 working fluid, in which total-to-static efficiency and power output were 69.35% and 4.504 kW. Sauret and Gu [27] 83 carried out 1D analysis and 3D simulation of a radial-inflow turbine working with R143a in different operating 84 conditions, including off-design conditions. Maximum efficiency and power output were 87.6% and 421.5 kW 85 respectively. Al Jubori et al. [28] presented a new methodology integrating an ORC based on a small-scale axial 86 turbine. Their results showed that, using working fluid R123 for a turbine of 70 mm mean diameter, the maximum 87 isentropic efficiency was 82% and power output 5.66 kW, leading to cycle efficiency of 9.5%. Russell et al. [29] 88 conducted a design and testing process for a 7 kW radial-inflow turbine using R245fa as the working fluid. The 89 maximum total-to-total efficiency was approximately 76%, with approximate power output of 7kW for a pressure 90 ratio of around 3.5. Persico et al. [30,31] conducted a 3D CFD aerodynamic analysis of small-scale centrifugal 91 turbine cascades, which found efficiency to be higher than estimated by 1D analysis. Casati [32] proposed a 1D 92 preliminary design of an example 10 kWe centrifugal turbine for a mini-ORC, for power systems driven by heat 93 recovery. The preliminary design results showed efficiency in excess of 79%. Nithesh and Chatterjee [33] designed 94 a small-scale laboratory radial-inflow turbine with 2 kW power output for an ocean thermal energy conversion

95 application. R134a was used as the working fluid, with a turbine speed of 22,000 rpm. The results showed turbine 96 efficiency of 70% with turbine rotor tip radius of 35.5 mm. Sung et al. [34] designed and built a 200 kW ORC, 97 based on a radial-inflow turbine, for a waste heat recovery application with a heat source temperature of 140 °C and 98 R245fa as the working fluid. The experimental results showed power output and ORC thermal efficiency of 99 177.4 kW and 9.6% respectively.

100 High turbine efficiency is necessary in order to achieve high system performance in small-sized power output 101 applications from a low-temperature heat source and low mass flow rate. Therefore, the aim of this study is to 102 investigate the potential of a small-scale radial-outflow (centrifugal) turbine, compared with a small-scale axial 103 turbine, for a small-scale ORC for low-scale power generation (i.e. 5 kW - 15 kW) applications, driven by a low-104 temperature heat source. In particular, a new methodology based on 1D mean-line design and 3D CFD analysis is 105 conducted and integrated with ORC modelling to provide accurate performance assessment for both turbine 106 configurations. To the authors' knowledge, this significant aspect has not previously been considered in the 107 literature for this application, especially with radial-outflow turbines. This integrated approach allows us to 108 exchange the assumption of constant isentropic turbine efficiency for dynamic isentropic turbine efficiency. 109 Dynamic isentropic efficiency is unique to each turbine configuration, working fluid and operating conditions. The 110 ORC modelling and mean-line design of the turbines (both axial and radial-outflow) is implemented using the Engineering Equation Solver (EES) software. and ANSYS<sup>R17</sup>-CFX was utilised to predict 3D viscous flow and 111 112 turbine performance. To achieve accurate prediction and real behaviour of the five organic working fluids, real gas 113 formulation is used in the ORC/turbines model. In order to highlight the advantages, the results present and compare 114 design and off-design conditions for each turbine configuration using available low-grade heat sources such as solar 115 and geothermal energy. Furthermore, there exists gap in the knowledge regarding the development of efficient 116 small-scale axial and radial-outflow turbines for low power output capacity below 15 kW. This research offers better 117 understanding of small-scale axial and radial outflow turbines performance by providing more results on organic 118 working fluids, turbines size, power output, turbines and cycle efficiencies.

119 2. Organic working fluid selection

Selecting the organic fluid is essential to the design and performance analysis of an ORC system. The selection of organic working fluid and the accomplished efficiency varies considerably based on the selected temperature levels, the turbine type, ORC cycle configuration, environmental impact, and application and power-sized. 123 Substances selected as working fluids for low-temperature ORCs are outlined in Table 1. The working fluids have 124 an enormous effect on turbine design for ORCs. The thermo-physical properties of organic fluids substantially affect 125 turbine size and performance, system efficiency, system stability and safety (critical pressure and temperature), cost 126 and availability, and environmental issues such as global warming potential (GWP), ozone depletion potential 127 (ODP), safety and life time, as presented in Table 1. Also, some thermal-physical properties of organic fluids should 128 be taken into account in selecting appropriate working fluids for low-temperature heat source applications, such as 129 their latent heat and specific volume. The lower latent heat of vaporisation of organic fluids is preferred due to its 130 generating higher flow rates of working fluid vapour for the same amount of heat. The choice of working fluids is a 131 major challenge for ORC turbines designers and it is based on an acceptable balance between the abovementioned 132 criteria, environmental concerns, thermodynamic performance, commercial availability and cost (e.g. R245fa is 133 selected based on these criteria and is recommended in literature as a suitable working fluid for low temperature heat 134 sources application).

With a low-temperature heat source, the choice of isentropic and dry working fluids (dT/ds slope >1) are more favourable for ORC because expansion in the turbine will be in the superheated regime, as shown in the T-s (Temperature-entropy) diagram in Fig. 1. This will alleviate concerns over the existence of droplets of organic liquid in the rotor stage, compared with wet fluids' expansion in the wet regime, which requires preheat equipment, as shown in Fig. 1a. This feature considerably reduces turbine maintenance and evaporator size requirements, leading to reduced capital cost of the ORC system.

|           | <b>Table 1.</b> Physical, safety, and environmental properties for five organic fluids. |                  |             |                       |      |          |                  |        |  |
|-----------|---|------------------|-------------|-----------------------|------|----------|------------------|--------|--|
| Fluid     | Mol.  | T <sub>nbp</sub> | $T_{cr}(K)$ | P <sub>cr</sub> (kPa) | ODP  | GWP      | Atmospheric life | Safety |  |
|           | weight  | (K)              |             |                       |      | (100 yr) | time (yr)        | group  |  |
|           | (g/mol)   |                  |             |                       |      |          |                  |        |  |
| R141b     | 116.95  | 305.05           | 480         | 4460                  | 0.12 | 725      | 9.2              | A2     |  |
| R245fa    | 134.05  | 288.14           | 426         | 3610                  | 0    | 950      | 7.7              | B1     |  |
| R365mfc   | 148.07  | 313.18           | 459.9       | 3266                  | 0    | 850      | 8.7              | n.a.   |  |
| Isobutane | 58.122  | 272.51           | 408         | 37.96                 | 0    | <10      | 0.018            | A3     |  |
| n-Pentane | 72.15   | 309.1            | 469         | 3360                  | 0    | ~20      | 0.01             | A3     |  |

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Fig.1. T-s diagram of dry fluid (a), T-s diagram of five organic working fluids (b). and

### 159 **3. ORC system modelling**

160 The layout of the proposed recuperative ORC system is displayed in Fig. 2. The system consists of five main 161 components: evaporator, turbine, condenser, pump, and recuperator. The subcritical ORC cycle is considered in this 162 study to avoid the complexity and safety concerns of high-pressure systems. Heat and pressure losses through the 163 connecting pipe of the ORC system are negligible. Steady state operating conditions are assumed. Heat added from 164 the low-temperature heat source is given by:

$$\dot{Q}_e = \dot{m}(h_1 - h_6)$$
 (1)

165 Net power output from the ORC cycle is given by:

$$\dot{W}_{net} = \dot{W}_t \eta_{mech} \eta_{gen} - \dot{W}_p \tag{2}$$

166 where  $\eta_{mech}$  and  $\eta_{gen}$  are mechanical efficiency and generator efficiency.

167 ORC thermal efficiency is given by:

$$\eta_{\rm th} = \frac{W_{\rm net}}{\dot{Q}_{\rm e}} \tag{3}$$

168 The second law efficiency can be defined as the proximity of the real thermal efficiency of the cycle to the169 Carnot cycle efficiency as:

$$\eta_{\text{sec eff}} = \frac{\eta_{\text{th}}}{\eta_{\text{Carnot}}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{e}} \left(1 - \frac{T_{\text{L}}}{T_{\text{H}}}\right)}$$
(4)

The design input parameters of the ORC system modelling and turbine design are detailed in Table 2. The design parameter values are stated in terms of heat source temperature and heat sink temperature (cold side temperature) with five organic working fluids for different mass flow rates ranged within 0.3-0.7 kg/sec. The mass flow rate of organic fluid is used as the inlet condition in the ORC system analysis and turbine design to calculate the desired power output. Therefore, the turbine, and thus the ORC system can be sized to meet this specification. Also, the performance of the turbine and ORC system is expressed as a function of mass flow rate.

| Table 2. The input para      | ameters of the ORC mode | el.     |
|------------------------------|-------------------------|---------|
| Parameters                   | Unit                    | Value   |
| Heat source temperature      | К                       | 360     |
| Heat sink temperature        | К                       | 293     |
| Pump efficiency              | -                       | 0.75    |
| Generator efficiency         | -                       | 0.96    |
| Mechanical efficiency        | -                       | 0.96    |
| Recuperator effectiveness    | -                       | 0.8     |
| Working fluid mass flow rate | kg/s                    | 0.3-0.7 |



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Fig. 2. Schematic diagram of recuperative Rankine cycle components.

#### 186 4. Turbine Design

187 A mobile small-scale ORC system with power output of a few kW, based on axial and radial-outflow turbines, 188 is capable of converting the energy from a low-temperature heat source into useful power using organic working 189 fluids. The radial-outflow turbine has a low specific work per stage because of the reduction of peripheral velocity 190 through the expansion of the working fluid ( $U_2 < U_3$ ) compared with the axial turbine, as shown in Fig. 3. Thus, a 191 number of stages are required to increase the specific work compared with the axial turbine. The axial turbine is 192 described by a single stage mounted on the same disc, which limits the number of stages. Here, for low mass flow 193 range, low-temperature heat source and, hence a target application outputting only a few kW (5 kW- 15 kW), a 194 single stage is considered for both turbine configurations. In the axial turbine, the flow streamlines through the blade rows basically have a constant radius, compared with a considerable increase in radius through the blade rows in the radial-outflow turbine. It is evident from the aforementioned literature that there has been limited attention given to axial turbine and radial-outflow turbines. The industrial exploitation of radial-outflow configuration in the ORC application market is currently ongoing [35] and it is receiving more analytical studies by [30,31,32]. While the detailed studies are conducted of radial-inflow turbine that combined with mean-line design and ORC cycle analysis and followed by three-dimensional CFD simulation for a number of working fluids and operating conditions as reported by Sauret et al. [1,27,36] and other studies [2-11,15,17,21,22,26,29,33,34].

#### 202 4.1 Mean-line Design of axial and radial-outflow (centrifugal) turbines

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203 The mean-line design of the turbine stage is based on the 1D assumption that there is a mean streamline 204 through the turbine stage. The assumptions usually used in axial turbine design are applied to the radial-outflow 205 (centrifugal) turbine design. This novel design methodology is presented for the 1D mean-line design of centrifugal 206 turbine configurations in small-sized power output applications, borrowed from axial turbine design. The three 207 dimensionless parameters – loading coefficient, flow coefficient and reaction ( $\psi$ ,  $\phi$  and  $R_n$ ) were chosen in order to 208 predict the shape of the velocity triangle and the initial turbine efficiency for both configurations. The working fluid 209 enters the turbine through the stator at flow angle  $(\alpha_1)$  with absolute velocity  $(C_1)$  and exits at flow angle  $(\alpha_2)$  with absolute velocity (C<sub>2</sub>), as shown in Fig. 3. The relative velocity at the inlet of the turbine rotor is (W<sub>2</sub>) at angle ( $\beta_2$ ), 210 211 then the flow is accelerated to relative velocity ( $W_3$ ) at the outlet of the turbine rotor at angle ( $\beta_3$ ). The flow angles at 212 inlet and outlet of the turbine stage are calculated by [37]:

$$\begin{aligned}
\tan \beta_2 &= \frac{(\Psi - 2R_n)}{2\phi} \\
\tan \beta_3 &= \frac{-(\Psi + 2R_n)}{2\phi} \\
\tan \alpha_3 &= \frac{-(\Psi/2 - (1 - R_n))}{\phi} \\
\tan \alpha_2 &= \frac{(\Psi/2 + (1 - R_n))}{\phi}
\end{aligned}$$
(5)

In the centrifugal (radial-outflow) turbine, the blade chord and height had an influence on the distribution of the blade along the stage diameter of the machine. The stage diameter and the outlet section area are calculated as follows [32]:

$$D_{out} = D_{in} + b \tag{6}$$

$$A_{out} = H_{out}o = \frac{m}{\rho_{out}V_{out}N_{blds}}$$
(7)

217 Assuming a rectilinear suction blade end-side, the relationship between blade geometric discharge angle and outlet

$$o = S \cos(BDA) \tag{8}$$

where BDA is a blade geometric discharge angle, equivalent to  $\alpha_2$ ,  $\beta_3$  in Fig. 3.

220 Blade pitch S is calculated according to the following equation:

$$S = \frac{\pi D_{out}}{N_{blds}}$$
(9)

221 Blade height is calculated by rearranged equation (7) as:

$$H_{out} = \frac{m}{\rho_{out} V_{out} \cos(BDA) D_{out} \pi}$$
(10)

The original 1D mean-line code formulated in-house was developed based on the losses model by AMDCKO (Ainley and Mathieson, Dunham and Cam, Kacker and Okapuu), adopted to account for losses within the blade rows. Notably, the losses model is used to estimate the performance for both turbine configurations. The total pressure losses through the blade passage are expressed in terms of profile, secondary flow, trailing edge, and tip leakage losses, which are given by the following equation [38,39] and summarised in Table 3:

$$K_T = K_P f_{Re} + K_{Sec} + K_{TE} K_{TC} \tag{11}$$

227 The stage total-to-total and total-to-static isentropic efficiency in terms of enthalpy loss are as follows [40]:

$$\eta_{tt} = \frac{1}{1 + \left[\zeta_R W_3^2 / 2 + \left(\zeta_S C_2^2 / 2\right)(h_3 / h_2)\right] / (h_{01} - h_{03})}$$
(12)

$$\eta_{ts} = \frac{1}{1 + \left[\zeta_R W_3^2/2 + \left(\zeta_S C_2^2/2\right)(h_3/h_2) + C_3^2/2\right]/(h_{01} - h_{03})}$$
(13)

The pressure loss coefficient and enthalpy loss coefficient are approximately equal at a small value of enthalpy[38]. The full details of conversion from pressure loss to enthalpy loss are outlined in Moustapha et al. [39].

In the mean-line design of the outflow-radial turbine, the cascade losses are borrowed from axial turbines, such as those proposed by Ainley & Mathieson, Dunham & Came, Craig & Cox, and Kacker & Okapuu [38,39]. The losses model proposed by Craig & Cox [39,41] has been used by many researchers working on radial-outflow turbines, such as [30,31,32,42]. This losses model only includes the profile loss and secondary loss [32]. This paper considers end-wall losses, including the secondary, tip leakage and trailing edge losses [42], as detailed in Table 3.

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Solver software) code [43]. The EES code can discover a wide range of turbine configurations by accomplishing

inclusive studies in terms of different input parameters, as outlined in Table 5. The mean-line design methodology is

a highly iterative procedure; the flow chart in Fig. 4 shows the procedure for the turbine design methodology, while

the output of the mean-line design methodology is outlined in Table 6 for axial and radial-outflow (centrifugal)

turbines respectively. Fig. 4 shows the flow chart of the detailed 1D mean-line design and 3D CFD process carried

Table 5. Input parameters of mean-line design for axial and radial turbines and their ranges/values.

| For For States and States an | 8  | 0      |
|--|--|--------|
| Parameters   | Values/Range                                   | Unit   |
| Reaction (R <sub>n</sub> )   | 0.4-0.6  | -      |
| Loading coefficient $(\Psi)$   | 0.6-1.4  | -      |
| Flow coefficient ( $\phi$ )  | 0.2-0.8  | -      |
| Hub/tip radius ratio $(r_h/r_t)$   | 0.5-0.8  | -      |
| Rotational speed   | 18000-20000                                    | rpm    |
| Inlet total temperature  | 360  | K      |
| Degree of superheating   | 0-10   | K      |
| Inlet-total-pressure   | Corresponding saturated vapour                 | bar    |
|  | pressure at inlet temperature                  |        |
| Mass flow Rate   | 0.3 - 0.7                                      | kg/sec |
| Working fluids   | R141b, R245fa, R365mfc, n-butane and n-pentane | -      |

**Table 6.** Mean-line design output of the axial and radial-outflow turbines for  $\dot{m} = 0.7$  kg/s and five investigated working fluids

| Parameter R141b R245fa R365mfc Isobutane n-Pentane |        |               |        |         |         |  |  |
|--|--------|---------------|--------|---------|---------|--|--|
| Axial Turbine                                      |        |               |        |         |         |  |  |
| Tip diameter (d <sub>t</sub> ) mm                  | 68.47  | 66.75         | 63.07  | 70.51   | 73.82   |  |  |
| Hub diameter (d <sub>h</sub> ) mm                  | 46.17  | 46.39         | 44.33  | 46.31   | 48.0    |  |  |
| Blade height (H) mm                                | 11.15  | 10.18         | 9.37   | 12.10   | 12.91   |  |  |
| Tip clearance (mm)                                 | 0.35   | 0.35          | 0.35   | 0.35    | 0.35    |  |  |
| LE Blade Angle (deg)                               | -16.34 | -13.68        | -11.27 | -19.31  | -22.19  |  |  |
| TE Blade Angle (deg)                               | 65.80  | 63.75         | 63.50  | 65.25   | 70.50   |  |  |
| Stagger angle (deg)                                | 32.21  | 33.67         | 34.24  | 30.83   | 28.35   |  |  |
| Solidity (c/S) (-)                                 | 1.924  | 1.815         | 1.736  | 1.903   | 1.944   |  |  |
| Number of blade (-)                                | 25     | 21            | 21     | 25      | 27      |  |  |
| Turbine isentropic Efficiency %                    | 82.88  | 78.91         | 79.39  | 83.55   | 84.56   |  |  |
| Power output (kW)                                  | 14.55  | 13.054        | 13.885 | 15.052  | 15.947  |  |  |
|  | Radia  | l-outflow Tur | bine   |         |         |  |  |
| inlet diameter (D <sub>in</sub> ) mm               | 49.12  | 48.06         | 45.89  | 49.75   | 50.23   |  |  |
| Outlet diameter (Dout) mm                          | 93.66  | 86.52         | 81.52  | 102.72  | 108.72  |  |  |
| Blade height (H) mm                                | 11.94  | 11.26         | 10.295 | 12.86   | 13.625  |  |  |
| Tip clearance (mm)                                 | 0.35   | 0.35          | 0.35   | 0.35    | 0.35    |  |  |
| LE Blade Angle (deg)                               | -18.23 | -15.41        | -14.90 | -21.56  | -25.00  |  |  |
| TE Blade Angle (deg)                               | 67.10  | 65.50         | 65.25  | 68.40   | 69.50   |  |  |
| Stagger angle (deg)                                | 28.25  | 30.74         | 30.82  | 28.45   | 25.87   |  |  |
| Solidity (c/S) (-)                                 | 1.891  | 1.749         | 1.675  | 1.843   | 1.910   |  |  |
| Number of blade (-)                                | 30     | 28            | 26     | 30      | 34      |  |  |
| Turbine isentropic Efficiency                      | 81.04  | 77.58         | 79.07  | 80.25   | 82.00   |  |  |
| Power output (kW)                                  | 12.543 | 11.058        | 12.094 | 13.4225 | 14.4252 |  |  |



# 289 5. CFD Methodology

290 The CFD application becomes an essential step to investigate the ORC/turbines' performance and goes 291 hand-in-hand with their preliminary mean-line design due to the actual flow field in axial and centrifugal turbines' 292 being a strongly 3D, viscous and turbulent flow. Therefore, this section offers the 3D CFD analysis used in 293 predicting the aerodynamic performance of the axial and centrifugal turbines by conducting 3D CFD analysis across 294 the stator and rotor blade passage for both turbine configurations. The essential geometric characteristics (i.e. blade 295 height, inlet hub and tip radii, and angles of blade) of the applicant turbines, as presented in Table 6, are used to create the 3D geometry of the turbine stage (stator and rotor), utilising the ANSYS<sup>R17</sup>-BladeGen tool as shown in 296 297 Fig. 5. The pressure/suction and angle/thickness modes are employed respectively to define the curves for the hub, 298 shroud and blade profile for the stator and rotor blades.

The computational mesh is created using the ANSYS<sup>R17</sup>-TurboGrid meshing tool, tailored for CFD analysis via 299 300 hexahedral mesh and mostly based on an O-H grid. The adopted topology is constructed based on the H-type grid, 301 with the O-type grid added to increase the grid orthogonality around the blade. For mesh resolution purposes, the 302 computational mesh was increased by adding nodes in the hub-to-tip and blade passage (blade-to-blade) because of 303 the variation of  $y_+$ , where  $y_+$  is defined as boundary layer mesh size, which is a dimensionless distance from the wall and used to determine the first node away from the wall. The variation of y+ is determined by the first node 304 305 from the wall to a wetted surface in those two directions. The meshes are generated using the ATM optimised 306 algorithm; tip clearance is applied based on design and manufacture standards.

307 After the meshes are constructed, the 3D RANS equations with the k- $\omega$  SST turbulence model are solved using 308 the high resolution advection scheme. The turbulence model  $k-\omega$  SST is capable of automatic near-wall treatment to 309 capture the turbulence closure by determining y+. The value of y+ is required to be around unity, based on the k- $\omega$ 310 SST model as recommended in the CFX user's manual. The k- $\omega$  SST turbulence model is considered for flow 311 separation under an adverse pressure gradient, which accounts for the transport of the turbulent shear stress. 312 Turbulence intensity at the inlet was maintained at 5% – the recommended value when no information is available 313 about the inlet turbulence. The k-w transport equations carried out to find the turbulent kinetic energy and the 314 specific dissipation rate are:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j} \left( \Gamma_k \frac{\partial k}{\partial x_j} \right) + G_k - Y_k + S_K$$
(14)

$$\frac{\partial}{\partial t}(\rho\omega) + \frac{\partial}{\partial x_i}(\rho\omega u_i) = \frac{\partial}{\partial x_j} \left(\Gamma_k \frac{\partial\omega}{\partial x_j}\right) + G_\omega - Y_\omega + S_\omega$$
(15)

where  $G_k$  and  $G_{\omega}$  represent the generation of turbulent kinetic energy and its dissipation rate.  $Y_k$  and  $Y_{\omega}$  represent the fluctuating dilation in compressible turbulence.  $S_k$  and  $S_{\omega}$  are the source terms of the k- $\omega$  turbulence model.

The stage (mixing-plane) model is applied at the stator-rotor interface to equip connection (communication) across the domain of stationary and rotating blade rows. Periodic boundaries are applied for the stator and rotor blade passages. For stage analysis and the steady state flow, the GGI (Generalised Grid Interface) feature is employed in CFX setup. All CFD analyses were performed at steady state condition, and the convergence criterion of the CFX was equal to  $10^{-5}$  for all values of the residuals (RMS) with a time scale of  $0.5/\Omega$  as recommended in the CFX user's manual. The 3D CFD analysis of ORC turbines requires an accurate thermodynamic model to account for the variations in the properties of organic fluids. Therefore, the thermodynamic properties of the organic fluids were obtained using REFPROP software. The boundary conditions from the 1D mean-line design in Table 5, such as inlet total pressure and temperature, rotational speed, and mass flow rate are used to perform the 3D CFD simulations via ANSYS<sup>R17</sup>-CFX.

Grid independence was performed for the stator and rotor to ensure the meshes were sufficient in size. The initial mesh was generated and the 3D CFD solution completed; the turbine isentropic efficiency and dimensionless distance y+ were calculated. The meshes were then clustered and the simulation re-run and repeated until the grid-independent solution was achieved. The grid independence study is presented in Fig. 6 for both turbine configurations and summarised in Table 7 for all investigated working fluids and both turbine configurations. The computational meshes of the blade passage for both the stator and rotor blades are outlined in Fig. 7.

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**Table 7.** Summarised the grid size and the corresponding  $y^+$  values for both turbines and each working fluid.

| Working   | Axial turbine |             | Radial | Radial-outflow Turbine |             |        |
|-----------|---------------|-------------|--------|------------------------|-------------|--------|
| fluide    | Stator No.    | Rotor No.   | $y^+$  | Stator No.             | Rotor No.   | $y^+$  |
| IIulus    | of Elements   | of Elements |        | of Elements            | of Elements |        |
| R141b     | 450000        | 550000      | 0.9325 | 350000                 | 600000      | 0.9532 |
| R245fa    | 425000        | 525000      | 0.8951 | 300000                 | 575000      | 0.9257 |
| R365mfc   | 450000        | 500000      | 0.8612 | 300000                 | 550000      | 0.9125 |
| Isobutane | 450000        | 580000      | 0.9570 | 375000                 | 650000      | 0.9866 |
| n-Pentane | 475000        | 625000      | 1.025  | 410000                 | 650000      | 1.036  |



Fig. 5. 3D geometry for axial turbine stage (left) and radial-outflow turbine stage (right).



#### 377 6. CFD verification

378 The developed mean-line design described in section 4.1 is validated against published benchmark case namely 379 the Glassman case (code) as detailed in [44] for axial turbine configuration. The mean-line design results in terms of 380 total-to-total efficiency and power output (i.e. the global performance parameters) are in a good agreement with the 381 Glassman case and the deviations are within the acceptable margin for all working fluid as demonstrated in Fig. 8. 382 Furthermore, because of the lack of available experimental data for small-scale axial and radial-outflow turbines 383 operating with organic fluids, verification of the present 3D viscous simulation for both turbine configurations is 384 made against the mean-line design results at nominal boundary conditions (Table 5), as shown in Figs. 8 and 9. The 385 total-to-total isentropic efficiency and power output are compared for five organic fluids. The maximum difference 386 in efficiency between the CFD and preliminary mean-line design (PD) for the axial turbine was 3.92% with R141b 387 as the working fluid, while the maximum difference in the radial-outflow turbine was 4.61%, with R245fa as the 388 working fluid. The variance between mean-line design PD and CFD is mostly attributable to 1D-characteristic PD, 389 which is not able to capture all features of 3D flow fields. It may further be attributed to the details of the CFD 390 analysis of the turbine using 3D CFD modelling. Ultimately, these results showed better agreement than the 391 comparison between mean-line design and 3D CFD in Ref. [26,29], where the deviation in turbine isentropic 392 efficiency was around 6-9%.





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Fig. 9. Comparison between mean-line design and CFD for radial-outflow turbine.

#### 414 7. CFD Results

415 The 3D CFD analysis of small-scale axial and radial-outflow (centrifugal) turbines was performed in both 416 nominal conditions (Table 5) and off-design conditions for the five working fluids. The evaluation of small-scale 417 turbine performance (total-to-total isentropic efficiency and power output) with total-to-total pressure ratio is shown 418 in Figs. 10 and 11 for both configurations. However, the best turbine isentropic efficiency is obtained at a pressure 419 ratio of around 2.25, due to a lower mass flow rate. As can be seen in Figs. 10 and 11, at a pressure ratio of around 420 2.25, the maximum turbine isentropic efficiency was 82.5% with n-pentane as the working fluid for the axial 421 turbine, compared with 79.05% for the radial-outflow turbine with the same fluid. R245fa has a minimum turbine 422 isentropic efficiency of 76.25% and 74% for axial and radial-outflow turbines respectively, due to its being a heavier 423 (higher density) fluid. Also, Figs. 10 and 11 show that lighter fluids (n-pentane and isobutane) can produce 424 considerably higher power outputs compared with the heavier (high-density) fluids (R245fa, R365mfc) for both 425 turbine configurations. Lighter organic fluids (light molecular weight as n-pentane and isobutane) can produce 426 substantially higher power outputs compared with heavier fluids (fluids with high molecular weight/high density, 427 such as R245fa and R365mfc) for the same mass flow rate of organic working fluid, due to a relatively larger turbine 428 size and consequently higher specific work output, whereas the lighter organic fluids have a higher enthalpy drop 429 and consequently larger specific work. The high-density organic fluids such as R245fa and R365mfc have smaller 430 sizes due to their lower specific volumes and thus require a higher pressure ratio and rotational speed to achieve the 431 same power output as the lighter fluids.

The maximum power was 15.15 kW for the axial turbine and 13.625 kW for the radial turbine with n-pentane as the working fluid. With the rise in pressure ratio, both isentropic and actual enthalpy increase, leading to higher power output based on the definition of loading coefficient.



Figs. 12 and 13 show that maximum efficiency and power output were obtained at design rotational speed: 18,000 rpm for R141b, R245fa and R365mfc and 20,000 for lighter fluids (isobutane and n-pentane) for both turbine configurations. A maximum difference of 6.25% between the turbine efficiencies of n-pentane and R245fa in the axial turbine, compared with 5.05% in the radial-outflow turbine was predicted. From Fig. 13, the maximum power

459 output was with n-pentane for both turbine configurations. R245fa had the lowest power output for both turbine



460 configurations, due to its high density.



6.25% for the axial turbine, compared with 5.05% for the radial-outflow configuration at a mass flow rate of 0.7
kg/s. As shown in Fig. 15, better performance is achieved with lighter fluids such as n-pentane and isobutane, with
15.15 kW and 13.625 kW for the axial and radial-outflow configurations respectively.



The effect of working fluid mass flow rate on the overall size (tip diameter of the axial turbine and outer diameter of the radial-outflow turbine) is substantial, as depicted in Fig. 16. With rising mass flow rate, the drop in actual enthalpy increases, leading to a larger overall size based on loading coefficient. It can be seen in Fig. 17 that lighter organic fluids can yield considerably higher power output (more than 19.90%) compared with high-density

organic fluids, due to their lower specific volumes and thus smaller sizes, while lighter fluids lead to larger blade
height for both turbine configurations. Consequently, larger overall size and power output are achieved, as shown in
Figs. 16 and 17.



highest pressure values correspond to the lowest flow velocity. By contrast, the lowest pressure values are located on the suction side due to the highest flow velocity values at the throat area of the blade passage. The isentropic enthalpy drop (work) is provided by the area circumscribed by such pressure distribution curves, where the enclosed

area is indicative of the net torque producing aerodynamic force by the rotor turbine shaft. Fig. 19 shows the Mach number contour at 50% span. It is marked by low Mach number values, which are due to the high profile curvature in this zone. The maximum Mach is 0.95 at the exit of the axial turbine's stator, as shown in Fig. 19, with n-pentane as the working fluid.

Pressure (Pa) 000005 100000 Axial Turbine Radial-outflow Turbine 0 0.2 0.4 0.6 0.8

**Fig. 18.** Blade loading chart at rotor mid-span for both turbine configurations with n-pentane as the working fluid.

Streamwise (0-1)



- **Fig. 19.** Mach number at mid-span for both turbine configurations with n-pentane as the working fluid at nominal condition.

#### 567 8. ORC system analysis results

568 The assumption of constant arbitrary turbine isentropic efficiency, ignoring the possibility of the turbine's 569 reaching these efficiencies in realistic conditions, was investigated for different working fluids and a wide range of 570 boundary conditions. This does not essentially produce accurate results when each working fluid exhibits different 571 turbine performance in specific operating conditions, as shown in the previous section. Performance parameters such 572 as turbine isentropic efficiency and power output obtained from the 3D CFD analysis for each working fluid were 573 then inserted as the inputs into the ORC's model to calculate the ORC efficiency at nominal operating conditions, as 574 shown in Fig. 20 (Table 4). Inlet total temperature, mass flow rate and rotational speed are 360 K, 0.7 kg/s, and 575 20,000 rpm respectively.

576 It is evident that the axial turbine configuration reached about 11.74% compared with 10.25% for the 577 radial-outflow configuration with n-pentane as the working fluid due to its having the maximum turbine efficiency 578 and power output in both configurations. However, R245fa has the lowest cycle thermal efficiency, at around 9.15% 579 for the axial configuration and around 8.10% for the radial-outflow configuration, because of the low turbine 580 isentropic efficiency and power output compared with other working fluids, as shown in Fig. 20. The evaluation of 581 second law efficiency for each working fluid and both configurations is presented in Fig. 21, at the same nominal 582 operating conditions as above. The highest second law efficiency is for n-pentane, while R245fa has the lowest 583 second law efficiency of the working fluids because it has the lowest turbine performance, leading to lower second 584 law efficency. These results are better than in other reported studies, such as Ref. [15, 16 and 20], with maximum 585 cycle thermal efficiency 6.8% as reported in [15], and highlight the potential of this integrated approach for further 586 accurate prediction of ORC performance depending on small-scale axial and radial-outflow (centrifugal) turbines.







Figs. 22 and 23 show the variation in ORC thermal efficiency with the inlet total temperature of the turbine for the five different organic working fluids at mass flow rate of 0.7 kg/s and rotational speed of 20,000 rpm. As expected, the highest ORC thermal efficiency is detected at an inlet temperature of 360 K, as a design point where the maximum turbine isentropic efficiency and power output leads to maximum cycle thermal efficiency for both turbine configurations.



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#### 623 9. Conclusions

Limited consideration has been given to 3D CFD analysis, which is important to obtaining the accurate peak performance of the ORC turbine. This paper is the first to offer the full design procedure of efficient small-scale axial and radial-outflow (centrifugal) turbines working with sensible and realistic low-temperature heat sources such as solar and geothermal energy, with low mass flow rate and a range of organic fluids (R141b, R245fa, R365mfc, isobutane and n-pentane) for different ORC power generation applications.

629 Furthermore, there is limited literature concerning the design and 3D CFD analysis of ORC systems, based on 630 small-scale axial and radial-outflow turbines with power output up to 15 kW, for different electricity generation 631 applications, such as small buildings, rural areas, off-grid zones and isolated installations. The key contribution of 632 this paper is its development and demonstration of axial and radial-outflow turbine design and 3D analysis 633 integrated with ORC modelling for low-temperature heat source and small-scale power output applications. The 634 purpose of using 3D CFD analysis hand-in-hand with mean-line design is to predict turbine performance at the 635 design condition, allowing a comparison between the proposed turbine and the actual performance achieved by the 636 CFD model. The CFD results revealed a substantial difference in turbine efficiency of 6.25% between n-pentane and 637 R245fa for the axial turbine, compared with 5.05% for radial-outflow. The maximum turbine isentropic efficiency 638 was 82.5% and 79.05% for axial and radial-outflow turbines respectively, with respective power output of 15.15 kW 639 and 13.625 kW. The large overall size of the axial turbine was 73.82 mm as a tip diameter compared with 108.72 640 mm for the radial-outflow turbine.

The maximum cycle thermal efficiency of 11.74% by the axial turbine, compared with 10.25% by the radial-outflow turbine, is achieved with n-pentane as the working fluid. These results highlight the advantages of this integrated approach of using axial and radial-outflow in the ORC system compared to other literature. Finally, the 3D blade shape optimisation tool integrated with structural analysis will be developed and applied to these

645 turbines in aviable low-temperature heat source applications.

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