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Development of Three-dimensional Optimization of a Small-scale Radial Turbine for Solar Powered Brayton Cycle Application

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10 Abstract

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11 Numerical simulation was carried out to optimize the design of a small-scale radial turbine. Onedimensional (1D) Mean Line (ML) approach and three-dimensional computational fluid dynamic (3D 12 13 CFD) simulations, using 3D Reynolds-Averaged Navier-Stokes (RANS) models with the shear stress 14 transport (SST) turbulence model in ANSYS®15- CFX, were employed to achieve the best turbine 15 performance and consequently cycle efficiency. For the current study, a new methodology that 16 integrates the Brayton cycle analysis with modelling of a highly efficient small -scale radial turbine at 17 a wide range of inlet temperatures was developed. A multi-objective function was utilized for 18 optimizing the designed radial turbine power in the range of 1.5 to 7.5 kW. This method has been 19 developed in order to find the optimum design, from an aerodynamic point of view. After applying a 20 well-designed range of parameters for both the stator and the rotor, the results demonstrated an 21 excellent improvement in the turbine efficiency from 82.3% to 89.7% for the same range of output 22 power. Moreover, the effect of the turbine inlet temperature, rotational speed and pressure ratio was 23 further studied and presented in this paper. Finally, the overall cycle efficiency showed an excellent 24 improvement of about 6.5% for the current boundary conditions; and it yielded more than 10% with 25 the increase in the inlet temperature and the pressure ratio. Such results highlight the potential and the 26 benefits of the suggested methodology to achieve a high performance (i.e. turbine efficiency and cycle 27 efficiency).

Keywords: Small scale radial turbine, Solar Brayton cycle, CFD analysis, 3D Optimization, Genetic
 algorithm.

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31 1- Introduction

The demand for energy is continually increasing day after day, but at the same time, investigations around the world into sustainable sources of power are growing in number. Solar energy is considered one of the main renewable energy sources which can play an important role in decreasing CO2 emissions. It can be efficiently used to generate electricity using different types of thermal power cycles, such as the Brayton cycle.

38 Moreover, small scale turbines are considered as a promising technology because of their low initial 39 costs, low maintenance, durability and simple construction. Furthermore, they can offer a solution for 40 the power generation demand in domestic or even remote areas. In order to increase the cycle 41 efficiency, one of the main effective ways is to improve the turbines' performance.

42 Much research has been carried out regarding both the solar Brayton cycle thermodynamic analysis 43 and the selection of the appropriate boundary conditions of energy as heat sources such as [1-4]. For 44 example, an attempt to enhance the overall efficiency of the small-scale solar Brayton cycle, by 45 optimizing both the receiver and the parabolic concentrator, has been achieved by Le Roux at al. [5]. 46 Riazi and Ahmed [6] studied the effect of specific heat ratios for three different working fluids and for 47 air, helium and tetrafluoromethane, on the efficiency of small scale solar energy. A regenerative

48 closed Brayton cycle was analysed in terms of the influence of temperature ratio and the minimum to 49 maximum gas temperature. Their results showed that the higher the specific ratio of the analysed 50 fluid, the higher the cycle efficacy. Moreover, they also suggested that for small-scale Brayton cycles, the performance of lower specific ratios is better as this scale only accumulates a small amount of 51 52 heat. However, the performance of turbines was not included in all the studies mentioned above as on 53 the shelf turbines were used. Two important parameters have to be carefully considered, as they lead 54 to better preliminary design. Both the loading coefficient and the exit flow coefficient contribute in 55 [7]. Intensive analyses in order to enhance the performance of scroll expander were conducted in [8-56 10]. Mean line analysis for radial turbines for organic Rankin cycle applications was achieved by 57 many researchers [11-16]. However, no more 84% Total-To-Static efficiency for the studied models 58 of radial turbine have been attained [12]. It is clear that the 3D computational fluid dynamics (CFD) 59 leads to more enhancement in the aerodynamic performance. Three-dimensional optimization design 60 work on ORC radial turbines was demonstrated in [17-19]. On the other hand, modelling and 61 optimization of axial turbines was conducted in [20-21]. Sauret [22] carried out intensively numerical 62 work on a high pressure radial turbine. The author started his study from the preliminary design 63 passing through 3D simulation; then the impact of tip clearance as well as the importance of the 64 diffuser on the turbine's performance was examined at wide range of boundary conditions. The author 65 then validated her design against some experimental data from the literature. Regarding the 66 compressed air radial turbine, it can be seen that only some efforts have been made by different 67 researchers who studied the performance of radial turbines with different design factors as well as boundary conditions. With the aim of identifying the possible acoustic sources which occurs, Marsan 68 69 and Moreau [23] studied the effect of stator wakes and trailing edge on the impeller blades of radial 70 turbine. In their study the authors showed that the interface regime between the stator and rotor causes 71 pressure fluctuations to the flow passes through these surfaces. They emphasized that the tip clearance 72 losses should not be neglected in analytical studies. Together CFD and FE analysis of relatively high 73 pressure ratio and low inlet temperature, 5 bar and 400 K, radial turbine was conducted in [24]. The 74 output power and efficiency of the investigated turbine were 36.4 and 85% respectively. With 75 comparatively accepted deviation, the authors validated their simulation work using an experimental 76 data. For compressed air turbine optimizations, Tsalicoglou1 and Phillipsen [25] used an iterative 77 method that conducted in-house and commercial CFD codes to decrease the amount of working fluid 78 mass flow rate. This results in changing the turbine blade geometry modifications with some 79 improvement in efficiency. However, with the limitation of the allowed number of design parameters' 80 evaluations, multiple runs were required. As a single objective function, particle swarm optimization was used in their study. An integrated optimization of a 100 kW radial turbine was reported by Lei Fu 81 82 et al [26]. In their study, an enhancement in terms of aerodynamic and structural performance was 83 achieved. However, the optimization carried out on the rotor while the stator was not considered in 84 their study. Also, each part of optimization was completed separately in two different codes and that 85 might be the reason for the relatively low maximum efficiency value that has been achieved, 82%. 86 Zhang and Ma [27] used the multi-objective algorithm technique for optimizing only the rotor of a 87 radial turbine. Even though the authors claimed the optimized rotor experienced better performance 88 especially in off-design conditions with about an 8% increment in its efficiency, the maximum value 89 that the turbine reached was also low, about 77%.

90 An intensive study on the relation between optimization of computational time and the chosen range 91 of the database, as well as the selection of the suitable optimization method was reported in [28]. In 92 his study the author emphasized that selecting the closer setting to the optimum parameters not only 93 results in further improvements of the convergence, but also contributes to better rotor geometry 94 performance. Three-dimensional multi-objective optimization for a turbocharger radial turbine 95 impeller designed for automotive applications was applied in [29]. With the aim of maximizing its

total-to-static efficiency and the impeller moment of inertia, and at the same time keeping mechanical
stresses below a maximum allowable value, the maximum value of efficiency reached was only 80%.

To the best of the authors' knowledge no work has been published on the 3D multi-objective optimization of the two main parts of radial turbines: the stator and rotor, together especially for this scale of turbines. So, this paper tries to focus on design; such as turbine design which creates as high as possible efficiency and output power to the cycle but with keeping both the rotational speed and the mass flow rate at their minimum values.

104 105

106 2- Methodology

107

108 In this study the computational fluid dynamic CFD techniques, using ANSYS®15 VISTA, was 109 utilised in order to first initiate a Mean Line (ML) design for the Small Scale Radial Turbine SSRT 110 with relatively sufficient performance. Then, 3-D model was improved using ANSYS®15 CFX tool, 111 which precisely figure out the aerodynamic flow behaviour, analyse were followed in order to have 112 more accurate and better outcomes for the SSRT. After the best design shape for both the stator and 113 rotor was achieved, a number of the most influences parameters were determined and chosen with the 114 aim of optimizing the full blades shape of the SSRT using the 3D Design Exploration feature of 115 ANSYS®15 which employed the genetic algorithm for the multi-objective optimization. Moreover, 116 the results have been directly integrated with the Brayton cycle code which was initiated using the 117 Engineering Equation Solver (EES) software [30].

118

119

120 **3-** Thermodynamic Analysis of Brayton Cycle

121 The traditional thermal Brayton cycle consists of the compressor, the combustion chamber, 122 the turbine to extract the air's potential energy and transfer it to mechanical energy; and a pre-heater 123 to exploit the exhaust energy, which will be otherwise lost to the environment and also to preheat the 124 cold air before entering the source of heat. However, the solar powered Brayton cycle shown 125 schematically in Fig. 1, consists of a compressor (1-2), thermal receiver (3-4) and a turbine (4-5). A 126 recuperator, (2-3), is used to recover heat from the turbine exit's hot air. The current study aimed to 127 be fitted with the application of small scale solar powered Brayton cycle which of course its 128 efficiency can be enhanced by improving the efficiency of its component. The compressor power is 129 given by [5]:

130

$$V_{\rm C} = \frac{Cp \, \overline{T}_1(R_C K - 1)}{\eta_{\rm C}} \tag{1}$$

131 Where the compressor pressure ratio equals $R_c = P_2 / P_1$ and in contrast the turbine pressure ratio is: 132 $R_t = P_4 / P_5$

133 The amount of heat supply by the solar receiver per unit mass of working fluid flow is:

$$Q_{\rm Net} = (T_4 - T_3)$$
 (2)

The heated working fluid exits from the solar receiver and passes through the turbine to generate power; the power output from the turbine is given by:

$$W_{\rm T} = Cp \,\eta_{\rm T} T_4 (1 - R_T - K) \tag{3}$$

136 If the pressure loss coefficient is defined to be X, the above formula can be written as:

$$W_{\rm T} = Cp \,\eta_{\rm T} T_4 (1 - (XR_c) - K)$$

- 137 The exhaust's working fluid exits from the turbine to the atmosphere and on its way it will pass
- through the recuperator. The heat gained by incoming compressed air and the heat rejected through
- the leaving air is given by the next two equations respectively:

140
$$Q_c = \dot{m}(h_2 - h_3)$$

 $Q_{cRej} = \dot{m}(h_5 - h_2)$

141 The extent to which a recuperator approaches an ideal recuperator is called the effectiveness, ε , and is 142 defined as:

143
$$\varepsilon = \frac{H_3 - H_2}{H - H_2}$$

144 The net power output from the cycle is given by:

$$W_{net} = W_t - W_c$$

145 This can also be written as:

$$W_{net} = Cp \left[\eta_{\rm T} T_4 (1 - ({\rm XR}_c) - {\rm K}) - \frac{T_1 ({\rm R}_c {\rm K} - 1)}{\eta_{\rm C}} \right]$$
(9)

146 The thermal Brayton cycle efficiency is given by:

$$\eta_{th} = \frac{W_{net}}{Q_{Net}} \tag{10}$$

148 The above equation can be formulated in terms of temperatures and defined as the following:

147

$$\eta_{th} = \frac{\eta_t T_4 \left(1 - (XR_c)^{-K} \right) - T_1 \left(\frac{R_c^{K-1}}{\eta_c} \right)}{T_4 \left(1 - \varepsilon \left\{ 1 - \eta_t \left(1 - (XR_c)^{-K} \right) \right\} \right) - T_1 \left(1 - \varepsilon \right) \left(1 + \left(\frac{R_c^{K-1}}{\eta_c} \right) \right)}$$
(11)

150

As it is shown in Fig. 1 the cycle consists of different components and each one of these component needs to be carefully designed. The selection of the best design parameters of the turbine will lead to higher turbine isentropic efficiency, power output. This certainly will enhance the overall efficiency of the cycle and the system performance. Fig. 2 illustrates their T-S (temperature-entropy) diagram.

150 ui 157

158 159

4- Mean Line Design of Radial Turbine [12, 31-41]:

The initial shape blade as well as its dimensions such as nominal hub and shroud diameters, the blade number and thickness, the trailing and leading edges can be determined using the ML design [41]. Together Figs. 3 &4 show the velocity triangles and their relative thermodynamic processes. The two dimensionless parameters that have been used in the ML design of radial turbine are the loading (ψ) and flow (ϕ) coefficients. These two parameters are together used to determine the exact velocity triangle shapes and then calculate turbine efficiency through the stage as shown in equations 12-14.

(5)

(6)

(7)

(8)

(4)

CRIPT ED)

$$\Psi = \frac{\Delta h_{actual}}{U_4^2} \tag{12}$$

167

The relations that connect hub and tip diameter in radial turbine as well as the rotor number of vanes 168 169 are:

$$r_{5_{t}} = \sqrt{\frac{A_{5}}{\pi} + r_{5hub}^{2}}$$
(15)
$$Z_{rotor} = \frac{\pi}{20} (110 - \alpha_{2}) \tan(\alpha_{2})$$
(16)

170

$$Z_{rotor} = \frac{1}{30} (110 - \alpha_2) \tan(\alpha_2)$$
(16)

172 The losses in enthalpy due to the tip clearance are consisting of both the axial and radial clearance. 173 So, these clearances as well as the secondary losses are calculated by the next few equations 17-23,

$$\Delta h_{tip,clearnace} = \frac{U_3^4 Z_{rotor}}{8\pi} \left(0.4 \,\varepsilon_x C_x + 0.75 \,\varepsilon_r \,C_r - 0.3 \,\sqrt{\varepsilon_x \varepsilon_r C_x C_r} \right) \tag{17}$$

174

$$C_x = \frac{1 - \left(\frac{r_{5tip}}{r_4}\right)}{C_{m4} b_4}$$
(18)

175

$$C_r = \left(\frac{r_{5tip}}{r_4}\right) \frac{l_{rotorx} - b_4}{C_{m5} r_5 b_5}$$
(19)

 $l_{rotorx} = 1.5 (r_{5tip} - r_{5hub})$ (20)177

$$\varepsilon_x = \varepsilon_r = 0.04 (r_{5tip} - r_{5hub}) \tag{21}$$

$$\Delta h_{secondary} = \frac{C_4^2 d_4}{Z_{rotor} r_c} \tag{22}$$

179

182

$$C_{x} = \left[\overline{Re} \left(\frac{d_{4}}{2r_{c}}\right)^{2}\right]^{0.05} \left[\frac{W_{4} + \left(\frac{W_{5tip} + W_{5hub}}{2}\right)}{2}\right]^{2} \frac{l_{hyd}}{h_{hyd}}$$
(23)

180 The exit kinetic loss can be computed using the following equation,

181
$$\Delta h_{exit} = 0.5C_5^2 \tag{24}$$

183 In nozzle and volute, on the other hand, after obtaining the friction factor f from moody chart, the differences in the enthalpy because of the friction effect, and Reynolds number can be determined by 184 185 the next two equations:

$$\Delta h_{friction,nozzle} = 4 f_{nozzle} \bar{C}^2 \frac{l_{hyd,nozzle}}{d_{hyd,nozzle}}$$
(25)

$$f = 8 \left[\left(\frac{8}{Re}\right)^{12} + \left(\left[2.475 \ln \left(\frac{1}{\left(\frac{7}{Re}\right)^{0.9} + 0.27PR}\right) \right]^{16} + \left[\frac{37530}{\overline{Re}}\right]^{16} \right)^{-1.5} \right]^{\frac{1}{12}}$$
(26)

(27)

(28)

(29)

186

After determining the fluid mass flow rate, its density and absolute velocity, the maximum radius and 187 188 volute radius can be defined using the equations from 27-29,

189

$$A_{1} = \frac{m_{working fluid}}{\rho_{1}C_{1}}$$
$$r_{volute} = \sqrt{\frac{A_{1}}{(0.75\pi + 1)}}$$

190

 $d_{max} = 2(r_1 + r_{volute})$ 191

The losses which are associated with volute geometry and then the total losses in enthalpy and the 192 193 total to total efficiency are defined in the last three equations:

$$\Delta h_{loss,volute} = \frac{r_{volut} C_2^2}{2} \tag{30}$$

194

195

 $\Delta h_{loss,total} = \Delta h_{loss,volute} + \Delta h_{friction,nozzle} + \Delta h_{tip,clearance}$

$$+\Delta h_{secondary} + \Delta h_{frictio} + \Delta h_{exit}$$
(31)

$$\left(\eta_{turbine,stage,ts}\right)_{new} = \frac{\Delta h_{actual}}{\Delta h_{actual} + \Delta h_{loss,total}} \tag{32}$$

The designer can easily get the main specifications and thereby the ML deign of turbines. These 196 197 specifications, in brief, include Inlet boundary conditions such as pressures, temperature, pressure 198 ratio, and output power as well as design parameters like, flow coefficient, loading coefficients, hub to 199 tip radius ratio and number of blade. Finally, estimation the initial estimations of the overall efficiency 200 initial, tip clearance. Fig. 5 illustrates the main steps of the ML design for the SSRT in a flow chart. 201 Table (1) shows the ranges of input factors and boundary of the radial turbine ML design. The results 202 of the 3D base-line BL of the SSRT geometry from the ML design are shown in Table (2).

203 5- Numerical Analysis of the model

204 Once the ML design of the SSRT, using the Vista RTD tool, was completed, all the relevant 205 information for building the blade geometry of the rotor was prepared. So, the next step was to export 206 these dimensions in order to create the rotor blade geometry using the Blade-Gen feature in ANSYS 207 CFX. This tool in fact can be used to construct the stator blade as well as the rotor for the studied 208 turbine. The CFX Turbo-Grid was employed in order to generate the required elements for the 209 turbine's fluid domain. The 3D turbulent viscous flow simulation in the whole domain of the SSRT 210 was functioned using the Shear Stress Transport turbulence model with equations of Navier-Stokes.

211 The assumptions of the 3D CFD simulation are steady state, using a compressible single-phase ideal 212 air gas. Also, the first order upwind advection scheme was selected because it is numerically stable.

213 The SST/k-omega has the ability to treat the low velocity region near the wall through an automatic 214 wall function, to capture the turbulence zone by the specified first node after the wall. The SST/K-

omega model is specifically designed for the complex flow like turbomachinery flow as suggested by[43]. The flow direction was set to be normal to boundary and the average value of y+ was set aside

217 unity as recommended in the CFX-Solver theory guide [43]. The convergence criteria for the residuals

of the continuity, energy and the velocity equations were of the order of 10^{-6} . The solutions for all the

219 cases were obtained once the results of the convergence criteria were satisfied.

220 The CFD simulations were established in order to analysis the SSRT behaviour at both; the nominal 221 and the other off-design conditions. Figure 6A displays the full SSRT as created by the Blade-Gen 222 CFX; while Fig. 6b presents the density of the chosen mesh passage for the rotor blade. The rotor grid 223 number was about 1,200,000 nodes and the stator grid number was around 600,000 using a finer mesh 224 for the adjacent walls of blades. During the optimization procedure 15 parameters have been 225 nominated in order to achieve the required objective function. Those parameters are: the rotor blade 226 number; the rotor blade shape, which is represented by 12 parameters; the stager angle of the stator; 227 and the stator blade number. More details are found in section 6.

228

229 6- Validation

Because they have relatively sufficient data, the references [46 & 47] have been chosen and deeply 230 231 investigated in order to validate the current work. The validation results showed good agreement 232 between the current work and the two chosen ones. While the first validation, which was with 233 reference [46], the CFD technique was used in order to validate the current work, however, in the 234 second validation, with reference [47], only the 1Dimensional approach was used because relatively 235 little information was provided about the 3Dimensional approach. Regarding the mesh sensitivity, it 236 worth to mention that the turbine model had around 1,000,000 nodes for stator and 2,000,000 for the 237 rotor. It is worth noting that in the zone near to the walls and blade's surface, the grid size was refined 238 in order to sustain a good agreement between computational costs and solution accuracy. The grid 239 sensitivity analysis was carried out based on turbine total to static efficiency as shown in Fig. 8 in 240 order to reach the satisfied number of elements for the chosen mesh type.

241

242 The schematic view of the radial turbine that used in Ref [47] is presented in Fig. 7.

243

Furthermore, Figs. 9A and B showed the comparison between the present work and the two mentioned
 journals respectively.

The authors referred to the secondary losses of the stator, friction losses of the vaneless-space surface, and the mixing trailing edge and wake losses as reasons for the uncertainty of their results. However, the uncertainty of Ref.[45] was because of the difference in the operating conditions of their rig; especially regarding the hot and normal clearances of the rotor, which increases the tip clearance losses.

252 7- Results of the Base-line Design

253 Figure 10 compares the CFD results of the base-line design against the ML design analysis at nominal 254 boundary conditions, shown in Table (1). The values of output power and isentropic efficiency were 255 compared for three different cases: inlet temperature, rotational speed and pressure ratio. Specifically, 256 Figs. 10A and 10A' show the values of output power and efficiency at different values of turbine inlet 257 temperatures. In this figure the relative over estimation for the ML analysis in terms of both the 258 turbine output power and the efficiency can be noticed. The main reason for this is the inability of the 259 1D analysis to elicit the exact behaviour of the fluid. Similarly, Figs. 10B, 10B', 10C and 10' represent 260 the turbine output power and efficiency at different values of its rotational speed and pressure ratio 261 respectively. It can be seen that at each specific boundary condition the output power and efficiency 262 of the turbine reached the peak. Furthermore, this figure shows that the maximum difference in

efficiency between the PD and CFD results was about 9% for the case of different inlet temperatures
and 6.9% in output power for the case of different pressure ratios because the ML design is not able to
capture all properties that the real flow behaves.

266 8- Multi-objective Optimization and Genetic Algorithm

267 Engineers still need to search for the best design among the available possible designs. Yet, the term 268 'best' can come with many meanings and what is excellent in some terms or applications may not be 269 the best in other applications; thereby this term does not have an absolute meaning. Therefore, 270 understanding the optimization procedure in depth will certainly lay the groundwork for optimization 271 of the turbines; especially the Small Scale Turbines (S.S.T.) whose sizes might add to the challenge of 272 their optimization. The term fitness in nature can be represented, from the engineering point of view, 273 as the most robust design. From this point the idea of initiating an Optimized Small Scale Radial 274 Turbine (O.S.S.R.T.) began and is presented in this paper. The operation of the GA starts with a 275 population of random strings and the design variables are represented by these strings. Thereafter, 276 each single string will be assessed to discover the fitness value. Three main well-known operators – 277 reproduction, cross over, and mutation will be run to create a new population of points and then drive 278 the population. The new population will be further assessed and tested in order to terminate the 279 process. However, if the termination criterion is not met, the population will iteratively run using the 280 three mentioned operators and again be evaluated. This procedure will be continued until the 281 termination criterion is reached. The structure of the algorithm is shown in Fig. 11. In this study the 282 objectives include the efficiency, the power output of the turbine, the mass flow rate of the working 283 fluid and the rotational speed. On the other hand, some constraints such as the stator throat area (to 284 deliver the required mass flow rate of air to the rotor), the tip clearance and blade thickness (for 285 manufactural purposes), are also provided.

In this study the multi-objective function was harnessed with the aim of optimizing the designedradial turbine.

288 Comparing it to single objective optimization, the multi-objective has the ability to maximize or 289 minimize many functions, depending on their constraints, simultaneously.

290

291 Differences between the Genetic Algorithm (GA) methods and other traditional optimization

292 methods:

293 1- Genetic algorithms (GA) work with the coding of the parameter set, not the parameters themselves.

- 294 2- GAs search for a population of points, not a single point.
- 295 3- GAs use the objective function information and not the derivative or second derivative.
- 296 4- GAs use stochastic transition rules, not deterministic rules.
- 297

298 8.1- Aerodynamic Optimization of SSRT

299 The design exploration package, which is linked in the CFD analysis, is based on the quadratic 300 Response Surface Method (RSM). This method is one of three popular techniques (Kriging, Radial 301 Basis Functions and polynomial Response Surface Model) in surrogate models. This method, which 302 has the best compromise between the computational expense and modelling accuracy, is provided 303 with a polynomial model. In this study, parameterization of the blade geometry and generation of the 304 design points, using the Design of Experiments (DoE) and design of exploration package in 305 ANSYS[®]15 was carried out. As the multi-objective optimization enables user to choose more than 306 one objective, each; power output and total efficiency of the turbine together with mass flow rate and 307 rotational speed were chosen as objective functions. Then, the response surface approximation (RSA)

308 [41], which is a statistical functions that connects the output parameters in terms of input parameters
309 (blade profile), will be employed using ANSYS®15. The GA is employed for global exploration.
310 After completing the numerical solution for each single design point, the discrete response for each of

them was obtained. The second order polynomial response can be formulated [44, 45] as:

$$f(x) = \beta_0 + \sum_{j=1}^{N} \beta_j x_j + \sum_{j=1}^{N} \beta_{jj} x_j^2 + \sum_{i \neq j}^{N} \beta_{ij} x_i x_j$$
(33)

312 Here, f(x) is the function to be optimized; β characterizes regression coefficients; and x point to a set 313 of the design parameters. However, in a constrained minimization problem, the objective function is 314 replaced by the penalized function which is as follows:

(34)

$$P(x) = f(x) + \sum_{j=1}^{N} u_j \langle g(x_j) \rangle^2 + v_k \sum_{k=1}^{K} [h_k(x)]^2$$

315 Where u_j and v_k are penalty coefficients, which are usually kept constant throughout the GA 316 simulation. The inequality constraints and equality constraints are g(x) and h(x) respectively. 317 However the fitness function, in terms of a penalized function is as follows:

$$F(x) = \frac{1}{(1+P(x))}$$
 (35)

318

If the RSA function is intensively in variation with the design parameters, this leads to redesigning the space sample between the design points and a rebuilt objective function [43]. The main target of the current 3D optimization is to enhance the blade geometry in order to minimize the losses through the passage, in terms of entropy generation; maximize the SSRT efficiency, as well as minimize both the mass flow rate of the working fluid and the rotor rotational speed. Fig. 12 explains the procedure followed during the 3D CFD optimization.

325 Both parameterization of the blade geometry and choosing the correct range of the parameters are 326 considered very important and critical steps in a successful optimization procedure. Therefore, they 327 need to be carefully selected in order to achieve the requested goal of optimization. Both; the rotor 328 and the stator blade geometries were conducted in the 3D CFD optimization. Unlike the axial rotor 329 blade (the airfoil), the rotor blade geometry is presented via a camber line and layered surface because 330 of its high curvature and the difficulty in representing the exact blade shape using two or three points. 331 As a result, besides the rotor blade number, twelve other parameters, (some of them in terms of the X 332 and Y coordinates), which together represent the rotor blade shape and required throat width, have 333 been selected as input parameters. By doing so, the full definition of the rotor blade will be figured 334 out. Regarding the stator geometry, there are only two parameters which have been chosen. The first 335 is the stagger angle which has a direct impact on the stator shape. The second parameter was chosen 336 to be a constraint objective in order to deliver the same constant mass flow rate to the rotor during the 337 optimization process.

338 In order to successfully achieve the optimum design for the SSRT, the optimization process was 339 implemented for the range of design boundary conditions shown in Table (1). The blade shape 340 parameterization is one of the fundamental aspects of a reliable optimization procedure, as it requires 341 generating wide ranges of acceptable blade geometry within groups of design variables and their 342 ranges. The parameterization of the blade's geometry was mainly accomplished for the rotor and only 343 the stator's stagger angle was included. In addition, the stator's blade number (for constrained mass 344 flow rate) was defined as a constraint function. Moreover, three different objectives were nominated 345 as objective functions. Maximizing both the output power and the turbine efficiency were two of the

three selected; and minimizing the rotor rotational speed was also desired as the third objective function in the MOGA.

348 8.2- Results of Optimization

349 After defining the parameters, the constraints and the objective functions, more than 1200 different 350 design points (proposed solutions) have been initiated depending on random distributed data. The 351 solution took more than 504 continuous hours on the Core I7 3.7 GHz processor and 48 GB RAM 352 Computer. Figs. 13 A, B and C show an example of some proposed solutions as candidate design 353 points. Due to the large number of them, the design points have been separated into three Figs. (A) to 354 (C). Each one has about 250 proposed solutions (which together represent only about 63% of all the 355 design points) in terms of the output power and the turbine efficiency. From the mentioned figures 356 one can see that some candidate points have about 92% turbine efficiency and even higher efficiency 357 than the chosen optimum point. However, because they didn't satisfy the minimum rotational speed objective, these design points have been omitted. Another important issue that can be concluded from 358 359 these figures is that not all the candidate points can deliver satisfactory results and come up with a 360 solution. It can be seen that some of them are unable to do so. Table (5) presents the optimum and the 361 base-line values of design variables of turbine, while Fig. 14 displays the two shapes of blade; for the 362 base-line and the design.

363

364 Figure 15 compares the CFD results of the base-line (CFD- Base) design against the Optimum Point 365 (OP) analysis at nominal and off design boundary conditions. The values of the output power and 366 isentropic efficiency were compared for: inlet temperature, rotational speed and pressure ratio. 367 Specifically, Figs. 15A & 15A' show the values of output power and efficiency at different values of 368 turbine inlet temperature. In this figure the effect of the turbine inlet temperature (fixing all other 369 BCs) is highlighted. While there is a positive relationship between the turbine output power and the 370 inlet temperature, the efficiency reached its maximum point at 450K, as it was the designed point. 371 Similarly, Figs. 15B & 15B' and 15C & 15C' represent the turbine output power and efficiency at 372 different values of its rotational speed and pressure ratio respectively. Furthermore, this figure shows 373 that the turbine efficiency and power output increased with a maximum improvement of about 9.35% 374 in turbine efficiency and 41.08% in power output. The main reason behind this enhancement relates to 375 the amount of losses (such as shock losses, secondary losses and incidence losses) that are largely 376 decreased when the blade shape reached its optimum configuration. Moreover, it is obvious from Fig. 377 15 that even in the off-design condition the overall performance of the optimized SSRT is by far 378 better than their peers at the BL for the all invistigated boundary conisions. Overall, in its optimum 379 design, the turbine behaves better than the CFD- Base design during the off design conditions 380 especially in terms of the amount of output power.

381

Fig. 16A demonstrates the loading distribution (represented in terms of pressure distribution) through all the passage of the rotor. The highest velocity of the flow, which is located at the throat area, results in lowest values of pressure on the suction surface. However, the opposite is the case for velocity distribution on the pressure side starting from the leading to the trailing edges. In Fig. 15B, the contours of pressure distribution through all of the turbine passage, stator and rotor, are presented. From this figure it can be seen how the pressure is distributed starting from the highest value in the stator to the lowest at the diffuser.

However, the loss through the rotor passage can be assessed by entropy generation which is an indicator of the amount of aerodynamic losses. From Fig. 17 it is shown that the 3D optimization which is done using the MOGA, was able to decrease the amount of entropy generated within mainly

the rotor blade, but also in the stator blade passage, compared to the base-line design and therebyenhance the flow aerodynamically.

9- Results of Brayton Cycle Analysis

396 Using the Brayton cycle analysis in equations 1-11 to calculate the cycle efficiency for the various 397 boundary conditions, the cycle efficiency at nominal and other boundary conditions, as well as the 398 enhancement that the cycle has achieved is clear in Fig. 18. These improvements are delivered by 3D 399 CFD results and inserted in the cycle modelling as inputs to calculate the overall cycle efficiency. This 400 improvement, which is not constant for all the investigated boundary conditions, is mainly due to the 401 increment that occured in the turbine efficiency. The pressure ratio and inlet temperature values range 402 from 2-4 and 450 K- 550 K shown in Figs. 18A, B and C respectively. At this point it is worth 403 mentioning that for the nominal conditions, the optimum pressure value that gives maximum cycle 404 efficiency is 3. However, with increasing the turbine inlet temperature the optimum pressure values 405 become 3.5 and 4 at 500 K and 550 K respectively. This is because of the fixed relationship between 406 the turbine inlet temperture and the turbine working pressure ratio. Moreover, for the other boundary 407 conditions of the cycle, with improving the turbine efficiency from about 82% to 89.5 %, the 408 improvement was about 6% at the current nominal conditions, and 5% and 4% at the three investigated 409 temperatures respectively. Finally, it can be seen that the maximum efficiency values which were 410 achieved were about 11%, 18.5% and 27% at the mentioned optimum values of the pressure ratio and 411 for the three inlet tempertures respectively.

412

413 **10- Conclusions**

414 Numerical simulation was carried out to optimize a small-scale radial turbine with output power in the 415 range of 1.5 to 7.5 kW. A one-dimensional Mean Line approach and three-dimensional CFD 416 simulations, using three-dimensional RANS with the SST turbulence model in ANSYS®15- CFX, 417 were employed to achieve the best turbine performance; and consequently, the highest reachable 418 efficiency for the small-scale solar powered Brayton cycle. This paper demonstrated the following 419 important outcomes:

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- The CFD simulation results show good agreement with mean line design results for turbine
 efficiency and power output.
- A relatively high increment in the isentopic efficiency, from 82.3% to 89.7% for the same range of output power was achieved. This was reached by optimizing the shape and the number of both the stator and the rotor components of the radial turbine.
 - The results showed that a small-scale radial turbine can achieve a power output of around 3.5 kW with isentropic efficiency about 89.5% at a relatively low inlet temperature; which can be obtained from solar energy through a parabolic concentrator dish as a heat source.
- It was clear from the 3D CFD optimization results that the MOGA optimizer is considerably beneficial in improving the turbine's performance in terms of both the turbine efficiency and the power output. Consequently, the thermal cycle efficiency has also been increased by about 6% as a result of increasing the turbine's efficiency.
- Also, the blade loading and losses (entropy generation) had significantly improved compared to the base-line design. So, it can be said that the integrated approach between Brayton cycle modelling, PD, 3D CFD simulation and optimization methodologies have potential advantages to achieve a high performance of a small-scale solar powered Brayton system.
- 437

- 438 The integrated aerodynamic and structural optimization is the next study in order to figure out the
- 439 mechanical stresses that accompanied the turbine structure during its service.

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- 443 Nomenclature

Symbol	Definition
A	Area (m ²)
b	Axial chord (mm), blade width (mm)
c	Absolute velocity (m/s)
d	Diameter (m)
f	Friction factor (-)
h	Enthalpy (J/kg)
Н	Blade height (mm)
i	Incident angle (deg.)
k	Loss Coefficient (-)
1	Length (m)
m	Mass flow rate (kg/s)
p pp	Pressure (Pa)
PK	Pressure ratio (-)
I Po	Radius (III) Paymolds No. ()
Ke	
s sc	Entropy (J/kg. K)
	Swiri coefficient (-)
I II	Peter blada valasity (m/s)
U	Rotor blade velocity (m/s) Relative velocity (m/s)
w W	Power (W)
7	Blade number in radial turbine ()
Creek symbols	Definition
	Absolute flow angle (deg.)
β	Relative flow angle (deg.)
г 8	Clearance (m)
η	Efficiency (%)
υ	Velocity ratio (-)
ρ	Density (kg/m^3)
φ	Flow coefficient (-)
Ψ	Loading coefficient (-)
ω	acentric factor (-)
ζ	Losses (-)
Acronyms	Definition
BL	Base-line Design
BCs	Boundary Conditions
CFD	Computational Fluid Dynamics
DOE	Design of Experiment
GA	Genetic Algorithm
	Leading edge
MOGA	Multi objective genetic algorithm
ORC	Organic Rankine Cycle
O.S.S.R.T	Ontimized Small Scale Radial Turbine
RANS	Reynolds-Averaged Navier-Stockes
RSA	Response Surface Approximation
SST	Shear Stress Transport
TE	Trailing Edge
Subscripts	Definition
1-6	Station
m	Meridional direction
r	radial
rel	relative
S	Isentropic
Х	Axial

t		Total, stagnation
t	8	Total to static
t	h	Thermal
e)	Tangential/circumferential direction

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References 446

- 447 [1] Ordóñez, Juan Carlos, and Adrian Bejan. "Entropy generation minimizations in parallel-plates counter flow 448 heat exchangers." International Journal of Energy Research 24.10 (2000): 843-864.
- 449 [2] Le Roux, Willem Gabriel, Tunde Bello-Ochende, and Josua P. Meyer. "Operating conditions of an open and 450 direct solar thermal Brayton cycle with optimised cavity receiver and recuperator." Energy 36.10 (2011): 6027-451 6036.
- 452 [3] Wu, Yuting, et al. "Dynamic simulation of closed Brayton cycle solar thermal power 453 system." ICSETN2004 (2004): 1-6.
- 454 [4] Le Roux, Willem Gabriel, Tunde Bello-Ochende, and Josua P. Meyer. "Optimum performance of the small-455 scale open and direct solar thermal Brayton cycle at various environmental conditions and 456 constraints." Energy46.1 (2012): 42-50..
- 457 [5] Le Roux, Willem Gabriel, Tunde Bello-Ochende, and Josua P. Meyer. "The efficiency of an open-cavity 458 tubular solar receiver for a small-scale solar thermal Brayton cycle." Energy Conversion and Management 84
- 459 (2014): 457-470.
- 460 [6] Riazi, H., and N. A. Ahmed. "Effect of the ratio of specific heats on a small scale solar Brayton 461 cycle." Procedia Engineering 49 (2012): 263-270.
- 462 [7] Chen, H., and N. C. Baines. "The aerodynamic loading of radial and mixed-flow turbines." International 463 journal of mechanical sciences 36.1 (1994): 63-79.
- 464 [8] Collings, Peter, and Zhibin Yu. "Modelling and analysis of a small-scale Organic Rankine Cycle system with 465 a scroll expander." Proceedings of the world congress on engineering. 2014.
- 466 [9] Giuffrida, Antonio. "Modelling the performance of a scroll expander for small organic Rankine cycles when 467 changing the working fluid." Applied Thermal Engineering 70.1 (2014): 1040-1049.
- 468 [10] Declaye, Sebastian, Sylvain Quoilin, and Vincent Lemort. "Design and experimental Investigation of a 469 small scale Organic Rankine Cycle using a Scroll Expander." (2010).
- [11] Erbaş, Murat, and Atilla Biyikoglu. "Design of low temperature Organic Rankine Cycle and turbine." Power Engineering, Energy and Electrical Drives (POWERENG), 2013 Fourth International 470 471 472 Conference on. IEEE, 2013.
- [12] Rahbar, Kiyarash, et al. "Modelling and optimization of organic Rankine cycle based on a small-scale 473 474 radial inflow turbine." Energy conversion and management 91 (2015): 186-198.
- 475 [13] Rahbar, Kiyarash, et al. "Parametric analysis and optimization of a small-scale radial turbine for Organic 476 Rankine Cycle." Energy 83 (2015): 696-711..
- [14] Han, Sangjo, JongBeom Seo, and Bum-Seog Choi. "Development of a 200 kW ORC radial turbine for 477 478 waste heat recovery." Journal of Mechanical Science and Technology 28.12 (2014): 5231-5241.
- 479 [15] Cho, S., C. Cho, and C. Kim. "A Study of Cycle Analysis and Turbine Design for Obtaining Small-Scaled
- 480 Power from the Organic Rankine Cycle Using R245fa." (2013).
- 481 [16] Fiaschi, Daniele, Giampaolo Manfrida, and Francesco Maraschiello. "Thermo-fluid dynamics preliminary 482 design of turbo-expanders for ORC cycles." Applied energy 97 (2012): 601-608.
- 483 [17] Harinck, John, et al. "Performance improvement of a radial organic Rankine cycle turbine by means of
- 484 automated computational fluid dynamic design."Proceedings of the Institution of Mechanical Engineers, Part A: 485 Journal of Power and Energy 227.6 (2013): 637-645.
- 486 [18] Sauret, Emilie, and Yuantong Gu. "Three-dimensional off-design numerical analysis of an organic Rankine 487 cycle radial-inflow turbine." Applied Energy135 (2014): 202-211.
- 488 [19] Rahbar K, Mahmoud S, Al-Dadah RK, Moazami N. "One-dimensional and three-dimensional numerical 489 optimization and comparison of single-stage supersonic and dual-stage transonic radial inflow turbines for the 490 ORC, Proceedings of the ASME 2016 Power and Energy Conference, Inpress accepted paper.
- 491 [20] Da Lio, L, Manente, G and Lazzaretto, A. New efficiency charts for the optimum design of axial flow 492
- turbines for organic Rankine cycles". Energy 2014; 77, pp. 447- 459.
- 493 [21] Ennil, Ali Bahr, et al. "Minimization of loss in small scale axial air turbine using CFD modeling and 494 evolutionary algorithm optimization." Applied Thermal Engineering 102 (2016): 841-848.
- 495 [22] Sauret, Emilie. "Open design of high pressure ratio radial-inflow turbine for academic validation." ASME 496 Paper No. IMECE2012-88315 (2012).

- 497 [23] Moreau, Aurélien Marsan-Stéphane. "ANALYSIS OF THE FLOW STRUCTURE IN A RADIAL
- 498 TURBINE."
- 499 [24] Odabaee, M., Mohsen Modir Shanechi, and K. Hooman. "CFD Simulation and FE Analysis of a High
- Pressure Ratio Radial Inflow Turbine." 19AFMC: 19th Australasian Fluid Mechanics Conference. Australasian
 Fluid Mechanics Society, 2014.
- [25] Tsalicoglou, Isaak, and Bent Phillipsen. "Design of radial turbine meridional profiles using particle swarm
 optimization." 2nd International Conference on Engineering Optimization. 2010.
- 504 [26] Fu, Lei, et al. "Integrated optimization design for a radial turbine wheel of a 100 kW-class 505 microturbine." Journal of Engineering for Gas Turbines and Power 134.1 (2012): 012301.
- [27] Zhang, Qiang, and Chaochen Ma. "Multiple-objective aerodynamic optimization design of a radial air
 turbine impeller." Remote Sensing, Environment and Transportation Engineering (RSETE), 2011 International
 Conference on. IEEE, 2011.
- [28] Van den Braembussche, R. A. Optimization of radial impeller geometry. VON KARMAN INST FOR
 FLUID DYNAMICS RHODE-SAINT-GENESE (BELGIUM), 2006.
- [29]. Mueller, Lasse, Zuheyr Alsalihi, and Tom Verstraete. "Multidisciplinary optimization of a turbocharger
 radial turbine." Journal of Turbomachinery135.2 (2013): 021022.
- 513 [30] Klein, SA Engineering equation solver. F-chart Software, Middleton, WI; 2013.
- [31] Balje, O. "Turbomachines. A guide to Design, Selection and Theory. JohnWiley & Sons." Inc., New
 York (1981).
- 516 [32] Rohlik H. E., "Analytical determination of radial inflow turbine design geometry for maximum efficiency,"
- 517 Tech. Rep. TN D-4384, NASA, Washington, DC, USA, (1968).
- 518 [33] Rogers C., "Mainline Performance Prediction for Radial Inflow Turbine in Small High Pressure Ratio
 519 Turbine, " VKI Lecture Series 1987-07, (1987).
- [34] Whitfield A. and Baines N., "Design of Radial Turbomachines", JohnWiley & Sons, New York, NY, USA,
 (1990).
- [35] Moustapha H., Zeleski M. F., Baines N. C., and D. Japikse, "Axial and Radial Turbines", Concepts NREC,
 White River Junction, Vt, USA, (2003).
- 524 [36] Aungier H., "Turbine Aerodynamics: Axial-Flow and Radial- Flow Turbine Design and Analysis", ASME
 525 Press, New York, NY, USA, (2006).
- 526 [37] Dixon, S.L. and Hall C., "Fluid mechanics and thermodynamics of turbomachinery". Butterworth527 Heinemann, Oxford, UK (2013).
- [38] Suhrmann, J.F., Peitsch, D., Gugau, M., Heuer, T., and Tomm, U., 2010, "Validation and development of
 loss models for small size radial turbines." Proceedings of ASME Turbo Expo 2010: Power for land, sea and
 Air GT 2010, Glasgow, UK, Paper No GT (2010)-22666.
- [39] Glassman AJ., "Computer program for design and analysis of radial inflow turbines". NASA TN 8164;
 (1976).
- 533 [40] Churchill SW. "Friction-factor equation spans all fluid-flow regimes". Chem Eng (1977);84:91–2.
- [41] Wilson, David Gordon, and Theodosios Korakianitis. The design of high-efficiency turbomachinery and
 gas turbines. MIT press, 2014.
- 536 [42] Al Jubori A, Al-Dadah RK, Mahmoud S, Khalil KM, Bahr Ennil AS. "Development of efficient small scale
- axial turbine for solar driven organic Rankine cycle". Proceedings of ASME Turbo Expo 2016: GT2016, Seoul,
 South Korea, paper no GT2016-57845.
- 539 [43] ANSYS 15 CFX-Solver Theory Guide.
- [44] Kim, Jin-Hyuk, et al. "Performance enhancement of axial fan blade through multi-objective optimization
 techniques." Journal of Mechanical Science and Technology 24.10 (2010): 2059-2066.
- 542 [45] Surekha, N., Srinivas Kolla, Deva Raj Ch, and K. Sreekanth. "Optimization of Principal Dimensions of
- Radial Flow Gas Turbine Rotor Using Genetic Algorithm." Int. J. Scientific& Engineering Research 2012;3:1-6.
 [46] McLallin, K.L.; and Haas, J.E., "Experimental Performance and Analysis of 15.04-cm-tip-diameter, Radial-
- 545 inflow Turbine with Work Factor of 1.126 and Thick Blading". NASA TP-1730, (1980).
- 546 [47] Jones, Anthony C. "Design and test of a small, high pressure ratio radial turbine." ASME 1994
 547 International Gas Turbine and Aeroengine Congress and Exposition. American Society of Mechanical
 548 Engineers, 1994.
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Fig. 13. Modulation of the SSRT efficiency and power output values with some of the investigated design points.



(A) Inlet temp, (B) Rotational speed and (C) Pressure ratio.





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Parameter	Range/value
Loading coefficient (-)	0.8-1.4
Flow coefficient (-)	0.1-0.5
Shroud _{Exit} /Shroud _{Inlet} (-)	0.8
Hub _{Exit} /Hub _{Inlet} (-)	0.22
Rotational speed (rpm)	50000-90000
Inlet total temperature (K)	450 - 550
Inlet total pressure (bar)	2 -5
Mass flow Rate (kg/sec)	0.03 - 0.05
Working fluids (-)	air
Cp(J/kg K)	1005
Inlet blade velocity (m/s)	253.1
Exit blade velocity (at shroud) (m/s)	202
Inlet relative velocity(m/s)	65.8
Exit relative velocity(m/s)	206.3
Inlet absolute velocity(m/s)	250.5
Exit absolute velocity(m/s)	14.7
Rotor inlet density(kg/m ³)	1,153
Rotor inlet Mach (abs) (-)	0.7
Rotor inlet Enthalpy (J/kg)	109463
Rotor outlet Enthalpy (J/kg)	95288.8
Rotor Enthalpy at the leading edge (J/kg)	107811
Rotor Enthalpy at the trailing edge(J/kg)	87117

809 Table 1: Input operating conditions of integrated the radial turbine and the Brayton cycle 810 model.

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Table 2: The meanline characteristics of the SSRT.

1	Nozzle	Value	Unit	Rotor	Value
	Meridional length	12.93	(mm)	Axial clearance	0.4
	Blade number	28	(-)	Blade number	15
	Tip width	1.7	(mm)	Axial length	11.671
	Exit height		(mm)	Tip width	1.7
	Inlet absolute flow angle	0	(deg.)	Inlet absolute flow angle	75
	Inlet height		(mm)	Inlet height	69.06
	Inlet radius	50.43	(mm)	Inlet radius	34.53
	Outlet absolute flow angle	75	(deg.)	Inlet relative flow angle	-9
	Outlet radius	39.71	(mm)	Outlet shroud radius	27.6
	TE thickness	0.4	(mm)	Radial clearance	0.4
	Throat area	95	(mm^2)	Throat area	95
	Aerofoil area	33.18	(mm^2)	Aerofoil area	25.38
	Camber length	17.75	(mm)	Camber length	30.975
	Cord length	15.63	(mm)	Cord length	27.82
	LE thickness	0.25	(mm)	LE thickness	0.25
	Pitch cord Ratio	0.448	(-)	Pitch cord ratio	0.365
	Solidity	2.22	(-)	Solidity	3.77
	Stagger angle	34.2	(deg.)	Stagger angle	-37.1
			(deg.)	Outlet absolute flow angle	-12.5
			(deg.)	Outlet relative flow angle (hub)	-76.3
			(deg.)	Outlet relative flow angle (root mean square)	-79.3
			(deg.)	Outlet relative flow angle(shroud)	-82.1
			(mm)	Outlet hub radius	7.6
_			(mm)	TE thickness	0.4

	(mm) Meridional length	22.193
15 Table 3: The d	lata used for validation as reported in re	eferences [46, 47].
Parameter	Reference 46	Reference 47
Working fluid (-)	Air	Air
Mass flow rate (kg)	0.3311224	0.2313
Inlet temperature (K)	1056.483	322.2
Inlet pressure (bar)	40.02673	1.379
Pressure ratio (-)	5.7	3.255
Rotational speed (rpm)	106588	31456
Output power (kW)	37.285	15.5
Turbine efficiency (%)	87	82.7

Table 4: The geometrical data used for validation	as reported in references [46, 47].
Parameter	Value
Rotor inlet diameter (m)	0.1504
Ratio of rotor exit tip to inlet diameters (-)	0.6275
Ratio of rotor exit hub to tip diameters (-)	0.4844
Datio of states wit to notes inlat diameters ()	1 11 1

Rotor inlet diameter (m)	0.1504
Ratio of rotor exit tip to inlet diameters (-)	0.6275
Ratio of rotor exit hub to tip diameters (-)	0.4844
Ratio of stator exit to rotor inlet diameters (-)	1.11 1
Ratio of stator inlet vane height to rotor inlet diameter (-)	0.0726
Ratio of stator exit bane height to rotor inlet diameter (-)	0.0537
Ratio of stator inlet to stator exit diameters (-)	1.163
Number of stator vanes (-)	29
Number of rotor blades (-)	12
Specific speed (-)	0.46
Blade speed ratio (-)	0.609

Table 5: The base-line and optimum design variable from CFD-MOGA optimization.

Design variables	Baseline	Optimum
Stator no. of blade (-)	32	36
Stator stagger angle (deg.)	33	37.739
Stator TE Beta Angle (deg.)	69	73.70
Rotor no. of blade (-)	13	15
Rotor LE beta angle (deg.)	16.2	-46.3109
Rotor Inlet Flow Angle (deg.)	-58.3	74.03
Rotor Stagger Angle at span 0.0 (deg.)	14.4	-16.8
Rotor Stagger Angle at span 0.5(deg.)	-44.7	-37.1
Rotor Stagger Angle at span 1(deg.)	-64.5	-59.3
Rotator TE beta angle (deg.)	-18.5	-9.9457
Rotor Outlet Flow Angle (deg.)	78.9	83.36
Tip width of rotor (mm)	1.35	2.1
Rotor tip clearance (mm)	0.45	0.33

823 Highlights:

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825	• 1D and 3D CFD analysis for compressed air radial turbine was carried out.
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827	• Small scale radial turbine with high efficiency.
828	
829	• 3D MOGA optimization for the radial turbine was achieved.
830	
831	• Enhancing the performance of both the turbine and the solar powered Brayton cycle.
832	• Excellent concernent between the summert CED results and two experimental works
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