A rapid, preliminary radial turbine expander design methodology for ORC waste heat recovery applications

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Abstract:

Waste Heat Recovery (WHR) technologies aim at recovering part of the otherwise wasted heat in the exhaust of a combustion engine and convert it to useful power, resulting in lower fuel consumption and pollutant emissions. Compared to other Waste Heat Recovery (WHR) technologies, Organic Rankine Cycle (ORC) systems are increasingly being regarded for field application as a technology of high potential due to their comparatively higher maturity, relative technological simplicity, lower cost and small back pressure impact on engine performance and fuel consumption. Among the ORC system components, the expander is the most crucial and expensive component in Organic Rankine Cycle (ORC) systems. The present paper present the results of the implementation of an improved Radial Turbine Design (RTD) code which expands on previously published versions and introduces improved blade design features. RTD improves on the stability and preliminary design solution response time, as well as preliminary design accuracy of turbine stage performance, compared to prior methodologies, many of which have been developed and validated for ideal gases and low expansion ratios. The RTD has been coupled with both REFPROP and COOLPROP thermodynamic property software to achieve turbine design solutions for the currently-available libraries of working fluids.

Keywords:

Blade design, Radial-inflow turbine, ORC.

1. Introduction

The significance of ORC systems in low temperature heat recovery applications and the criticality of expander efficiency on system performance, are well reported in literature [1-2]. These medium-sized ORC power systems, which are of the range of tens to hundreds of kilowatts, generally employ radial inflow type turbines for expansion. Radial inflow turbines are opted as they provide large work and flow coefficients at the lower power requirements down to 10-20kW. Compared to an axial turbine, the radial type configuration not only provides enhanced energy transfer due to the change in the radius of the flow path, it is also compact with higher density fluids [3-6].

Studies have shown that the performance of the turbine is highly dependent on the cycle parameters and on the characteristics of the working fluid. The organic fluids used in these systems are molecularly complex fluids with high gas density [7]. Moreover, as the single-stage, high-pressure radial turbines are generally operated close to the critical point of the working fluids, they will exhibit complex flow characteristics including shock wave interaction [6]. Also in small scale turbines, the losses increase due to the dominant effects of tip clearance flows and boundary layer effects. These provide additional challenges to the turbine designers as these fluids deviate from ideal gas behaviour and therefore conventional turbine design methodologies based on ideal gas assumptions cannot be implemented [3]. Therefore, the designer has to design the turbine in such a way that the turbine performs efficiently in these complex flow conditions. This calls for the development of a comprehensive turbine design methodology for ORC applications.

The preliminary design methodology which is the backbone of any turbomachinery design methodology must be capable enough to approximate these non-ideal flow behaviours. This calls for the need for efficient turbine designs that can improve the overall system performance. Although different approaches for ORC turbines have been reported in the literature, substantial advancement in small scale ORC turbine design is necessary for its design and manufacture in large volumes in view of the importance of expander efficiency to the commercial viability of ORC systems in various applications [8]. As a wide gamut of fluids are employed for ORC applications there is a need for a simple blade profiling technique that not only incorporates the complex fluid behaviour of organic fluids but should be easily adaptable for the designer based on working fluid characteristics. This paper opens avenues to the possibility of employing a simple blade profile generation technique for ORC turbines.

2. Turbine design methodology

Figure 1 describes a generalised turbomachinery design methodology using Computational Fluid Dynamics. Any turbomachinery design methodology starts with the thermodynamic analysis of the cycle. Once the cycle configuration is optimised and the type of turbomachinery is selected, the preliminary design is performed. This design determines the principal component dimension and the thermodynamic properties at the main thermodynamic stations. This is followed by the meanline analysis, where the performance of the machine is analysed based on the preliminary geometry obtained. This includes estimation of the overall performance of the machine and also the various losses using empirical correlations. This is followed by the calculation of the 2D and 3D blade profile which can then be subjected to computational analysis through which the complex flow characteristics are interpreted. Once the detailed geometry is obtained, the turbine can be subjected to structural and experimental analysis [9-11].

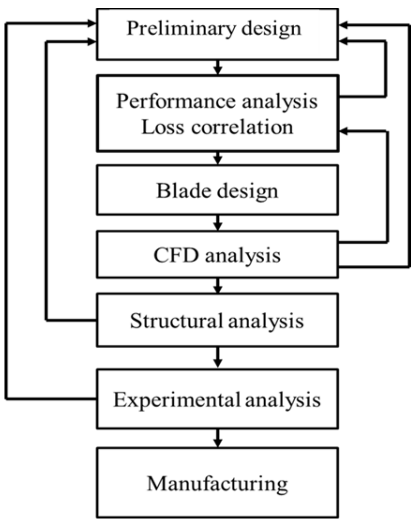


Figure 1. A generalised turbomachinery design methodology

The present paper extends the core concepts of the Radial Turbine Design (RTD) code developed at Brunel University London [12–14] for preliminary radial turbine expander design [12-13], optimisation [14] which were eventually experimentally validated on a dedicated ORC-equipped Diesel engine test rig at Brunel [15].

The code is extended in this work by re-visiting Hasselgruber's blade design technique [16] proposed by Balje [17], which provides the blade shape of the turbine rotor blades. Detailed profile of the blade passage including, the coordinates of the hub and shroud profile, channel width and depth thickness are obtained. Balje [17] has employed this technique for the design and flow path analysis of centrifugal compressors. Ghosh [18] has employed the same for the design of cryogenic turbines for air refrigeration and liquefaction plants. Chakravarty et al. [19] made use of Hasselgruber's technique for the design of cryogenic helium turbines and found to provide efficient performance on experimental analysis.

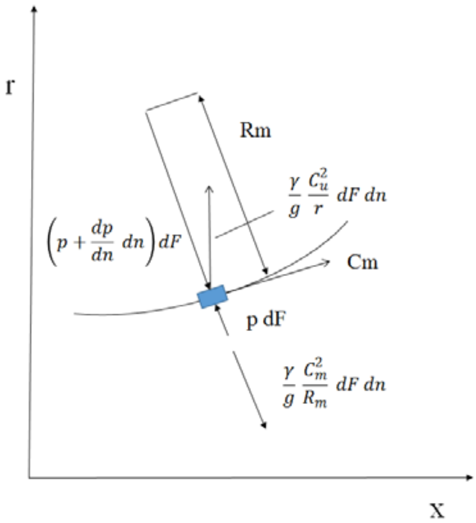


Figure 2. Radial equilibrium conditions of a fluid element [20]

The blade profile is obtained by solving the equations of motion for a fluid element in equilibrium in the meridional plane (Fig. 2). Figure 2 shows the equilibrium condition of the flow particle in the meridional plane. $\frac{\gamma}{g}\frac{C_u^2}{r}\,dF\,dn$ is the radial outward component of the centrifugal force and $\frac{\gamma}{g}\frac{C_m^2}{R_m}\,dF\,dn$ is the force opposing it. Considering equilibrium in the direction perpendicular to the flow, the pressure gradient ${dp}/{dn}$ can be written as,

$$\frac{dp}{dn} = \frac{\gamma}{g}\left(\frac{C_u^2}{r}\cos\delta - \frac{C_m^2}{R_m}\right) \tag{1}$$

Hasselgruber [16] has assumed this pressure gradient to be minimum, ${dp}/{dn} = 0$ so as to minimize the secondary flows. Equation (1) thus becomes,

$$\frac{C_m^2}{R_m} = \frac{C_u^2}{r}\cos\delta \tag{2}$$

This leads to an interrelation between the meridional velocity C_m, radius of curvature R_m, peripheral velocity C_u, radius r and inclination angle δ. Similarly solving for the equilibrium conditions in the other directions and using some transformations these equations can be represented in terms of velocity components from which the rotor profile can be designed [16, 20].

For the solution of the above equations Balje [20] has used certain conditions that dictate the flow behaviour within the passage. Balje has proposed the acceleration within the flow passage to follow the expression,

$$w^2 = w_1^2 + (w_2^2 - w_1^2)(\frac{l}{l_2})^{K_e} \tag{3}$$

Where l_2 denote the streamline length and K_e is a free parameter that controls the flow acceleration. Similarly the variation of the flow angle relative to the stream wise coordinate is given by

$$\frac{dl}{ds} = \frac{1}{\sin\beta_2^*} + C(1 - \frac{s}{s_2})^{K_h} \tag{4}$$

where s denotes the stream wise coordinate, β the flow angle and K_h, the second free parameter, that controls the change in blade angle. The value of K_h can vary between 1 and 20 and the value of K_e between 0 and 2. In essence, K_e and K_h control the curvature of the flow path and in turn dictate the boundary layer growth and separation within the flow passage. Hence, through proper selection of K_e and K_h values, the optimum turbine geometry with minimum losses can be obtained. The significant advantage of this method is that the blade profile can be uniquely generated by defining the free parameters. In this paper the authors try to incorporate the Hasselgruber's blade generation technique for the generation of 3D blade profile for ORC turbines.

4. The Application

The development of the turbine in this work is for a 20 kW ORC system for a stationary diesel engine. The input conditions for the turbine are given in Table 1. The best working fluid and the optimum properties at the station points were obtained based on the thermodynamic analysis of the ORC cycle. Novec649 an engineered fluid is chosen as the working fluid. Novec649 exhibits near zero GWP (global warming potential) and is non-flammable. The properties of Novec649 are described in Table 2.

Table 1. Design input parameters

Parameter	Values
Stagnation temperature at the turbine inlet	454.34 K
Stagnation pressure at the turbine inlet	418.15 K
Exit static pressure	105.21 k Pa
Pressure ratio	16
Mass flow rate	1.32 kg/s
Rotational speed	24330.43 rpm
Turbine power output	26.75 kW
Turbine efficiency (assumed)	75%

Table 2. Properties of the working fluid, Novec649

Fluid	T_{cr} (K)	P_{cr} (K)	Boiling point (K)	Molecular mass (g/kmol)	GWP	ODP

Novec649	441.81	18.69	322.2	316.04	1	0

Once the thermodynamic properties at the station points (nozzle inlet, turbine inlet and exit) were determined, the major geometrical parameters of the turbines are obtained. Figure 3 shows the meridional view of the turbine flow passage and the major geometrical parameters of the ORC turbine are described in Table 3.

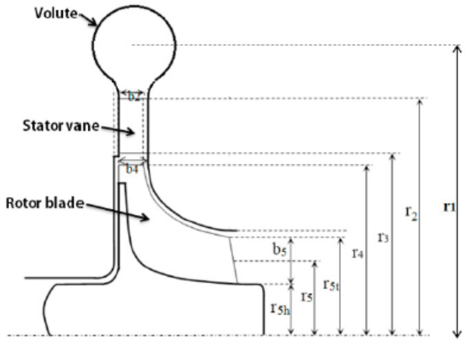


Figure 3. Meridional view of turbine flow passage

Table 3. Major geometrical parameters of the turbine

Number of blades	14
Volute inlet diameter	0.154 m
Diameter at stator inlet	0.137 m
Diameter at stator exit	0.112 m
Diameter at the turbine inlet	0.109 m
Mean diameter at the turbine exit	0.051 m
Tip diameter at turbine inlet	0.075 m
Hub diameter at turbine exit	0.026 m
Blade height at turbine blade inlet	0.003 m
Blade height at turbine blade exit	0.024 m

Based on the obtained geometrical parameters, the meridional and 3D blade profile needs to be generated. The Hasselgruber's technique can be employed at this point. The turbine design methodology incorporating Hasselgruber's technique is shown in Fig. 4.

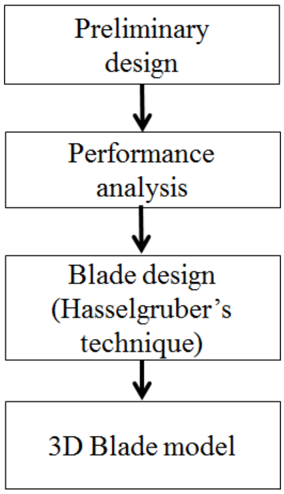


Figure 4. Turbine design methodology employing Hasselgruber's technique

5. Results and discussion

The turbine blade profile was generated based on the major geometrical parameters in Table 3. The radial turbine design code was written in Engineering Equation Solver (EES). As ORC fluids obey real gas behaviour, real gas property values were used for the calculations. In this analysis the value of K_h was varied from 4 to 16 and that of K_e from 0.25 to 1.25.

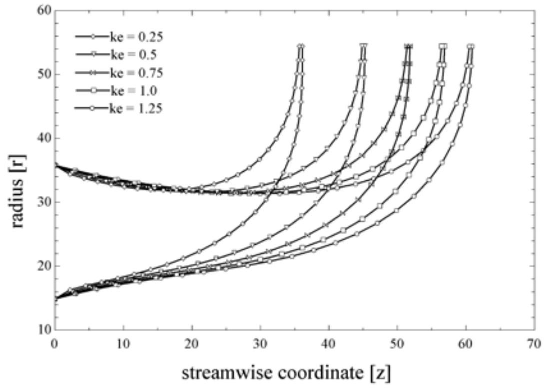


Figure 5a. Meridional profiles for K_h = 4

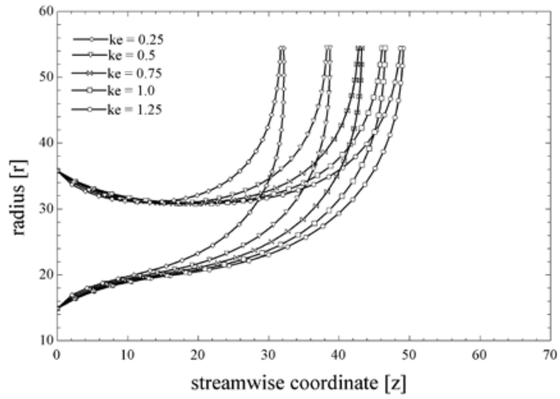


Figure 5b. Meridional profiles for K_h = 8

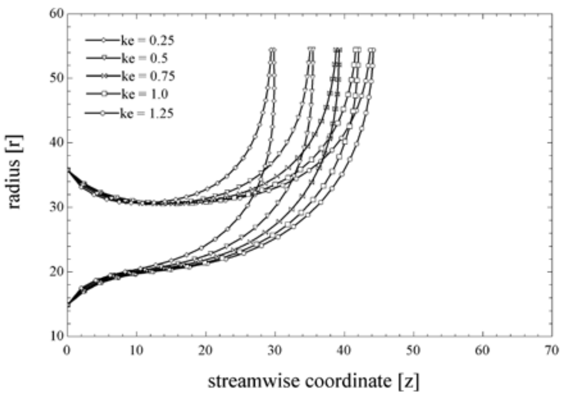


Figure 5c. Meridional profiles for K_h = 12

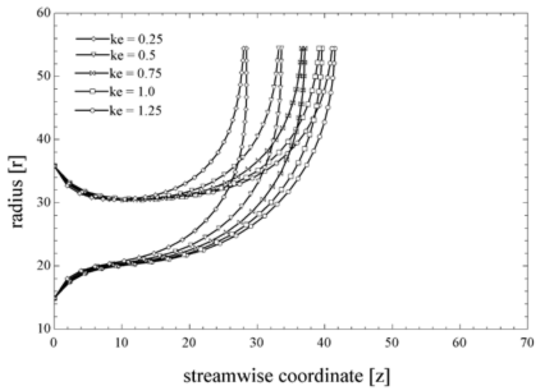
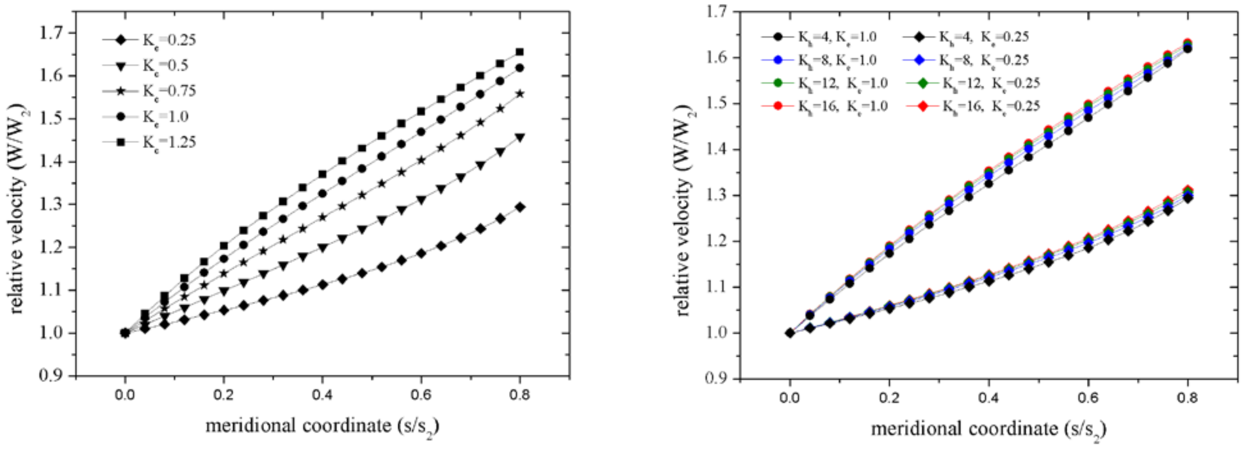


Figure 5d. Meridional profiles for K_h = 16

Figure 5 shows the meridional profile of the hub and the shroud of the radial turbine for different K_h and K_e values. It can be seen that, the axial length to diameter ratio of the turbine wheel is decreased as K_h is increased from 4 to 16 and when K_e is decreased from 1.25 to 0.25. In figure 5a for K_h = 4, with decrease in K_e from 1.25 to 0.25 the axial length to diameter ratio decreases from 0.56 to 0.33. A similar decrease in K_e for K_h=16 (Fig. 5d), reduces the axial length to diameter ratio from 0.38 to 0.26. These variations in the axial length are due to the variation in the radius of curvature with change in K_e and K_h values. With increase in K_h or decrease in K_e, the radius of curvature (R_m) decreases with a corresponding reduction in axial length. A closer observation of Fig. 5a – Fig.5c shows that for higher values of K_h, the trailing edge part (z = 0) curves to form a trumpet shaped cross-section, which may not be suitable for small scale turbines considering manufacturing constraints.



(a) K_h = 4 *(b) K_e = 0.25 and 1.0*

Figure 6. Relative velocity in the turbine flow path

Figure 6 shows the relative velocity distribution within the turbine flow passage. The relative velocity increases with increase in the acceleration free parameter, K_e (Fig. 6a). It may be observed that the rise is relative velocity becomes more pronounced and rapid as K_e is increased from 0.25 to 1.25. Figure 6b, compares the relative velocity ratios for different values of K_h for K_e = 0.25 and 1.0. It may be noted that for constant $K_{e,}$ the velocity ratio increases with increase in K_h, which can be attributed to the reduction in the turbine axial length with increase in $K_{h.}$

The change in the radius of curvature is reflected in the variation of relative flow angle (β) within the turbine flow passage (Fig. 7). It can be seen that for smaller values of K_h the radial (β =90°) to axial bend starts nearer the leading edge whereas for larger values of $K_{h,}$ the point of deviation from radial to axial is located further downstream, with the smaller the radius of curvature resulting in greater flow velocity in that region. As K_h is increased, the radius of curvature is decreased with a reduction in the axial length to diameter ratio.

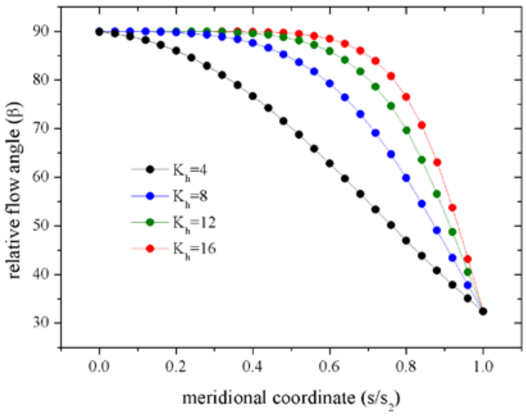
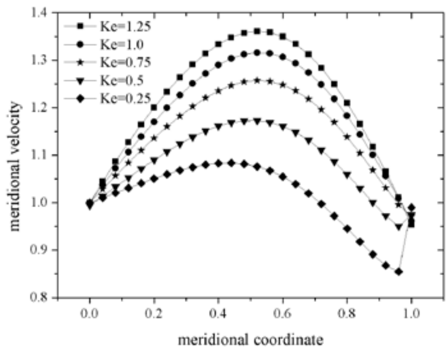
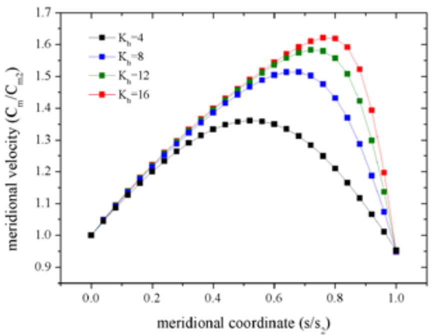


Figure 7. Flow angle distribution within the turbine flow path



(a) $K_h = 4$



(b) $K_e = 1.25$

Figure 8. Meridional velocity in the turbine flow path

The variation of meridional velocity along the turbine flow passage is shown in Fig. 8. The meridional velocity ratio first increases and then decreases along the flow passage. It may be observed that for a constant value of K_h, the maximum value for meridional velocity ratio increases with increase in K_e. At smaller values of K_e a sudden increase in the meridional velocity towards the trailing edge can be observed, which results in a dip in the shroud contour which in turn also leads to a trumpet shaped cross-section. Figure 8b shows the variation in meridional velocity ratio for K_e=1.25 for different values of K_h. The maximum meridional velocity ratio increases with increase in K_h. This is due to the smaller radius of curvature for higher values of K_h. The higher the velocity gradient, the greater the probability of the formation of secondary flow structures within the turbine flow path. As the secondary flow structures result in performance loss, the blade profile should be designed such that the velocity gradient is minimised. The analysis of the meridional velocity ratio distribution within the flow passage provides a measure of the actual three-dimensional flow effects within the turbine. For a detailed analysis of these three dimensional flow characteristics, this preliminary analysis is then coupled to a comprehensive CFD campaign, which is the subject of on-going work.

5. Conclusion

In this paper, attempt has been made to improve the existing turbine design methodology for ORC turbines. A combination of two techniques, primarily based on Hasselgruber's blade generation method, have been implemented; the method is a proven blade profiling technique for radial turbine rotors and it has been incorporated into the RTD radial turbine code for generating the blade profiles for ORC turbines. The preliminary results obtained in this study are encouraging and provide scope for in-depth parametric analysis of the turbine flow field. For efficient performance of the turbine, optimum turbine geometry with minimum loss generation is required to be achieved, through suitable combinations of K_e and K_h. Once, the effect of free parameters K_e and K_h on the flow characteristics of ORC turbines is established, designers can obtain efficient turbine profiles for further detailed design and optimisation work. RTD features a quick turn-around of these profiles, as detailed in prior work of this group.

References

[1] Karvountzis-Kontakiotis, A., Pesiridis, A., Zhao, H., Alshammari, F. et al., "Effect of an ORC Waste Heat Recovery System on Diesel Engine Fuel Economy for Off-Highway Vehicles," SAE Technical Paper 2017-01-0136, 2017

[2] Alshammari, F., Karvountzis-Kontakiotis, A., Pesyridis, A., & Alshammari, M. Expander Technologies for Automotive Engine Organic Rankine Cycle Applications. Energies, 11(7), 2018.

[3] Anand N., Vitale S., Pini, M., Otero, G. J., Pecnik, R., Design methodology for supersonic radial vanes operating in non-ideal flow conditions. ASME Journal of Engineering for Gas Turbines and Power 2019:141:022601-1 – 022601-9.

[4] Bo Z., Sang Z., Lv X., Weng Y., Effects of different working fluids on the performance of a radial turbine in an organic Rankine cycle power system. Journal of Mechanical Science and Technology 2018:32(9):4503-4515.

[5] White M. T., Markides C. N., Sayma A. I., Working fluid replacement in supersonic organic Rankine cycle turbines. ASME Journal of Engineering for Gas Turbines and Power 2018:140: 091703-1 – 091703-10.

[6] White M. T., Markides C. N., Sayma A., Supersonic flow of non-ideal fluids in nozzles: An application of similitude theory and lessons for ORC turbine design and flexible use considering system performance, In: 1st International seminar on non-ideal compressible fluid dynamics for propulsion and power, 2017 IOP Conference series: Journal of Physics: Conf. series 821:012002.

[7] Wheeler A. P. S., Ong J., The role of dense gas dynamics on organic Rankine cycle turbine performance. ASME Journal of Engineering for Gas Turbines and Power 2013:135:102603-1 – 102603-9.

[8] Mounier V., Olmedo L. E., Schiffman, J., Small scale radial inflow turbine performance and pre-design maps for organic Rankine cycles. Energy 2018:143:1072-1084.

[9] Moustapha H., Zelesky M. F., Baines N. C., Japikse D., Axial and radial Turbines. Concepts NREC; 2003.

[10] Casey M. V., Computational methods for preliminary design and geometry definition in turbomachinery. In Turbomachinery design using CFD. AGARD lecture series; 1994.

[11] Jung H., Krumdieck S., Meanline design of a 250 kw radial inflow turbine stage using r245fa working fluid and waste heat from a refinery process. Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy, 2016:230(4):402–414.

[12] Alshammari, F., Karvountzis-Kontakiotis, A., Pesiridis, A., & Minton, T. Radial Expander Design for an Engine Organic Rankine Cycle Waste Heat Recovery System. Energy Procedia, 129, 285-292, 2017.

[13] Alshammari, F., Karvountzis-Kontakiotis, A., Pesiridis, A., & Giannakakis, P. Off-design Performance Prediction of Radial Turbines Operating with Ideal and Real Working Fluids. Energy Conversion and Management, 171, 1430-1439, 2018.

[14] Alshammari, F., Karvountzis-Kontakiotis, A., & Pesyridis, A. Effect of radial turbo-expander design on off-highway vehicle organic Rankine cycle system efficiency. International Journal of Powertrains, 7(Nos. 1/2/3), 72-93, 2018.

[15] Alshammari, F., Pesyridis, A., Karvountzis-Kontakiotis, A., Franchetti, B., & Pesmazoglou, I. (2018). Experimental Study of a Small Scale Organic Rankine Cycle Waste Heat Recovery System for a Heavy Duty Diesel Engine with Focus on the Radial Inflow Turbine Expander Performance. Applied Energy, Volume 215, 1 April 2018, Pages 543-555.

[16] Hasselgruber, H., Stromungsgerechte gestaltung der laufrader von radialcompressoren mit axialem laufradeintrict Konstruction. 1958:10(1). (in German).

[17] Balje O. E., Turbomachines, John Wiley and Sons; 1981.

[18] Ghosh P., Analytical and Experimental Studies on Cryogenic Turboexpander. Ph.D dissertation, IIT Kharagpur, India; 2002.

[19] Chakravarty A., Singh T., High Speed Miniature Cryogenic Turboexpander Impellers at BARC. Indian Journal of Cryogenics 2011:36:1-9.

[20] Balje O. E., Loss and flow path studies on centrifugal compressors – Part II. Journal of Engineering for Power, ASME. 1970:287-300.