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(coordinadores)

# Proceedings of the 7th International Seminar on ORC Power Systems

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# ORC

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# Robust and Fast Meanline Methodology for Radial Inflow Turbines with Non-ideal Gases for System Level Optimisation of Organic Rankine Cycles

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## ABSTRACT

Waste heat recovery aims to increase the overall system efficiency and thus, reduce emissions. One technology commonly considered is the Organic Rankine Cycle (ORC) which uses a refrigerant as the working fluid. For an efficient operation of the ORC, the different components, e.g., the turbine, need to be optimised considering the entire system. This paper investigates the capability of meanline modelling for nozzled turbines to model the non-ideal gas effects of different refrigerants. For optimal implementation at a system level, it is important to have sufficient accuracy and low computational cost. The meanline model was compared with CFD for three different working fluids. A 3D parametric model was generated, and a numerical experiment was performed using Latin Hypercube Sampling. Seven meanline design parameters were investigated over a broad design space to show the influence of each respective one on turbine performance. Subsequently, the meanline model was calibrated against a subset of geometries and compared to the entire data sample. The average error stayed below 5%, which is sufficiently accurate for preliminary-stage calculations.



## 1 INTRODUCTION

The development of new, zero-emission propulsion systems for heavy-duty vehicles is necessary to meet the emission restrictions to NetZero by 2050 imposed by the governments. Additionally, it is also required to incorporate existing or in-development technological solutions capable of meeting the new limited emissions target during the decarbonization journey. The Organic Rankine Cycle (ORC) System is an innovative technology poised to be attractive for mobility/heavy-duty applications, where other, less polluting powertrains than conventional ICE systems are not able to meet the torque demand under certain operating conditions. Leveraging the principles of the Rankine Cycle, the ORC system enables the efficient conversion of waste heat into usable energy, making it a viable solution for enhancing the emissions for the transitions of these machines to a NetZero scenario.

The ORC system utilizes organic fluids with low boiling points to extract heat energy from exhaust gases, engine coolant, or other waste heat sources. This heat is then converted into mechanical energy, which can be used to power auxiliary systems, reduce reliance on conventional fuels, and improve overall energy efficiency. Thereby, the recovery of heat from exhaust gases of internal combustion engines (ICE) could mean a significant option for reducing emissions and fuel consumption in the heavy-duty sector. This approach has been widely investigated in literature (Maizza & Maizza, 2001; Teng et al., 2007) and different cycle configurations have been proposed (Nemati et al., 2017; Yu et al., 2016). Among the options studied, the Rankine cycle operated with organic fluids (Organic Rankine Cycle – ORC) is a promising solution (Sun et al., 2017) for two reasons:

i. It provides good thermal efficiency over a wide range of operating temperatures, suitable for integration with engines used in heavy vehicles in the transport sector.

ii. The organic fluids used have low global warming and ozone depletion potentials (Desideri et al., 2016; Eyerer et al., 2016; C. M. Invernizzi et al., 2016).

The implementation of waste heat recovery via ORC-based power units for heavy-duty solutions has been studied under different configurations on conventional/unconventional power train systems. Some of these solutions using the ORC system for the recovery of residual thermal energy to the exhaust allow the thermal efficiency of the power production unit to be increased to values of more than 40% (C. Invernizzi et al., 2007; Noriega et al., 2020).

To increase system efficiency, the turbine must be appropriately designed, based on the application under study and potential constraints. The initial turbine design is commonly delivered using low-order models (0D-1D), following the meanline modelling approach. This approach assumes that the flow behaviour through the turbine can be represented by the flow through all stations of the turbine from the volute inlet to the rotor exit along the "meanline" path, being this the streamline at mid-span. Meanline modelling is computationally cost-effective, thus allowing the assessment of a broad range of design possibilities, however, it usually requires calibration against experimental data on similar designs and operating conditions. Baines (Whitfield & Baines, 1990) developed a meanline model commonly used in the most recent research on radial turbines. Originally formulated using the ideal gas equation with air as a working fluid, this model has been adapted by other authors to implement the equation of state for the real gas behaviour of refrigerants normally present in an ORC system (Rahbar et al., 2015). One of the key aspects when using the meanline modelling approach is the calibration of the loss coefficients used to model losses occurring in each station. There is limited data available for radial turbines operating with organic fluids in literature (Da Lio et al., 2017; Desideri et al., 2016; Eyerer et al., 2016).

To address this lack of available data and provide a universal tool in the preliminary stage of a radial turbine in an ORC application, this work aims to calibrate the meanline model for a radial turbine using CFD performance data from a range of geometries by applying different refrigerants, such as R1233zd(E), R1234ze(Z) and Benzene. In the first step, a sensitivity analysis is conducted for the different meanline design parameters, and this analysis gives insights into which key parameters should be investigated in the preliminary design stages. Subsequently, the data is used to calibrate a meanline model to capture the overall performance of the turbine under different organic fluids. A comparison is made between CFD and meanline modelling to verify the applicability of the design methodology for a range of fluids. Thus, a robust methodology is developed which can be embedded into the system-level assessment of ORC applications without the need for time-consuming CFD simulations. 92

## 2 1D MODELLING APPROACH

The meanline approach used in this assessment was developed in Python and adapted from the work conducted by (Hohenberg et al., 2021). The model was further extended during the current study focusing on two main areas of development: firstly, the model was extended to consider an additional turbine station – the nozzle. Secondly, the model was adapted to consider the properties of different refrigerants, and non-ideal gas effects were included in the model by *look-up* tables instead of ideal gas equations (Bell et al., 2013).

Meanline modelling describes the flow through a turbine based on 1D equations for continuity, momentum and energy solved at mean height between two stations. To account for losses, semi-empirical enthalpy loss models are introduced based on physical processes in the turbine. Losses in the stator part are accounted for by assuming a pressure loss and swirl loss coefficient (Japikse & Baines, 1997). The main losses in the rotor are as follows: passage loss due to friction on the wall (Wasserbauer & Glassman, 1975), incidence loss due to flow incidence at the rotor leading edge (Futral & Wasserbauer, 1965) and tip clearance loss due to secondary flows at the rotor tip clearance (Spraker, 1987). A blockage factor is included in all stations to account for the blockage effects in the mass flow produced by the geometry. All these loss models need to be calibrated against experimental/ computational data to allow for an accurate prediction of the turbine performance. Figure 1 summarizes the meanline approach, which allows for prediction of total-to-static isentropic efficiency $\eta_{t-s}$ and mass flow $\dot{m}$ based on a given pressure ratio, $PR$, and rotational speed, $N$. Additionally, the implemented loss models are given for which the loss coefficients need to be calibrated against experimental or numerical data.

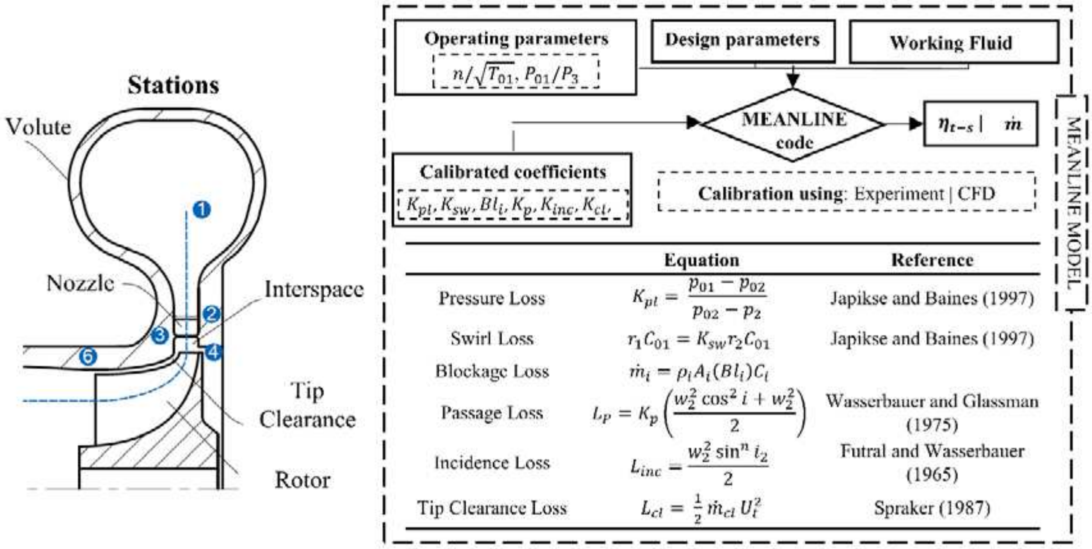


**Figure 1.** Meridional section of a radial turbine showing the meanline stage (left) and meanline modelling approach (right)

## 3 WORKING FLUID SELECTION



For the selection of working fluids to be investigated, a simple ORC configuration consisting of a pump, evaporator, turbine and condenser (Figure 2) was modelled by basic thermodynamic equations. No pressure drop was considered through the piping connecting the elements nor through the heat exchangers, and constant isentropic efficiency of 75% was assumed across the turbomachinery elements.

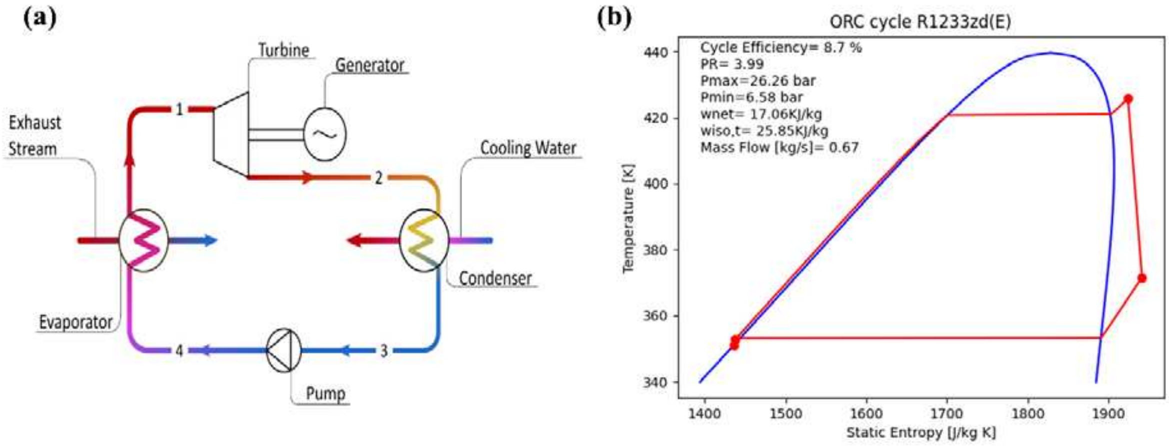


**Figure 2.** ORC diagram (a) and resulting h-s Diagram for R1233zd(E) (b)

The application consisted of exhaust gases from a heavy-duty engine which takes the ORC fluid to a maximum temperature of 430 K, while the minimum temperature was set to 350 K, based on the cooling limitations of the application reported in the literature (Lang et al., 2013; Robertson, 2019). The evaporating pressure were calculated as the saturation pressure at the maximum temperature minus a superheating of 5 K if the temperature was below the critical temperature. If the maximum temperature of the cycle was above the critical temperature, the evaporating pressure was taken as the saturation pressure at the critical temperature minus 2 K. The condensing pressure was set as the saturation pressure at the minimum temperature of the cycle plus 2 K subcooling. This way, using different working fluids leads to varying pressure ratios across the turbine. The efficiency of the ORC cycle was evaluated as the ratio between the net-specific work and the evaporating heat. An example of the h-s diagram and cycle properties for, R1233zd(E) is shown in Figure 2 (b). After a preliminary analysis using different fluids, R1233zd(E), R1234ze(Z) and Benzene showed the highest cycle efficiency and were selected to investigate the turbine design. A summary of the ORC efficiency using different working fluids is shown in Table 1:

**Table 1.** ORC efficiency using different working fluids.

| **Fluid** | **R245fa** | **R1233zd(E)** | **R1234ze(Z)** | **R1234ze(E)** | **Benzene** | **D4** | **D5** |
|---|---|---|---|---|---|---|---|
| $\boldsymbol{\eta_{ORC}}$ **[%]** | 8.19 | 8.7 | 8.23 | 4.11 | 10.3 | 7.9 | 7.7 |

R1233zd(E) and R1234ze(Z) are 3rd generation refrigerants, substitutes of the widely used R245fa, with very similar thermophysical properties, but zero global warming potential, GWP, and ozone depletion potential, ODP (Eyerer et al., 2019; Yang et al., 2019). Benzene is an aromatic hydrocarbon with a higher critical temperature and pressure than the other two and a low environmental footprint too, but higher levels of toxicity and flammability. Table 2 summarizes the boundary conditions of the turbine depending on the fluid subsequently used to calculate the aerodynamic performance of the different turbine designs.

**Table 2.** Turbine boundary conditions based on the ORC system for the fluids of study.

| **Fluid** | **R1233zd(E)** | **R1234ze(Z)** | **Benzene** |
|---|---|---|---|
| $\boldsymbol{PR}$ **[-]** | 4 | 3.9 | 5.5 |
| $\boldsymbol{P_{evap}}$**[bar]** | 26.3 | 33.7 | 5.6 |
| $\boldsymbol{P_{cond}}$**[bar]** | 6.6 | 8.6 | 1.0 |



## 4 NUMERICAL SETUP

A 3D model developed in the parametric CAE platform CAESES (FRIENSHIP SYSTEMS, n.d.) was used to analyse the influence of geometric parameters on the turbine performance for the three different working fluids. For this purpose, Latin Hypercube Sampling (LHS) was conducted and the CFD data generated was used both for the sensitivity analysis of key design parameters as well as verification for the meanline model.

The parametric model consisting of both a nozzle and a rotor was initially introduced by (Alvarez-Regueiro et al., 2022). Both vanes and blades are defined through the parametrisation of the meridional profile as well as camberline and blade surface thickness distribution. In Figure 3(a) a representation of the rotor profile is given along with the defining parameters. The rotor is radial-fibred and thus, one definition of the camber line is sufficient to describe the full blade. The nozzle meridional profile is defined by the rotor blade height as well as the inlet and the outlet radii which are a function of the rotor inlet diameter. The geometry has a constant number of vanes (13) and blades (11). Finally, the geometry is defined and flow domains (cf. Figure 3 (a), right) can be exported for CFD simulations.

Simulations were conducted ANSYS CFX with mesh generation using Turbo-Grid. The numerical setup shown in Figure 3 (b), is a single-passage RANS model with SST turbulence model. To calculate the thermophysical properties of each working fluid, Peng-Robinson equation of state was used. Mesh studies were conducted for all three fluids, with the maximum average $y_{+,ave,Nozzle} = 24$ at the nozzle and $y_{+,ave,Rotor} = 17$ at the rotor for R1234ze(Z). As can be seen in Figure 3(b), turbine efficiency is insensitive to the applied mesh size and based on this study a mesh was selected containing of 2.2 million elements, resulting in a computational time of less than 24 core hours per simulation.

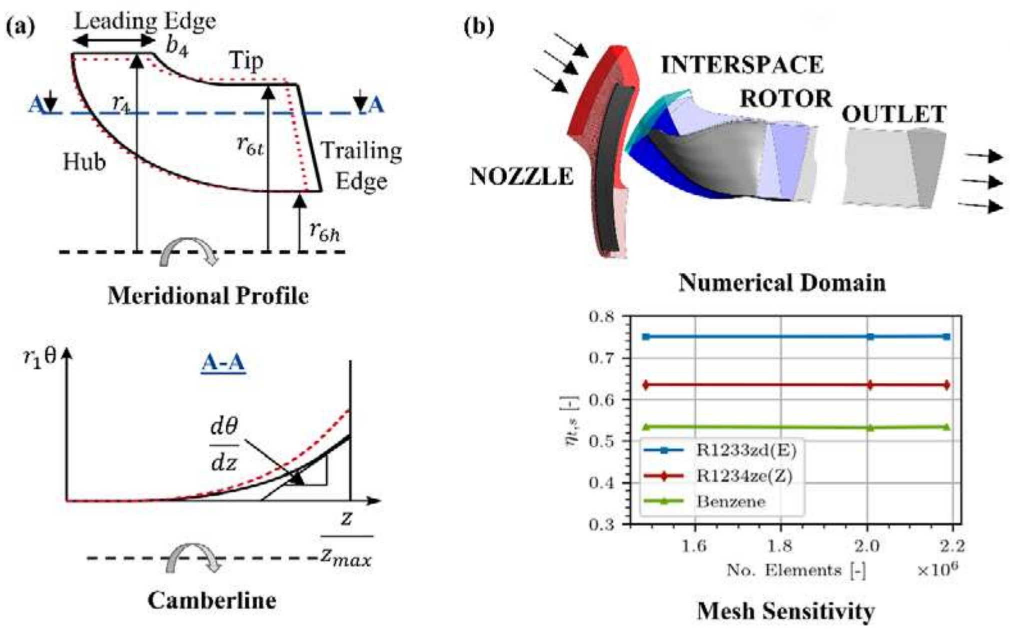


**Figure 3.** Meridional blade and camberline parametrisation of the rotor (a) and numerical setup (b)

A frozen rotor model was applied as it proved to be more robust than the stage interface: for a sample of 100 designs, 93% converged using frozen rotor compared to 56% using stage interface. The mean absolute deviation between both models was 0.44% in relative deviation for mass flow and 1% in percentage points for efficiency.

## 5 RESULTS



### 5.1 Design of Experiment

Six geometrical meanline parameters as well as the rotational speed, $N$, were selected given their influence on the turbine performance. An overview of those parameters is given in Table 3 with both baseline value and lower/ upper boundaries of the sample. All over parameters are kept constant during all simulations. Latin Hypercube Sampling (LHS) was performed with 500 samples for each working fluid with a convergence rate of approximately 80%. For each working fluid, the design vectors were kept the same, thus, only the fluid itself and respective boundary conditions were changed.

**Table 3.** Design parameters: baseline value and design space for sampling

| Design Parameter | Baseline Value | Design Space |
|---|---|---|
| $r_4\ [mm]$ | 24 | [22.0 – 32.0] |
| $r_{6s}/r_4\ [-]$ | 0.82 | [0.78 – 0.85] |
| $r_{6h}/r_4\ [-]$ | 0.24 | [0.18 – 0.27] |
| $\alpha_4\ [deg]$ | 77 | [73.0 – 80.0] |
| $b_4\ [mm]$ | 5 | [2.00 – 8.00] |
| $\beta_6 [deg]$ | -63 | [-72 – -55] |
| $N\ [krpm]$ | 50 | [30.0 – 90.0] |

To assess the results of the sensitivity analysis, Figure 4 (left) presents the resulting mass flow, $\dot{m}$, against rotational speed. The mass flow is normalized by the respective mean value $\dot{m}_{mean}$ of the dataset, with values of $2.07\frac{kg}{s}$ (R1233zd($E$)), $2.32\frac{kg}{s}$ (R1234ze(Z)) and $0.33\frac{kg}{s}$ (Benzene). Next to the data points, a trend line (curve fitting) is implemented which is also presented for the remaining design parameters in Figure 4 (right). It can be noted that the overall mass flow for R1233zd(E) and R1234ze(Z) are at similar levels. Simulations conducted with Benzene will result in lower mean mass flow explained by the overall lower inlet pressure and density. Furthermore, it can be noticed that all the exit angle $\beta_6$ and the radius ratios $r_{6s}/r_4$ and $r_{6h}/r_4$ have negligible effect on the resulting $MFP$, whereas there is a nearly linear positive correlation with both the rotor inlet diameter and blade height. Additionally, the nozzle inlet angle $\alpha_2$ and the mass flow share a negative relationship.

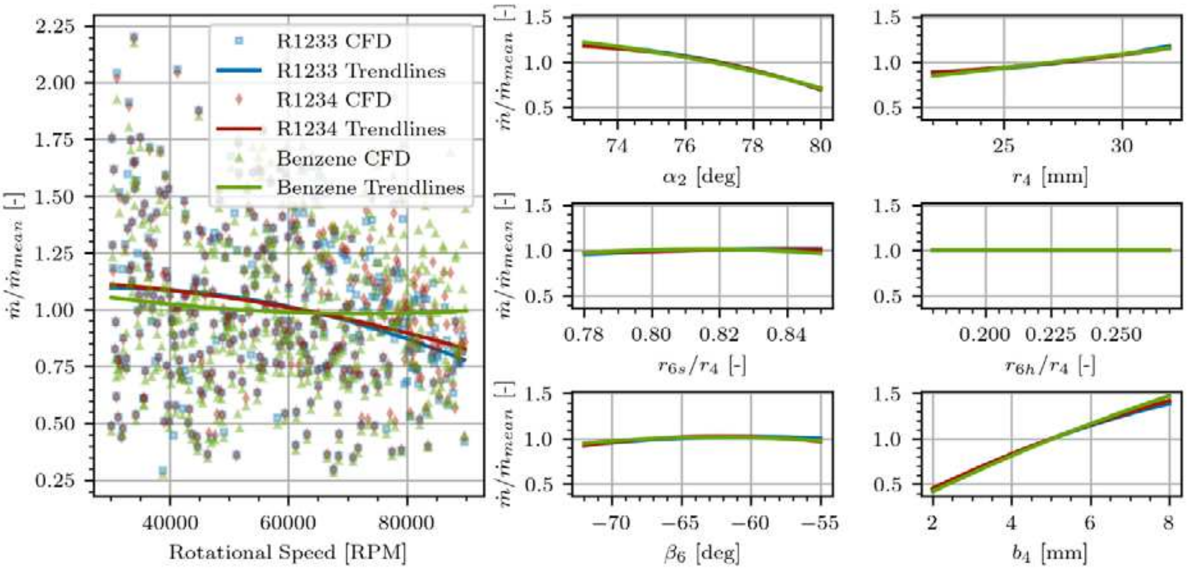


**Figure 4**. Mass flow sensitivity of each design parameter for the three different fluids

All of the above can be explained by the mass flow equation Eq. ( 1 ) at the nozzle inlet. The key driver of mass flow difference between the three working fluids is the density level of the fluid while alteration of geometrical parameters results in modification of effective area $A_{ref}$.

$$\dot{m} = A_{ref} * C_m * \rho = 2\pi r_2 * b_2 * C_2 * \cos(\alpha_2) * \rho_2 \quad (1)$$

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The results of the sensitivity analysis for the second performance parameter of interest, the isentropic efficiency $\eta_{t-s}$, is presented in Figure 5. Again, R1233zd(E) and R1234ze(Z) show very similar trends, whereby R1234ze(Z) seem to result in slightly higher performance than R1233zd(E) when applied on the same turbine. The peak efficiency of the turbine operating with Benzene is also slightly lower than for both refrigerants.

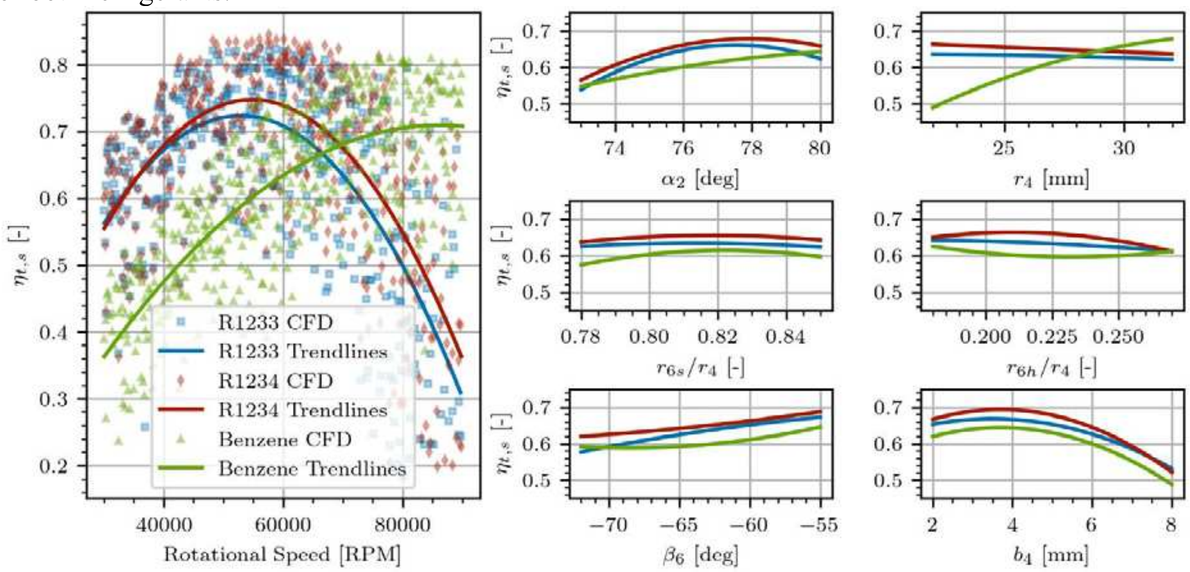



**Figure 5**. Efficiency sensitivity of each design parameter for the three different fluids

Interestingly, all geometrical parameters show similar influences on the performance of the respective turbine, apart from the rotor inlet radius $r_4$ and rotational speed $N$. It is obvious, that while the optimum efficiency for R1233zd(E) and R1234ze(Z) is around 55 kRPM, the optimum for Benzene is just reached at the upper limit for 90 kRPM. This can be explained by the difference in $PR$ boundary conditions between the fluids. Benzene turbine has a higher $PR$, around 5.5 at a higher rotational speed to achieve high efficiency in the expansion process.

In terms of blade inlet radius, no clear correlation is found for the two refrigerants while there seems to be a clear increase in performance towards higher radii for Benzene. This can be analysed by looking at the isentropic velocity ratio $\frac{U}{C_{is}}$, shown in Eq. ( 2 ):

$$\frac{U}{C_{is}} = \frac{2\pi r_2 N}{\sqrt{2\Delta h_{0is}}}, \qquad with\ \frac{C_{is}^2}{2} = \Delta h_{0is} = \frac{\gamma R T_{01}}{\gamma - 1}\left[1 - \frac{1}{PR}^{\frac{\gamma-1}{\gamma}}\right] \tag{2}$$

In the case of radial inflow turbines, the peak efficiency theoretically falls on a value of approximately 0.7. Given the higher pressure ratio needed across the turbine when running with Benzene, either $r_4$ or $N$ need to be increased for a similar velocity ratio. For all nozzle inlet angle $\alpha_2$, radius ratio $r_{6s}/r_4$, and rotor inlet blade height $b_4$ the turbine performs best at similar values, especially for the latter parameter, a clear optimum can be found. Overall, it can be concluded through the sensitivity analysis of the geometrical design parameters that a geometry optimized for a specific working fluid should perform similarly for the other fluids if the rotational speed is modified for the different loads.

### 5.2 Verification of Meanline model

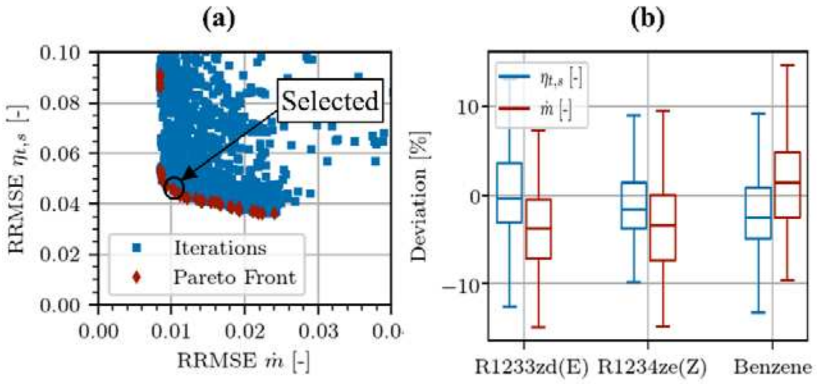
As mentioned in previous sections, meanline modelling approaches based on semi-empirical loss models need to be calibrated for a given application. Thus, the CFD model described in Section 4 was used to calibrate the loss coefficients of the model for each working fluid, respectively. For this purpose, a genetic algorithm is used minimising the RRMSE of both predicted meanline and CFD efficiency and mass flow values, given a specific range of each loss coefficient. A small LHS consisting of 20 samples in total was performed for the calibration, varying between the minimum and maximum of each design parameter. 97

A total of 1000 iterations were performed for each calibration resulting in a *Pareto* front between prediction accuracy for both prediction parameters at which the prediction of one parameter cannot be further improved without reducing the accuracy of the other, as shown in Figure 6(a). The selected set of loss coefficients thus forms a compromise between mass flow and efficiency accuracy. Both the parameter ranges and the resulting loss coefficients are given in Table 4. The calibrations for R1233zd(E) and R1234ze(Z) find similar coefficients, whereas the one for Benzene results in a higher pressure loss coefficient for the nozzle $K_{pl,N}$ as well as passage loss coefficient at the rotor $K_p$.

**Table 4.** Loss coefficient ranges and results from calibration

| Parameter | Range | R1233zd(E) | R1234ze(Z) | Benzene |
|---|---|---|---|---|
| $Bl_2, Bl_4, Bl_6$ | [0.8 – 1.0] | 0.80, 0.93, 0.83 | 0.87, 0.94, 0.80 | 0.80, 0.99, 0.91 |
| $K_{pl,N}$ | [0.0 – 0.3] | 0.09 | 0.07 | 0.28 |
| $K_{inc}$ | [0.6 – 1.4] | 0.72 | 0.64 | 0.91 |
| $K_p$ | [0.0 – 0.4] | 0.08 | 0.03 | 0.27 |
| $K_{cl}$ | [1.0 – 2.0] | 1.13 | 1.48 | 1.01 |

The Sampling performed in Section 5.1 was subsequently used for verification of the different models and resulting deviations from CFD for both efficiency and mass flow, see Figure 6(b), together with the corresponding mean and standard deviation. For all three fluids, the mean error is close to zero, with a standard deviation of approximately 5%, proving a good agreement between CFD and the proposed meanline methodology.




(c)

| | $\Delta\eta_{t-s}$ Mean, Std | $\Delta\dot{m}$ Mean, Std |
|---|---|---|
| R1232zd(E) | -3.88 [%], 4.56 [%] | 0.13 [%], 5.43 [%] |
| R1234ze(Z) | -3.47 [%], 5.37 [%] | -1.09 [%], 4.64 [%] |
| Benzene | 1.21 [%], 4.86 [%] | -1.75 [%], 4.65 [%] |

**Figure 6**. Pareto front for R1233zd(E) calibration (a), error distribution compared to LHS (b) and resulting Mean and Standard deviation (c)

Lastly, a sensitivity study for the different design parameters similar to Figure 5 was performed for the meanline approach, resulting in similar influences on the selected design space. As an example, Figure 7 compares the impact of rotational speed on turbine efficiency which shows a good comparison between 1D and 3D approaches.

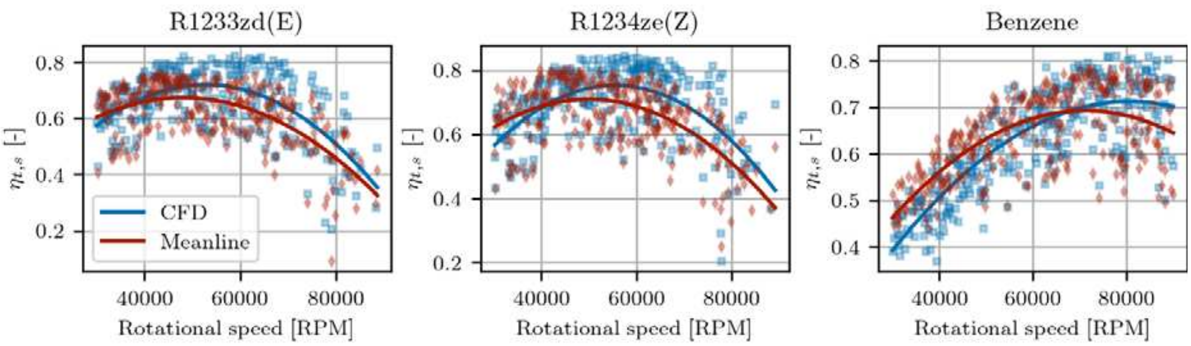



**Figure 7**. Influence of rotational speed on the efficiency for the three different fluids, both CFD and meanline

## 6 CONCLUSION

The research conducted aimed to investigate the accuracy of the meanline modelling approach when considering the non-ideal gas effects of different refrigerants in turbine performance prediction. Three different working fluids were selected. A sensitivity analysis was performed for a broad range of design parameters using a 3D parametric CFD model. Subsequently, calibration was performed for each working fluid using a small data subset of this investigation.

For the two refrigerants R1233zd(E) and R1234ze(Z), the results of the calibration proved to be similar whereas the calibration coefficient for the last working fluid, Benzene, deviated significantly. A comparison between CFD and meanline modelling results was carried out to assess the accuracy of the meanline performance prediction. Results showed that the 1D modelling approach is sufficiently accurate and capable of capturing the real-gas effect on the turbine performance parameters when compared to reference data given an average error within 5%. Therefore, the proposed meanline methodology seems to be suitable to allow a preliminary design analysis of main geometrical parameters when integrated on the system level, and, therefore, can help overcome the computational effort of CFD simulations in the preliminary stage.

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## NOMENCLATURE

| | | |
|---|---|---|
| $A$ | Area | ($m^2$) |
| $b$ | Blade height | (mm) |
| $Bl$ | Blockage factor | (-) |
| C | Absolute velocity | (m $s^{-1}$) |
| C_p | Heat capacity | (J $kg^{-1}$ $K^{-1}$) |
| CFD | Computational fluid dynamics | |
| GWP | Global Warming Potential | (-) |
| $\Delta h_{0is}$ | Isentropic enthalpy drop | (J $kg^{-1}$) |
| $i$ | Incidence angle | (°) |
| $K$ | Loss coefficient | (-) |
| LHS | Latin Hypercube Sampling | |
| $L$ | Loss | (kJ $kg^{-1}$) |
| $\dot{m}$ | Mass flow rate | (kg $s^{-1}$) |
| n | Rotational speed | (rps) |
| $N$ | Rotational speed | (rpm) |
| ORC | Organic Rankine Cycle | |
| ODP | Ozone Depletion Potential | (-) |
| $P$ | Pressure | (Pa) |
| $PR$ | Pressure ratio | (-) |
| $r$ | Radius | (mm) |
| RRMSE | Relative mean square error | (-) |
| $T$ | Temperature | (K) |
| $U_t$ | Blade tip speed | (m $s^{-1}$) |
| $W$ | Relative velocity | (m $s^{-1}$) |
| $y_+$ | Y plus | (-) |
| $z$ | Axial position | (mm) |
| $Z_{max}$ | Rotor length | (mm) |

**Greek Letters**

| | | |
|---|---|---|
| $\alpha_2$ | Nozzle vane angle | (°) |
| $\beta_6$ | Rotor blade angle | (°) |
| $\gamma$ | Adiabatic coefficient | (-) |
| $\theta$ | Circumferential angle | (°) |
| $\rho$ | Density | (kg $m^{-3}$) |
| $\eta_{t-s}$ | Total-to-static efficiency | (-) |
| $\eta_{ORC}$ | Cycle efficiency | (-) |

**Subscripts**

| | |
|---|---|
| 0 | Stagnation conditions |
| 1 | Volute inlet |
| 2 | Nozzle inlet |
| 3 | Nozzle outlet |
| 4 | Rotor inlet |
| 6 | Rotor outlet |
| evap | Evaporating |
| cond | Condensing |
| inc | Incidence |
| is | Isentropic |
| p | Passage loss |
| pl | Pressure loss |
| sw | Swirl loss |
| tc | Tip clearance loss |
| LE | Leading edge |
| TE | Trailing edge |



## REFERENCES


Alvarez-Regueiro, E., Yang, B., Barrera-Medrano, E., Martinez-Botas, R., & Rajoo, S. (2022). Optimization of an organic Rankine cycle radial turbine using a reduced-order model coupled with computational fluid dynamics. *Journal of Engineering for Gas Turbines and Power*, *144*(10).

Bell, I. H., Quoilin, S., Wronski, J., & Lemort, V. (2013). Coolprop: An open-source reference-quality thermophysical property library. *ASME ORC 2nd International Seminar on ORC Power Systems*.

Da Lio, L., Manente, G., & Lazzaretto, A. (2017). A mean-line model to predict the design efficiency of radial inflow turbines in organic Rankine cycle (ORC) systems. *Applied Energy*, *205*, 187–209.

Desideri, A., Gusev, S., van den Broek, M., Lemort, V., & Quoilin, S. (2016). Experimental comparison of organic fluids for low temperature ORC (organic Rankine cycle) systems for waste heat recovery applications. *Energy*, *97*, 460–469.

Eyerer, S., Dawo, F., Kaindl, J., Wieland, C., & Spliethoff, H. (2019). Experimental investigation of modern ORC working fluids R1224yd(Z) and R1233zd(E) as replacements for R245fa. *Applied Energy*, *240*, 946–963.

Eyerer, S., Wieland, C., Vandersickel, A., & Spliethoff, H. (2016). Experimental study of an ORC (Organic Rankine Cycle) and analysis of R1233zd-E as a drop-in replacement for R245fa for low temperature heat utilization. *Energy*, *103*, 660–671.

FRIENSHIP SYSTEMS. (n.d.). *Computer aided engineering system empowering simulation, CAESES* (v. 4.4.2). https://www.caeses.com.

Futral, S. M., & Wasserbauer, C. A. (1965). Off-design performance prediction with experimental verification for a radial-inflow turbine. *National Aeronautics and Space Administration, NASA TN D-2621*.

Hohenberg, K., Martinez-Botas, R., Łuczyński, P., Freytag, C., & Wirsum, M. (2021). Numerical and experimental investigation of a low order radial turbine model for engine-level optimization of turbocharger design. *Journal of Engineering for Gas Turbines and Power*, *143*(10).

Invernizzi, C., Iora, P., & Silva, P. (2007). Bottoming micro-Rankine cycles for micro-gas turbines. *Applied Thermal Engineering*, *27*(1), 100–110.

Invernizzi, C. M., Iora, P., Preißinger, M., & Manzolini, G. (2016). HFOs as substitute for R-134a as working fluids in ORC power plants: A thermodynamic assessment and thermal stability analysis. *Applied Thermal Engineering*, *103*, 790–797.

Japikse, D., & Baines, N. C. (1997). *Introduction to turbomachinery*. Concepts ETI, Inc.

Lang, W., Colonna, P., & Almbauer, R. (2013). Assessment of waste heat recovery from a heavy-duty truck engine by means of an ORC turbogenerator. *Journal of Engineering for Gas Turbines and Power*, *135*(4).

Maizza, V., & Maizza, A. (2001). Unconventional working fluids in organic Rankine-cycles for waste energy recovery systems. *Applied Thermal Engineering*, *21*(3), 381–390.

Nemati, A., Nami, H., Ranjbar, F., & Yari, M. (2017). A comparative thermodynamic analysis of ORC and Kalina cycles for waste heat recovery: A case study for CGAM cogeneration system. *Case Studies in Thermal Engineering*, *9*, 1–13.

Noriega, S. B., De Servi, C., & Colonna, P. (2020). Hybrid electric powertrain for long-haul trucks and buses: preliminary analysis of a new concept based on a combined cycle power plant. *Journal of the Global Power and Propulsion Society*, *4*, 63–79.

Rahbar, K., Mahmoud, S., Al-Dadah, R. K., & Moazami, N. (2015). Parametric analysis and optimization of a small-scale radial turbine for Organic Rankine Cycle. *Energy*, *83*, 696–711.

Robertson, M. C. (2019). *Thermofluid optimisation of turboexpanders for mobile organic Rankine cycle systems* [PhD Thesis]. Imperial College London.

Spraker, W. A. (1987). Contour clearance losses in radial inflow turbines for turbochargers. *ASME Paper*, *87-ICE-52*.

Sun, W., Yue, X., & Wang, Y. (2017). Exergy efficiency analysis of ORC (Organic Rankine Cycle) and ORC-based combined cycles driven by low-temperature waste heat. *Energy Conversion and Management*, *135*, 63–73.

Teng, H., Regner, G., & Cowland, C. (2007, April 16). *Waste heat recovery of heavy-duty diesel engines by organic Rankine cycle part II: working fluids for WHR-ORC*.

Verbruggen, F. J. R., Hoekstra, A., & Hofman, T. (2018). Evaluation of the state-of-the-art of full-electric medium and heavy-duty trucks. *31st International Electric Vehicle Symposium and Exhibition, EVS 2018 and International Electric Vehicle Technology Conference 2018, EVTeC 2018*.

Wasserbauer, C. A., & Glassman, A. J. (1975). FORTRAN program for predicting off-design performance of radial-inflow turbines. *National Aeronautics and Space Administration, NASA TN D-8063*.

Whitfield, A., & Baines, N. C. (1990). *Design of radial turbomachines*. John Wiley and Sons Inc.

Yang, J., Ye, Z., Yu, B., Ouyang, H., & Chen, J. (2019). Simultaneous experimental comparison of low-GWP refrigerants as drop-in replacements to R245fa for Organic Rankine cycle application: R1234ze(Z), R1233zd(E), and R1336mzz(E). *Energy*, *173*, 721–731.

Yu, G., Shu, G., Tian, H., Huo, Y., & Zhu, W. (2016). Experimental investigations on a cascaded steam-/organic-Rankine-cycle (RC/ORC) system for waste heat recovery (WHR) from diesel engine. *Energy Conversion and Management*, *129*, 43–51.

This book contains the compilation of works contributed to the *7th International Seminar on Organic Rankine Cycle Power Systems* (ORC 2023), held in Seville between the 4th and 6th of September 2023. The event was hosted by Universidad de Sevilla on behalf of the Knowledge Centre on Organic Rankine Cycle Technology (KCORC), incorporated in The Netherlands.

The ORC conference, organized biennially, stems as the only conference that is specific to ORC technology, therefore gathering a diverse community whose affiliation spans across all the interested stakeholders, not only in this particular technology but also and in a broader context, in the energy transition. Original equipment manufacturers, professional associations, end-users, investors, policy makers, academics, scientists feel at home at ORC 2023.

The almost 100 proceedings in this book cover a wide variety of topics, from fundamentals to system integration through component design, accounting for thermodynamic performance as well as component design. In addition to this, and as a new track in 2023, works on heat pump technology were also accepted in order to raise awareness of the strong ties between both technologies, specifically in energy storage applications.

This book provides an excellent overview of the current maturity of power systems based on Organic Rankine Cycle technology for applications as diverse as geothermal and waste heat recovery in industry or downstream of other prime movers (e.g., marine applications). It is also an excellent source of information to understand the current challenges faced by the technology, stemming from a very competitive market and increasingly stringent environmental regulations.

The organizers of ORC 2023 hope that the reader finds this work as exciting as the attendees to the conference and, maybe, make the decision to join the 8th edition to the conference in 2025.

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