THERMODYNAMIC AND DESIGN CONSIDERATION OF A MULTISTAGE AXIAL ORC TURBINE FOR COMBINED APPLICATION WITH A 2 MW CLASS GAS TURBINE FOR DEZENTRALIZED AND INDUSTRIAL USAGE

René Braun[1]*, Karsten Kusterer[1], Kristof Weidtmann[1], Dieter Bohn[2]

[1]B&B-AGEMA,
Aachen, NRW, Gemany
braun@bub-agema.de

[2]Aachen University,
Aachen, NRW,Germany
Dieter.bohn@rwth-aachen.de

* Corresponding Author

ABSTRACT

The continuous growth of the part of renewable energy resources within the future mixture of energy supply leads to a trend of concepts for decentralized and flexible power generation. The raising portion of solar and wind energy, as an example, requires intelligent decentralized and flexible solutions to ensure a stable grid and a sustainable power generation.
A significant role within those future decentralized and flexible power generation concepts might be taken over by small to medium sized gas turbines. Gas turbines can be operated within a large range of load and within a small reaction time of the system. Further, the choice of fuel, burned within the gas turbine, is flexible (e.g. hydrogen or hydrogen-natural gas mixtures). Nevertheless, the efficiency of a simple gas turbine cycle, depending on its size, varies from 25% to 30%.
To increase the cycle efficiencies, the gas turbine cycle itself can be upgraded by implementation of a compressor interstage cooling and/or a recuperator, as examples. Those applications are cost intensive and technically not easy to handle in many applications. Another possibility to increase the cycle efficiency is the combined operation with bottoming cycles. Usually a water-steam cycle is applied as bottoming cycle, which uses the waste heat within the exhaust gas of the gas turbine. In small to mid-sized gas turbines the temperature and heat amount within the exhaust gas are often not sufficient to operate a water steam cycle efficiently. Further, in industrial applications, a part of the heat, within the exhaust gas, is often used in secondary processes which lower the total amount of heat, which can be transferred to a bottoming cycle.
An alternative to water-steam bottoming cycles can be given by organic Rankine cycles (ORC) based on organic fluids. An advantage of organic fluids is the characteristic of evaporating at lower temperatures and lower heat amounts and thus, the usability in a Rankine cycle even at low heat source temperatures.
This paper discusses an ORC process design for a combined application with a simple cycle gas turbine in the 2 MW class. The thermodynamic cycle configuration is shown and it will be pointed out that the cycle efficiency (simple GT) of 26.3% can be increased to more than 36% by application of a bottoming ORC cycle (simple GT+ORC). A key component within the ORC cycle is the turbine. This paper shows the results of an extended aerodynamic ORC axial turbine pre-design based on the thermodynamic cycle considerations. Within the design study the real gas properties of the organic fluid are taken into account. Based on the outcome of the pre-design a 3D aerodynamic axial turbine design is investigated. By application of CFD simulations the turbine design has been optimized and the ORC power output could be increased from 636kW (0D-design) to 659 kW (CFD-design), which lead to a combined cycle efficiency of more than 36%.

1. INTRODUCTION

During the last decade the importance of intelligent, flexible and sustainable energy conversion systems is steadily growing. Thereby the application of systems based on organic Rankine cycles (ORC) get more and more in focus in several technical disciplines as Quoilin et al. (2013) have shown in their survey of ORC systems. The technology of Organic Rankine cycles are nowadays applied to many kinds of technology fields, such as solar thermal, geothermal or biomass power plants, in offshore platform applications, waste heat recovery systems and even in heavy trucks.
This paper deals with the application of ORC system as bottoming cycles for usage of waste heat from a primary cycle. Due to the trend of decentralized power generation, as a consequence of the growing part of renewables within the future energy mixture, small to mid-sized flexible power generation systems are required. The application of gas turbines can full fill the requirements in terms of system flexibility and variable sizes. A disadvantage of simple gas turbine cycles are the low efficiencies. A possibility to increase the simple cycle efficiency is the application of a bottoming cycle. In many cases a conventional bottoming cycle based on water/steam is not economical. In such cases a bottoming cycle based on organic fluids can be a successful alternative. Nguyen et al. (2012) and Pierobon et al. (2013) have shown in their investigations the significant benefit which can be achieved by the application of ORC system as bottoming cycle to a gas turbine for offshore solution. Kusterer et al. (2013) have shown the potential of a combined cycle configuration of a 2 MW class gas turbine with an ORC within a conceptual process design study.
As various kinds of organic fluids are available, a usage-oriented working fluid selection process is necessary, in order to find the best candidate for the process of interest. As the fluid properties of organic fluids are different in a huge range and the consideration of the real gas behavior of such fluids is of highest importance within those selection analyzes, many researchers have investigated in this field (e.g. Quoilin et al. (2012); Saleh et al. (2007)) For the best choice of suitable fluids the practicability and feasibility of the thermodynamic results have to be taken into account, but also the design of the cycle, the heat transfer behavior within the heat exchanger and the relating volume flows through the expansion machines are of highest interest for an ORC fluid selection. There is a significant amount of literature dedicated to the evaluation of suitable working fluids, due to the complexity of the topic (e.g. Kusterer et al.(2014); Fernandez et al. (2011); Lai et al. (2011); Sauret et al. (2011); Chen et al. (2011)).
The calculation of ORC processes has to consider the real gas properties of the analyzed fluids, as the approximation with ideal gas equations is not valid for the most of the fluids. Therefore the development of suitable equation of states for analyzing of organic fluids is investigated by a lot of researchers (e.g. Weingerl et al. (2001), Wei et al. (2000), Miyamoto and Watanabe (2003)). Harink et al (2010) have analyzed the influence of different CFD solvers, turbulence models and equation of states to the performance of a radial ORC turbine predicted by the application of different CFD solvers. They have shown that the accuracy of the results is highly addicted to the related models and solvers.
This paper deals with the applicability of conventional steam turbine design procedures and how far those are transferrable to the design of axial turbines using organic mediums. The analyzed design is based on a thermodynamic process study of a combined application of an ORC and a simple gas turbine (2 MW class). As organic working fluid pentane is used in the ORC cycle

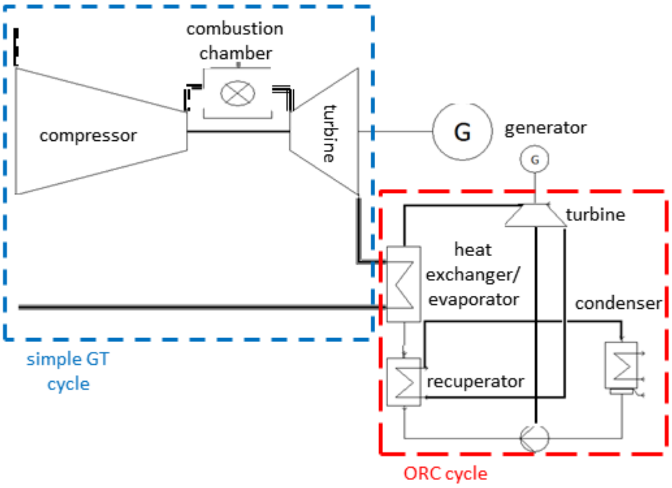
2. Thermodynamic Cycle Modeling

2.1 Software tool for process calculations

The thermodynamic cycle modeling of the combined gas turbine and ORC has been performed by a “Thermodynamic Design Tool” (TDT) application, developed by B&B-AGEMA. TDT supports the design and calculation of single and combined energetic processes based on a 0D thermodynamic approach, using different fluids selected by the user. Combined cycles (e.g. gas turbine + ORC) can be investigated directly and with consideration of interactions. It comprises real gas behavior of several fluids and mixtures. The thermodynamic processes can be visualized in parametric thermodynamic diagrams, e.g. enthalpy/entropy, temperature/pressure, including precisely tabled thermodynamic values. The calculation method for compression and expansion processes under real gas behavior are based on Lüdtke (2004).

2.2 Combined Cycle Calculation (0D-TDT)

The cycle design of the combined gas turbine and organic Rankine cycle configuration is shown in Figure 1. The combined cycle contains a simple cycle gas turbine and a recuperated organic Rankine cycle. On the right hand side of the Figure the thermodynamic parameter of the single components are illustrated in order to classify the cycles. The simple gas turbine cycle results in an efficiency of 26.3%, this efficiency includes a thermal to electrical factor of 0.893, which approximates the mechanical losses of rotating parts and transmission losses within the generator.



process thermodynamic parameter

GT		ORC	
compressor		pump	
η_s	81.00%	η_s	75%
π	10.4	π	23.8
mass flow	8 kg/s	mass flow	6.5 kg/s
combustor			
pressure loss	3.40%		
turbine		turbine	
η_s	89.40%	η_s	85.00%
exhaus diffusor			
pressure loss	4.5%		
Both cycles			
heat exchanger/evaporator/recuperator/condenser			
pressure loss	2%		
effectiviness	90%		
efficiency	90%		

Figure 1: combined cycle configuration of simple gas turbine and organic Rankine cycle

The resulting combined cycle efficiency is 35%, whereas the power output of the ORC is 625.6 kW (without consideration of the condenser energy consumption and an assumed thermal to electrical factor, similar as for the GT cycle, of 0.9). Thus, the efficiency of the simple cycle could be increased by more than 8%pts., by application of a bottoming ORC cycle.
As the focus of this paper is on the design of the axial turbine, the calculated turbine power within the thermodynamic 0-D process calculation (TDT) has to be seen as reference and starting point for the more detailed turbine design. The ORC 0-D net turbine power output is 707 kW which is conform to a specific work of 108.8 kJ/kg. For the further design study the specific work is set as value of comparison.

3. ORC Turbine Design

3.1 Definition of Main Turbine Parameter and Boundary Conditions

As first, a simple 1D design approach has been performed by calculating the characteristic machine parameters ψ_{yM}, δ_M and σ_M. The characteristic machine parameters are defined as:

$$\psi_{yM} = \frac{\left(\frac{n}{n-1}\right) RT \left(\pi^{\frac{n-1}{n}} - 1\right)}{\frac{1}{2}u^2} \tag{1}$$

$$\delta_M = \frac{|\psi_{yM}|^{\frac{1}{4}}}{\left|\frac{4\dot{V}u}{\pi D_B^2}\right|^{\frac{1}{2}}} \tag{2}$$

$$\sigma_M = \frac{\left|\frac{4\dot{V}u}{\pi D_B^2}\right|^{\frac{1}{2}}}{|\psi_{yM}|^{\frac{3}{4}}} \tag{3}$$

Based on the characteristic machine parameters a first assumption of rotational speed and stage numbers can be defined for the turbine by using the Cordier diagram (see Figure 2) and by variation of the reference diameter and the rotational speed. The Cordier diagram presents ranges of machine characteristic parameters and is based on empirical data of real single stage machines. The diagram has been developed by O. Cordier in 1953. The Cordier diagram and the definitions of the characteristic machine parameters are only defined for single stage compressors and turbines, but they are useful to get a first impression of the needed stage numbers, the reference diameters and the rotational speed of compressors and turbines. Thus, the diagram is used in order to determine the rotational speed and stage number for the ORC turbine.

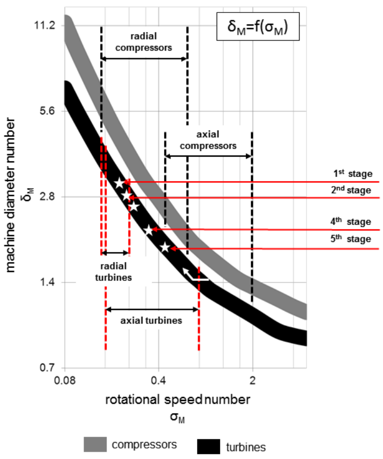


Figure 2: Cordier Diagram

The analysis based on the Cordier diagram and the related formulas (Equ.2 and 3) leads to a rotational speed of the turbine of 300 s^{-1}, which is related to a σ_M of 0.14 for the whole turbine. The number of stages is evaluated with 5, by calculating the characteristic parameters for each stage and manipulating the stage loading and influencing parameters till all stages are within the range of experienced axial machines of the Cordier diagram. By this procedure the required number of stages can be assumed and has to be verified as feasible in the detailed turbine design.
For simplification purposes the 5 stages of the turbine shall be designed as quasi-repetition stages, with constant stage flow coefficients φ

$$\varphi = \frac{c_m}{u} \tag{4}$$

where u is the circumferential velocity of the blade outlet and c_m the velocity of the flow in the meridian section. By the stage load coefficient ψ_h the loading of each stage is characterized:

$$\psi_h = \frac{\Delta h}{u^2 \frac{1}{2}} \tag{5}$$

where Δh describes the difference of the specific total enthalpy between outlet and inlet of the stage. Another important parameter within the design of a turbomachine is the enthalpy reaction number ρ_h:

$$\rho_h = \frac{\Delta h_{vane}}{\Delta h_{vane} + \Delta h_{vblade}} \tag{6}$$

it describes the load distribution between the vane and the blade of a stage.
Within the iterative 1D mean line design approach φ, ψ_h, ρ_h, the rotation rate, the mass flow of the organic medium and the diameter of the hub contour are initial values. Those values are necessary, in order to start the iterative process presented within the following section. The characteristic stage parameters are given in Table 1.

Table 1: turbine stage parameter

	1. stage	2. stage	3. stage	4 stage	5. stage
φ	0,3	0,3	0,3	0,3	0,3
ρ_h	0,5	0,5	0,5	0,5	0,5
ψ_h	-2,1	-2,1	-2	-2	-1,6

3.2 1D Mean Line ORC Turbine Design

In order to evaluate the main design parameters (e.g. geometrical parameters, flow parameters, etc.) of the ORC turbine, a 1D design has to be performed. A common design practice for gas or steam turbines is the application of a 1D Mean Line approach. This paper analyses the applicability of the common design procedure for the design of the axial ORC turbine and if the conventional design laws and methods are also applicable for turbines working with organic mediums. Therefore the changes of state, which have to be calculated within the mean line approach, have been iteratively solved by consideration of real gas equations and by implementation of property tables based on the data base implemented within the thermodynamic design tool (TDT). The iterative solving of the change of state is illustrated. The iteration is done till two criteria (ϵ_P and ϵ_T) are fulfilled. The main equations (7 and 8) to solve the change of state are based on RIST (1996) and shall not be further explained here. Thus, the 1D mean Line approach for application for ORC turbines becomes more complex as for example for gas or steam turbines in cause of missing data correlations or applicable real gas models.

$$\frac{p_2}{p_1} = \left[1 + \frac{\Delta h_s + (K_T \cdot v)_{1,2} \cdot p_1 \left(\frac{\tilde{p}_2}{p_1} - 1\right)}{c_{p_{1,2s}} \cdot T_1}\right]^{\left(\frac{\kappa \cdot [1+K_T]}{\kappa \cdot [1-K_p]-1}\right)_{1,2}} \tag{7}$$

$$\frac{p_2}{p_1} = \left(\frac{T_2}{T_1}\right)^{\left(\frac{\kappa \cdot [1+K_T]}{\kappa \cdot [1-K_p]-1}\right)_{1,2}} \cdot \exp\left(-\frac{\Delta s}{(K \cdot R_N \cdot [1+K_T])_{1,2}}\right) \tag{8}$$

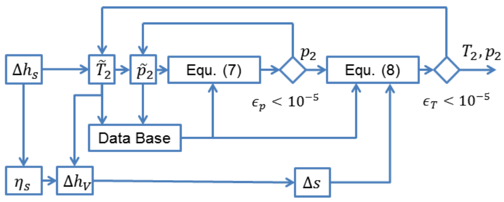


Figure 3: iterative scheme of solving the change of state

The 1D mean line process is illustrated in Figure 4. As it can be seen, due to the consideration of the real gas behavior of the organic fluid, the design process of the 1D mean line approach becomes complex and requires an iterative solutions. However, the results of the mean line approach are the flow angles in the mean line and the geometrical information about the hub and casing contour. Due to feasibility issues, additional boundary (BC) conditions have been considered within the design approach. Therefore the minimal size of a blade height is set to 5mm and additionally, in order to avoid a supersonic flow within the cascade, the inflow Mach number (Ma) should be lower than 0.3 and the outflow Ma lower than 0.9 (for each blade). Aerodynamic losses as well as profile losses have been approximated within the 1D design approach by empirical loss models (LM) with corresponding loss coefficients ξ. Those models are addicted to empirical models used for the design of steam turbines, as specific models are not available for organic mediums. The mean line approach is mainly divided into two sections, the thermodynamic (TD) and aerodynamic (AD) part as illustrated in Figure 4.

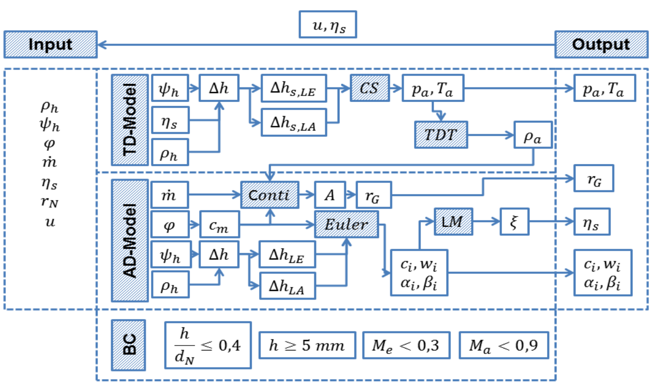
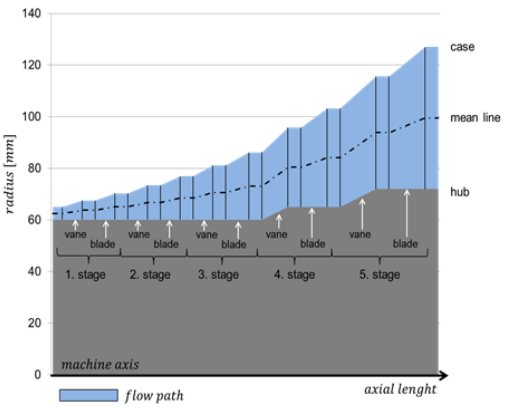
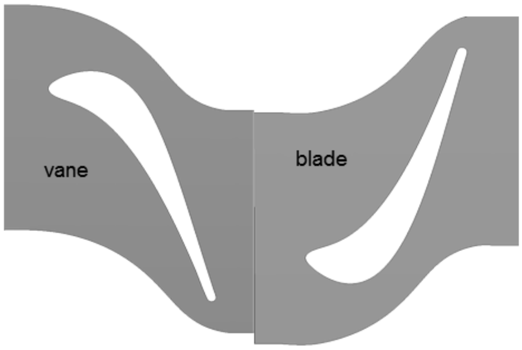


Figure 4: mean-line design process

The mean line approach results in the main geometrical information for the turbine design, as the hub and casing contour as well as the flow angles within the mean line section. Based on this information the main flow path and the profiles of the vanes and blades can be designed, as illustrated in Figure 5.



turbine cross section view



1st stage mean line profile

		vane	blade	vane	blade	vane	blade	vane	blade	vane	blade
aerodynamic mean line results											
velocity (abs. frame of reference)	in	35,350	130,662	37,222	137,367	39,206	143,180	41,525	163,851	47,954	174,503
	out	130,662	37,222	137,367	39,205	143,180	41,525	163,851	47,954	174,503	57,021
velocity (rel. frame of reference)	in	-	36,455	-	378,465	-	399,075	-	45,467	-	57,185
	out	-	133,353	-	140,667	-	147,798	-	171,435	-	187,073
angle (abs. frame of reference)	in	90,00	16,015	98,355	15,939	99,090	16,184	95,804	16,106	97,438	17,699
	out	16,015	98,355	15,939	99,091	16,184	95,804	16,106	97,438	17,699	99,266
angle (rel. frame of reference)	in	-	81,433	-	85,386	-	90,289	-	91,363	-	111,918
	out	-	163,969	-	164,026	-	163,767	-	163,897	-	162,493

Figure 5: mean line results - flow passage and flow angles

Due to the quasi-repetition stage design the mean line is not following on a constant radial position, but increasing in its radial position in axial direction. The resulting flow passage and the mean line location can be seen in on the left hand side in Figure 5. On the right hand side the contour of the vane and blade profiles of the 1st stage is exemplarily shown. Based on the resulting flow angles, the profiles have been designed within an in-house profile generator, which is not be further explained. As the hub and casing contour is known, a 3D turbine design can be evaluated. The first design considers non twisted airfoils, which means constant shapes in radial direction.
The calculated specific work is 104.54 kJ7kg, a difference of 3.9% to the 0-D process calculation. The reasons can be seen in the application of an iterative solving of the changes of state and the application of empirical loss models.

4. ORC Turbine CFD Simulation

The CFD simulations within this study are performed with the commercial code STARCCM+. The simulation set up considers the realizable k-ε turbulence model for the turbulence expression and considers the thermodynamic fluid properties of pentane as pressure and temperature depended tables for the thermal conductivity and dynamic viscosity. The specific heat has been prescribed as polynomial function. To consider the real gas behavior the Peng Robinson real gas model has been selected. The inlet conditions are prescribed as total conditions of temperature and pressure and the outlet is considered as a static pressure outlet.

4.1 CFD Simulation of first mean line design approach

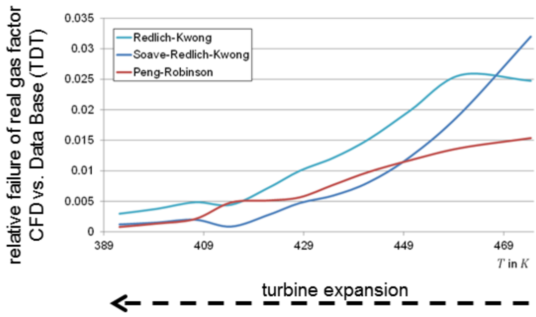
The results of the numerical analysis (CFD 1st) in comparison to the 1D and 0-D design results are listed in Figure 6. It can be seen that small deviations are calculated by each approach by a direct comparison. The main key factors why there are differences are driven by two main factors: i) the consideration of the real gas behavior and ii) the consideration of losses. Within the numerical analyses those two factors are directly connected to each other, in the 1D design there are indirect connected via loss models and in the 0-D design the losses are indirect considered by the prediction of isentropic component efficiencies. Thus, the different applications of the losses lead to small deviations, which need to be adjusted and fine-tuned.

Nevertheless, in a direct comparison of CFD and 1D design, the turbine principally shows the expected behavior by comparing the calculated pressure expansion line through the turbine, shown on the right hand side of Figure 6. The pressure lines are perfectly overlapping, whereas the static temperature distribution shows a difference. The difference of the static temperature is mainly produced within the first stage, the location where the applied real gas model within the CFD shows its highest failure in the calculation of the change of state, which has then direct influence to the prediction of the static parameters. After the first stage the static temperature lines are nearly parallel, this shows a similar prediction of CFD and 1D.

The accuracy of applicable real gas models in STARCCM+ in regard of the real gas factor determined by the real gas model versus the data base is as well illustrated in Figure 6. The failure rate is decreasing along the expansion line and has its maximum in the first stage for all models, but the Peng Robinson model shows the smallest deviations. Within the first stage the real gas factor of pentane is around 0.5 and as smaller the real gas factor as higher the failure rate of the real gas models. The models are not validated for pentane and thus have increasing failure rates for low real gas factors and this failure rate is then transported throughout the calculation of the static parameters.

Nevertheless, it can be said that the numerical investigation has shown a good performance of the turbine and the results are within an acceptable range of deviation for the first run.

		CFD 1st	1D-mean-line	0D-TDT
Mass flow kg/s		6.72	6.5	6.5
Inlet	temperature in K	474.65	474.41	474.41
	pressure in bar	29.734	29.772	29.77
Outlet	temperature in K	400.88	392.04	390.38
	pressure in bar	1.467	1.467	1.354
Spezific work a_T in kJ/kg		-111.515	-104.544	-108.8



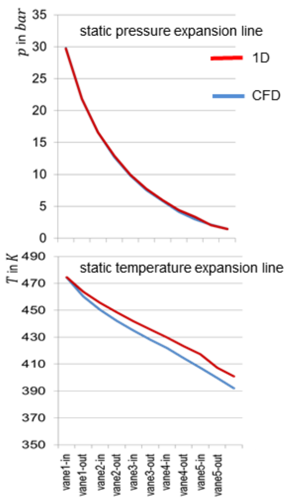


Figure 6: CFD results compared to 1D mean line design parameter

In order to compare the results, the specific work can be taken into account. It can be seen, that the prediction of the CFD is 6.3% higher than the estimated work by the 1D mean line design, whereas the deviation between the 1st CFD and the thermodynamic process calculation is 2.5%. Of course, a certain value of uncertainty is not avoidable but it can be assumed, that the implemented models within the 1D-design are not perfectly transferrable to the analyzed ORC turbine, as most of the correlations (especially for loss prediction, etc.) are specialized for steam turbine designs. Nevertheless, the 1D-design based on common practices of steam turbine design procedures is applicable and the resulted design is already close to a feasible design also for an organic fluid application. Based on the CFD results a turbine optimization process can be started to further increase the power output and to optimize the aerodynamic behavior. This can be done for instance by optimizing the shape of the profiles and/or to twist the blades.

4.2 CFD based optimization: consideration of radial twisting

Despite the deviations between the CFD and the 1D mean line results, an optimization of the turbine can be performed based on the results and experiences observed from the CFD analysis. Thus, a twisting of the blades has been analyzed and compared on the basis of a numerical delta analysis with the first step CFD calculation of section 4.1. The evaluation of the twist parameters can be evaluated on two approaches: i) directly by evaluation of flow angles within the CFD or ii) by empirical models. For the determination of the twist parameters of the blades the second method has been chosen, as empirical models can be directly applied to a 1D Mean Line approach and thus show more potential to considered in an early design stage. Two models have been compared: a free vortex model and a stator outflow angle model (Schuh (2012)). The models determine the radial flow angle distribution of the blades based on the result of the mean line calculation. The twist of the blade can then be realized within the profile generator. In order to validate the applicability of the models, the predicted flow angles from the two models have been compared to the flow angles from the CFD analysis. A comparison is shown exemplarily for the outlet of the 3rd vane in Figure 7. Within the analysis the free vortex model has been evaluated as best practice for an application.

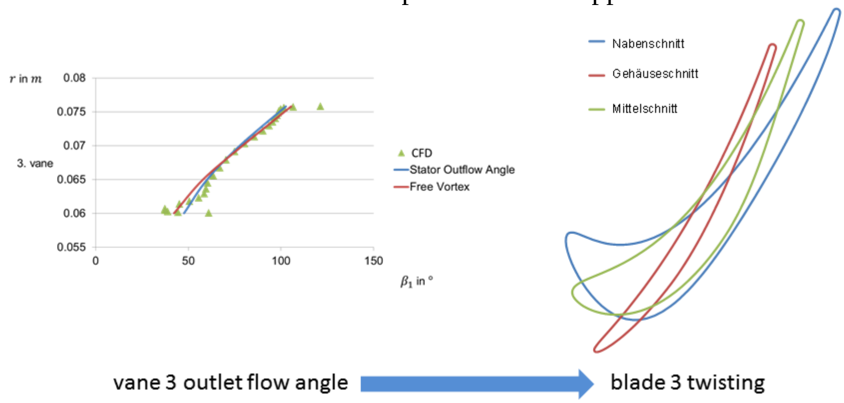


Figure 7: radial outlet flow angle distribution of vane 3 and blade 3 twist

The main results of the CFD considering the twisted profiles are shown and compared in Figure 8. Due to a twist of the profiles the flow losses induced by incidence could be reduced and thus, the specific work has been decreased by 1.4%.

		CFD twist	CFD-1st	1D-mean-line	0D-TDT
Mass flow kg/s		6.48	6.72	6.5	6.5
Inlet	temperature in K	474.76	474.65	474.41	474.41
	pressure in bar	29.81	29.734	29.772	29.77
Outlet	temperature in K	399.8	400.88	392.04	390.38
	pressure in bar	1.467	1.467	1.467	1.354
Specific work a_T in kJ/kg		-113.034	-111.515	-104.544	-108.8

Figure 8: comparison of results

6. CONCLUSIONS

Within this paper the design of an ORC turbine has been presented. Based on the results of a 0D thermodynamic process calculation a 1D turbine design has been performed on the basis of a mean line approach. The turbine has been designed as quasi repetition stages with consideration of empirical loss models and correlations applied from the design of steam turbines. The design has been transferred to a 3D turbine design and calculated by CFD. The main results of the analysis are that the design rules and methods of designing steam turbines are applicable for the design of ORC turbines. Nevertheless, some deviations and uncertainties are carried out, which require an adjustment of correlations and loss models in order to increase the accuracy of the ORC turbine design. The analysis by CFD requires a high accuracy in defining the fluid properties and the chosen physical models within the calculation. In order to minimize the design effort of an ORC turbine, it is intended to achieve a good turbine design already in early design steps, as within the 1D design approach. Therefore the implementation of empirical models and correlations are necessary. Those models are available for steam turbine design but not verified for organic turbines. The analysis of empirical models for twisted blade designs has shown the applicability of those models as well for a pentane axial turbine.

The thermodynamic process design has shown the potential of the application of ORC systems as bottoming cycles to small and middle size gas turbines as the cycle efficiency could be increased from 26% to 36%. Especially the application of axial turbine designs offer a high potential, as high turbine efficiencies can be achieved.

NOMENCLATURE

A	specific work	[kJ/kg]
A	Flow channel area	[mm²]
c	velocity magnitude (absolute system)	[m/s]
d	diameter	[mm]
w	velocity magnitude (relative system)	[m/s]
c_m	velocity in meridian section	[m/s]
D_B	reference diameter	[m]
h	specific enthalpy	[kJ/kg]
P	static pressure	[kg/m²]
r	radius	[m],[mm]
R	specific gas constant	[J/(kg*K)
T	static temperature	[K]
u	circumferential velocity	[m/s]
$\dot{V}$	volume flow	[m³/s]
α	absolute flow angle	[°]
β	relative flow angle	[°]
δ_M	machine diameter number	
σ_M	rotational speed number	
φ	flow coefficient	
ψ_h	stage loading number	
ψ_{yM}	machine loading number	
ρ_h	enthalpy reaction number	
η	efficiency	
π	pressure ratio	

Subscript

a	outlet position
e	inlet position
G	casing
i	inlet position
N	hub
n	counting variable
s	isentropic

Abbreviation

AD	Aerodynamic
CFD	Computational Fluid Dynamics
Conti	Continuity
CS	Change of State
Euler	EULER equations
GT	Gas Turbine
LA	blade
LE	vane
LM	Loss Models
ORC	Organic Rankine Cycle
TD	Thermodynamic
TDT	Thermodynamic design Tool

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