EFFICIENCY CORRELATIONS FOR OFF-DESIGN PERFORMANCE PREDICTION OF ORC AXIAL-FLOW TURBINES

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ABSTRACT

Organic Rankine Cycle (ORC) power systems are efficient and cost-effective to convert heat from low/medium temperature heat sources into electrical or mechanical power. Depending on the time behaviour of the heat and cold sources, the ORC can be operated close to the design point or at part-load, which is generally affected by lower efficiency with respect to the nominal point. Axial-flow turbines are the dominant type of expander for large-scale ORCs and their behaviour is crucial for the overall ORC performance. While correlations for efficiency estimation of the design of ORC axial-flow turbines are already available, only very few works have studied the off-design of this component. Two MATLAB® mean-line tools of ORC axial-flow turbines, one for the design, AxialOpt, and one for the off-design, AxialOff, are here presented and used to develop part-load efficiency correlations based on simple thermodynamic quantities. The codes have been validated against data and measurements available in literature. A total of eight turbines from different fields of application have been designed and performance maps have been developed to define the efficiency based on the relative difference in specific enthalpy over the turbine and the relative outlet volume flow rate with respect to the nominal point. The coefficient of determination for the fitting was larger than 99%. A test case not included in the curve fitting was studied to prove the prediction capabilities of the proposed correlations. They could predict the part-load behaviour of the turbine with coefficients of determination above 90% for one, two and three stages. The results of the work can be used in ORC system analyses to estimate the turbine performance at part-load conditions prior to its design.

1. INTRODUCTION

Organic Rankine Cycle (ORC) power systems are an efficient and cost-effective technology to produce electricity or heat and power from low/medium-temperature energy sources that are not technically or economically recoverable with conventional energy systems (Macchi and Astolfi 2017). In Tartière and Astolfi (2017), it is reported that 2.7 GW of installed capacity were reached in 2017, with ORC units ranging from some kW_e to multi-MW_e. They also show that most of the units have an installed power larger than 500 kW_e, over which axial-flow turbines are the dominant type of expander (Meroni 2018). Several codes have been developed to design and optimize axial-flow turbines for ORC, including Axtur (Macchi and Perdichizzi 1981), TURAX (Meroni et al. 2016b), and AxialOpt (Agromayor and Nord 2019b).

The preliminary design is based on mean-line models, where losses and flow deviations are estimated by means of empirical correlations (Craig and Cox 1970; Aungier 2006; Dixon 2005). Mean-line models can be integrated with the optimization of ORC power systems, but the computational time increases significantly as the number of stages increases (La Seta et al. 2016). To overcome this limitation, Astolfi and Macchi (2015) developed efficiency correlations to estimate the design-point efficiency of ORC axial-flow turbines with one, two, and three stages. The correlations were based on the results from Axtur considering the working fluid as an ideal gas with a ratio of specific heat capacities γ = 1.05. The efficiency was reported for optimal specific speed as a function of the size parameter and isentropic volumetric ratio. The correlations showed a good agreement with the mean-line model (maximum absolute deviations below 0.5%) and can be easily integrated with the optimization of ORC power systems, as the authors have done (Pili et al. 2019).

Depending on the dynamics and fluctuations of the heat and cold sources, the ORC expander can be operated close to the design point or be affected by part-load operation, which results in lower efficiency with respect to the nominal point (Pili et al. 2017; Jiménez-Arreola et al. 2018). Since the conversion from thermal to mechanical energy occurs in the expander, the efficiency of this component is crucial for the overall performance of the ORC power system. The goal of this paper is to extend the work of Astolfi and Macchi (2015) and develop correlations to estimate the efficiency of axial-flow turbines at off-design conditions based on the change in specific enthalpy across the turbine and the exhaust volume flow rate. Section 2 introduces the mean-line design and off-design models, including their validation against data and measurements available in literature, and Section 3 presents the eight reference cases used to develop the correlations. These reference cases are representative of different ORC fields of application and were selected to cover a wide range of operating conditions. Finally, in Section 4, the part-load efficiency correlations that generalize the results from the eight reference cases are reported and tested against a benchmark case not included in the curve fitting. The correlations developed in this work can be used in ORC system analyses to estimate the turbine performance at part-load conditions prior to its design.

2. MEAN-LINE DESIGN AND OFF-DESIGN OF AXIAL-FLOW TURBINES

2.1 Design optimization model

The mean-line model AxialOpt (Agromayor and Nord 2019a) was used to design the axial-flow turbines analyzed in this work. In order to formulate the turbine design as an optimization problem it is necessary to define the: 1) objective function, 2) fixed parameters used as input, 3) independent variables, and 4) constraints that limit the design space. In this work, the optimization was carried out by maximizing the isentropic efficiency given by Eq. (1). This objective function follows the definition given by Macchi and Astolfi (2017), where the subscripts “in” and “out” refer to the inlet and outlet of the turbine, respectively, and ϕ_E is the efficiency of the diffuser (= 0.5).

$$\eta = \frac{h_{0,\text{in}} - h_{\text{out}}}{h_{0,\text{in}} - h_{\text{out,s}} - \phi_{\text{E}} \dfrac{v_{\text{out,a}}^{2}}{2}}. \tag{1}$$

To evaluate the turbine model, the working fluid, mass flow rate, stagnation temperature and pressure at inlet and static pressure at outlet are given as fixed parameters and the variables summarized in Table 1 are given as independent variables for the optimization. Note that number of independent variables depends on the number of turbine stages n_{st} and is given by (3 + 13 n_{st}). In order to ensure that the design is realistic, the independent variables are constrained with the bounds reported in Table 1 and the design space is further limited with the nonlinear inequality constraints summarized in Table 2. Moreover, the nonlinear equality constraints from Table 2 are imposed to ensure that 1) the static pressure at the outlet of the turbine matches with the design specification and 2) that the entropy at the outlet of each cascade is consistent with the loss model.

Once the fixed parameters and optimization variables are specified, the flow variables are calculated sequentially from the inlet to the exit of the turbine using the principles of conservation of mass and rothalpy, the velocity triangle relations, and the equations of state of the REFPROP fluid library to compute the thermodynamic properties of the working fluid (Lemmon et al. 2017). In addition, the flow deviation at the outlet of each cascade is computed using the correlations by Ainley and Mathieson (1951) for subsonic flows and with the correlations by Vavra (1969) and Deich et al. (1965) for super-

Table 1: Optimization variables and bounds used in AxialOpt

Optimization variables	Lower Bounds	Upper Bounds
Global		
Specific speed, -	0.01	5.00
Specific diameter, -	0.01	5.00
Reduced inlet absolute velocity, -	0.01	0.95
For each cascade		
Reduced outlet relative velocity, -	0.01	0.95
Relative outlet flow angle (stator/rotor), °	+40/-83	+83/-40
Pitch to chord ratio, -	0.30	1.00
Blade aspect ratio, -	0.80	5.00
Ratio of stator-outlet to turbine-inlet entropy,	0.99	corresponding to η = 25%
Incidence angle, °	-40	+40
Group 2 losses (only rotor), -	0	1

sonic flows with converging and converging-diverging cascades, respectively. The losses within the turbine were estimated with the model from Craig and Cox (1970) using the enthalpy loss coefficient definition, given by Eq. (2). This is consistent with the loss coefficient definition used to develop the Craig and Cox method. Note that in this case, the subscript "out" refers to the outlet of each cascade.

$$Y = \frac{h_{\text{out}} - h_{\text{out,s}}}{w_{\text{out}}^2 / 2} \tag{2}$$

The constrained optimization problem was solved using the Sequential Quadratic Programming (SQP) algorithm of the of the MATLAB® Optimization Toolbox and a multi-start strategy with 400 different starting points was adopted to avoid solutions converging to local optima.

A comparison between AxialOpt and Axtur for the case of two single-stage turbines using R125 and hexane as working fluid is depicted in Table 3. It can be seen that the discrepancies between the isentropic efficiencies of the optimized turbines are below 3%. For the turbine with Hexane, a larger mean diameter with lower rotational speed is designed in AxialOpt.

2.2 Off-design performance prediction model

The off-design calculation tool AxialOff is based on the results of the design optimization tool and was developed to predict the overall performance of existing turbine designs under part-load conditions. The off-design model requires: 1) the geometry of the optimized turbine configuration, 2) the inlet stagnation temperature and pressure, 3) the total-to-static pressure ratio, and 4) the rotational speed as input.

Table 2: Equality and inequality constraints used in AxialOpt

Constraints			
Equality			
Static outlet pressure error		Efficiency drop for group 2 losses in the CC model for rotor	
Loss coefficient error at each cascade		Incidence angle error for minimal incidence loss	
Inequality			
Max. relative Mach number at rotor inlet, -	0.8	Min./Max. axial chord to mean diameter ratio at rotor, -	0.001/0.200
Min./Max. blade height to mean diameter ratio at rotor, -	0.001/0.25	Min./Max degree of reaction of each stage, -	-0.1/0.9
Min./Max. flaring angle of any cascade, °	0/30	Min./Max. throat to spacing ratio at stator, -	0.0018/0.1000
Min. relative flow angle at the inlet of the stator/rotor cascade, °	+30/-30		

Table 3: Comparison of turbine design optimization between AxialOpt and results with Axtur

Working fluid →	**R125**			**Hexane**		
Quantity ↓	Axtur	AxialOpt	Difference, %	Axtur	AxialOpt	Difference, %
Inlet stag. temperature, °C	155.0	155.0	-	155.1	155.1	-
Inlet stag. pressure, bar	36.200	36.200	-	8.29	8.29	-
Outlet static pressure, kPa	15.685	15.685	-	0.250	0.250	-
Mass flow rate, kg/s	11.89	11.89	-	2.04	2.04	-
Volumetric ratio, -	2.293	2.312	0.8	34.39	35.31	2.7
Size parameter, m	0.036	0.036	0.8	0.089	0.090	1.1
Rotational speed, rpm	31 000	29 660	-4.3	28 000	24 044	-14.1
Mean diameter, m	0.086	0.086	0.5	0.180	0.233	29.4
Isentropic efficiency, %	87.2	87.1	-0.1	79.5	81.5	2.5

It then returns the turbine isentropic efficiency and mass flow rate as output. The off-design model is not explicit and it requires the solution of a system of nonlinear algebraic equations. This system of equations is formulated as an optimization problem by setting the objective function to zero and using the equality and inequality constraints summarized in Table 4. Note the inequality constraint according to which the turbine mass flow rate has to be lower or equal to the critical mass flow rate in each cascade (the constraint is active in cascades that are choked). The optimization problem was solved using the Sequential Quadratic Programming (SQP) algorithm of the of the MATLAB® Optimization Toolbox with a single-start approach and every operating point was calculated using the previous solution as initial guess to speed up convergence.

AxialOff has been validated against the experimental data from Kofskey and Nusbaum (1972). The main geometric parameters and inlet stagnation conditions for turbines with one and two stages were replicated and the performance of the turbine was computed for different pressure ratios and angular speeds. The considered cases range from 50 % to 110 % of the nominal rotational speed, and the results are shown in Figs. 1 and 2. The points refer to the measurements available in Kofskey and Nusbaum (1972), whereas the solid lines refer to the simulated cases. In the single-stage case, the maximum absolute relative error amounts to 2.4% for the equivalent mass flow rate and 4.3% for the equivalent torque, whereas for the two-stage turbine the maximum absolute relative errors are 3.2% and 5.1%, respectively.

3. CASE STUDIES

To characterize the part-load behaviour of ORC axial-flow turbines by means of efficiency correlations, eight different case studies available in literature are considered and summarized in Table 5. Different fields of application (biomass, geothermal, and waste heat recovery from industrial processes and maritime propulsion) are included in the analysis to consider a broad range of operating conditions, where the stagnation inlet temperature varies from 128 to 305 °C, the pressure ratios from 2 to 124, the isentropic power output from 250 kW to 2.5 MW and molecular mass of the working fluid between 72 kg/kmol and 237 kg/kmol.

Table 4: Equality and inequality constraints in AxialOff

Constraints	
Equality	
Static outlet pressure error	Efficiency drop for group 2 losses in the CC model for rotor
Loss coefficients error at each cascade	Mass flow rate error at each cascade outlet
Inequality	
Mass flow rate at cascade outlet has to be lower than critical mass flow rate	

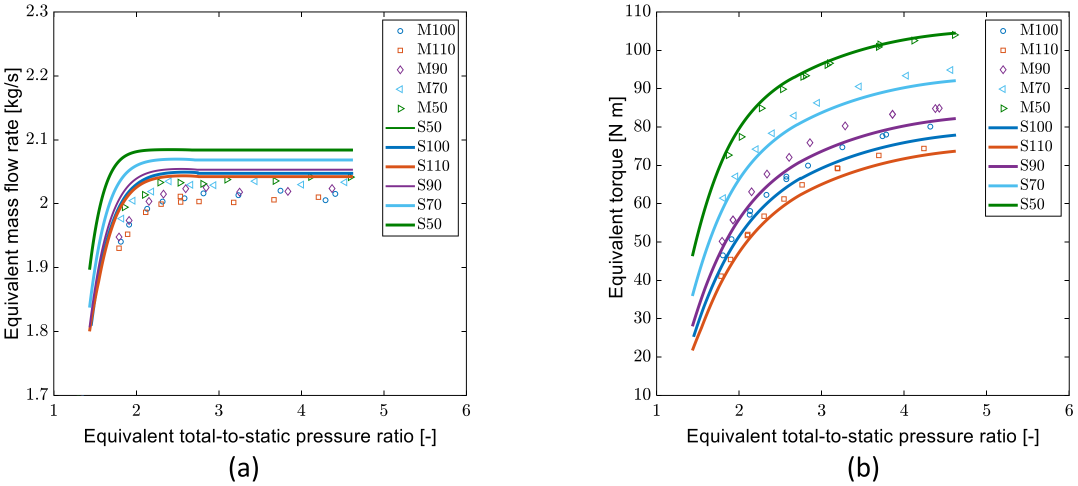

Figure 1. Comparison of AxialOff with the single-stage turbine in Kofskey and Nusbaum (1972).

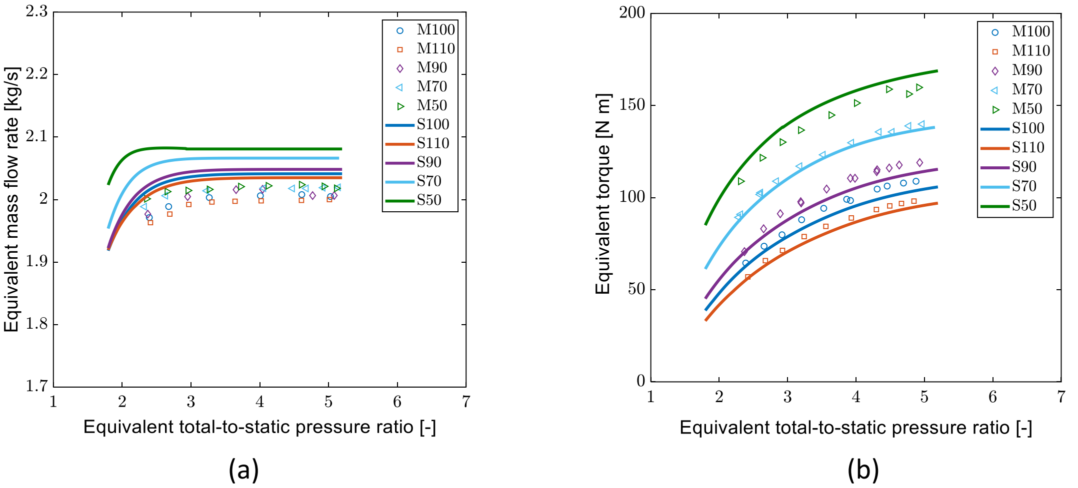

Figure 2. Comparison of AxialOff with two-stage turbine in Kofskey and Nusbaum (1972) .

4. RESULTS

The turbines listed in Table 5 have been designed with AxialOpt considering one, two and three stages. The isentropic efficiencies are ranging between 76 % and 90 %, increasing as the number of stages increases, see Table 6. The turbines designed with AxialOpt show a maximum relative difference in efficiency of 6% with respect to the predictions made by the correlation proposed by Macchi and Astolfi (2017). The highest discrepancies are found for single stage turbines and large volume ratios, where uncertainties in the loss models are higher because of supersonic flow. The efficiency of AxialOff at nominal point are also shown, which are very close to the design predictions as expected (< 2%).

Off-design maps have been developed with AxialOff, by keeping the turbine rotational speed constant as in usual stationary applications, where the rotational speed is synchronized with the grid frequency through a constant gear ratio between the turbine and the electric generator. The pressure ratio across the turbine is varied by increasing the static outlet pressure from nominal point. The nominal stagnation inlet temperature and pressure are kept constant, since their impact on the efficiency is minimal for a given pressure ratio, in agreement with similarity laws. The Trust-Region algorithm of the Curve Fitting Toolbox™ from MATLAB® is used to develop correlations to predict the isentropic efficiency η in Eq. 1 as a function of the difference in specific enthalpy over the turbine Δh and outlet volume flow rate $\dot{V}_{out}$. These two quantities are chosen since they are representative of the loading and flow rate across the turbine, and largely affect its performance. The diffuser is assumed to have constant efficiency at part-load ($\phi_E = 0.5$). All the values are normalized with respect to the design point ('D'):

Table 5: Design inputs for different case studies

No.	Application	Working fluid	Stagnation inlet temperature, °C	Stagnation inlet pressure, bar	Static outlet pressure, bar	Mass flow rate, kg/s
1[a]	Biomass	MDM	305.00	7.92	0.22	5.46
2[b]	Biomass	Toluene	292.02	21.90	0.41	13.69
3[c]	Geothermal	R1234yf	128.50	42.57	8.44	190.73
4[d]	WHR Cement	Pentane	162.00	19.40	1.03	16.67
5[e]	WHR Ship	Benzene	225.34	19.66	0.16	3.06
6[f]	WHR Steel	Toluene	290.85	5.21	0.15	11.74
7[g]	n/a	R125	155.00	36.20	15.69	11.89
8[g]	n/a	Hexane	155.10	8.29	0.25	2.04

References: [a](Colonna et al. 2008); [b](Martelli et al. 2015); [c](Manente et al. 2016); [d](Bavarian State Ministry of Environment 2001); [e](Mondejar et al. 2017); [f](Pili et al. 2019); [g] (Macchi and Astolfi 2017).

$$\frac{\eta}{\eta_\mathrm{D}} = a + b\left(\frac{\Delta h}{\Delta h_\mathrm{D}}\right) + c\left(\frac{\Delta h}{\Delta h_\mathrm{D}}\right)^2 + d\left(\frac{\dot{V}_\mathrm{out}}{\dot{V}_\mathrm{out,\,D}}\right) + e\left(\frac{\dot{V}_\mathrm{out}}{\dot{V}_\mathrm{out,\,D}}\right)^2 + f\left(\frac{\Delta h}{\Delta h_\mathrm{D}}\right)\left(\frac{\dot{V}_\mathrm{out}}{\dot{V}_\mathrm{out,\,D}}\right) \tag{3}$$

The coefficients resulting from the fitting procedure and the corresponding coefficient of determination R^2 are given in Table 7. The good agreement between the performance maps in AxialOff and the correlations is confirmed by the high value of R^2 (> 99%).

Table 6: Results of design for case studies

No.	Working fluid	Isentr. volume ratio, -	Isentr. size parameter, m	Isentropic efficiency, %								
				AxialOpt			Axtur (diff, %)			AxialOff (diff, %)		
				stages			stages			stages		
				1	2	3	1	2	3	1	2	3
1	MDM	41.91	0.13	82.3	85.6	86.9	-1.8	-0.8	0.1	-1.3	0.0	0.0
2	Toluene	58.74	0.18	84.7	86.3	87.5	-5.2	-1.9	-1.0	-0.6	0.0	0.0
3	R1234yf	6.14	0.16	88.4	78.4	79.1	-1.3	-0.2	0.6	0.0	0.0	0.0
4	Pentane	23.17	0.14	82.4	87.1	88.7	1.1	-2.0	-0.8	0.9	0.0	0.0
5	Benzene	112.15	0.12	76.3	87.7	88.3	-5.5	3.4	5.1	-1.6	-0.6	-0.4
6	Toluene	31.82	0.29	82.7	85.5	86.6	0.7	-0.2	0.2	-1.1	0.0	0.0
7	R125	2.29	0.04	87.1	87.7	88.3	0.7	0.9	1.0	0.0	0.0	0.0
8	Hexane	34.35	0.09	81.5	85.0	86.0	-2.0	-1.5	-0.4	-2.0	0.0	0.0

The validity of the correlations is tested on an additional turbine working with R245fa that was not considered during the curve fitting. The design parameters of this turbine were retrieved from Meroni et al. (2016a). The turbine has an isentropic volume ratio of 2.83 and a size parameter of 0.082 m at nominal point. The isentropic efficiency at part-load for single-, two- and three-stage turbines is shown with solid lines in Fig. 3. It can be seen that the cases with higher number of stages have a better efficiency than the cases with less stages in the entire depicted range.

The prediction using Eq. 3 is illustrated with dashed line. The part-load of the turbine is predicted with

Table 7: Regression coefficients of the off-design efficiency correlations

Coefficients	Number of stages, -		
	1	2	3
a	0.245	0.418	0.548
b	1.632	1.066	0.937
c	-1.940	-0.568	-0.609
d	0.033	0.057	-0.056
e	-1.085	0.000	-0.043
f	2.112	0.035	0.228
R^2	0.994	0.993	0.992

a coefficient of determination R^2 equal to 96.6%, 90.0% and 94.9% for respectively single-, two- and three-stage turbine. It can be seen that the discrepancy between the performance map with AxialOff and the efficiency correlations increases as the operating point moves further from nominal conditions at pressure ratio 2.78, but the qualitative trend is respected also low pressure ratios.

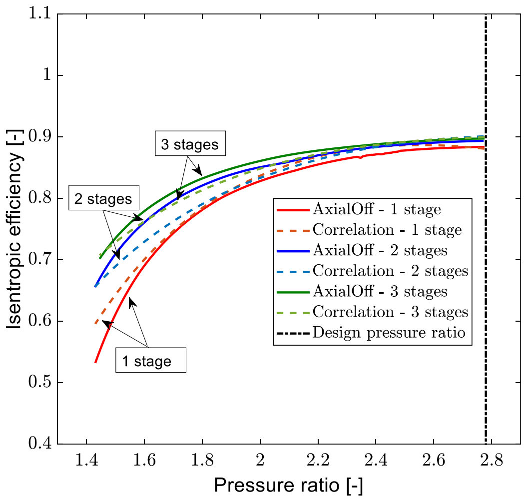

Figure 3. Comparison of performance map of R245fa turbine between AxialOff and part-load correlations.

5. CONCLUSIONS

Organic Rankine Cycle (ORC) systems are affected by part-load operation when the hot and cold source conditions differ from the design point. To assess the penalty caused by part-load operation and develop off-design strategies, it is necessary to predict the performance of the turbine under different operating conditions. Two MATLAB® mean-line tools for axial turbines with any number of stages were presented: one for design optimization, AxialOpt, and one for off-design performance prediction, AxialOff. The models were validated against data from the literature, with maximum discrepancy of 5%. Eight turbines from a broad range of applications and working conditions have been designed and performance maps have been developed. With these, efficiency correlations have been defined for the prediction of the turbine part-load performance based on the off-design-to-design ratio of the specific enthalpy change across the turbine and the outlet volumetric flow rate. The correlations have been tested on a benchmark case not included in the curve fitting, showing a coefficient of determination R^2 not lower than 90% for single-, two- and three-stages, showing a good agreement with the results from AxialOff. As part of future work, a broader range of turbines will be analyzed, and a further validation with measurements from operating ORC axial-flow turbines will be carried out.

NOMENCLATURE

h	specific enthalpy	(J/kg)	w	relative velocity	(m/s)
n_{st}	number of stages	(-)	Y	loss coefficient	(-)
R^2	coefficient of determination	(-)	ϕ_E	efficiency of diffuser	(-)
v	absolute velocity	(m/s)	η	isentropic efficiency	(-)
$\dot{V}$	volume flow rate	(m^3/s)			

Subscripts

0	stagnation	out	outlet
a	axial	s	isentropic
in	inlet		

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