EFFECTS OF DESIGN CONDITIONS AND FLOW LAYOUT ON THE DESIGN OF ULTRA-LOW ASPECT RATIO RADIAL OUTFLOW TURBINES

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ABSTRACT

Small-scale radial outflow turbines offer a promising alternative to conventional axial turbines, for example in waste heat recovery applications. One of the design challenges with these machines is that the common single flow design approach leads to the need for axial thrust compensation. A convenient solution to this challenge is to use a double flow layout, which, however, reduces the turbine efficiency due to increased aerodynamic losses. The public literature lacks information on the design of these turbines under different design conditions and power levels. Therefore, this study aims to fill this research gap by examining three design cases as both single and double flow turbines. It hereby shows that the aspect ratio and tip-clearance losses mostly explain the observed differences between the flow layouts. The role of high flow turning in the rotor losses is also highlighted under different design conditions.

KEYWORDS

STEAM TURBINE, RADIAL OUTFLOW TURBINE, BIOMASS, WASTE HEAT RECOVERY

NOMENCLATURE

Latin alphabet

c_x	axial chord	[m]
H	blade height	[m]
Re	Reynolds number	[-]

Greek alphabet

ε	deflection	[°]
ζ_s	stator loss	[-]
ζ_r	rotor loss	[-]
ξ	turning loss	[-]

Abbreviations

DF	double flow
FEM	finite element method
SF	single flow

INTRODUCTION

There are various applications in the energy sector in which small-scale steam turbines have good potential as energy conversion devices. These applications include, for example, large ships, where Lion et al. (2020) found an approximately 5% improvement in engine efficiency when a steam turbine was used to recover waste heat. In addition, Uusitalo et al. (2019) predicted up to 1.5% improvement in cruise ship energy production efficiency using a small-scale Rankine process. The utilization of sawmill by-products also offers possibilities for these turbines, as discussed by

Leino et al. (2016). Later, Grönman et al. (2020a) discussed that in some biomass power plants, the possibility to produce electricity during periods when steam is not needed by consumers offers a good opportunity for small turbines.

Typically, small-scale steam turbines are axial impulse designs, which can have poor design and off-design performance and a relatively large physical size. High-speed technology based small-scale radial outflow turbines offer an alternative with the potential to overcome the abovementioned challenges. There are several reasons for the expected improved performance; for example, Persico et al. (2013) mentioned that radial outflow turbines can have lower Mach numbers than axial ones and are also capable of handling higher volume flows. Further, these turbines can help avoid partial admission losses. Some of the improved off-design performance may be explained by the small predicted influence of the incidence on the blade outlet flow angle distribution, as presented by Grönman et al. (2017).

Radial outflow turbines are typically single flow designs (see, e.g., Song et al. (2017), Doğu et al. (2018) and Casati et al. (2014)). The typical flow layout leads to the need for axial thrust compensation, which can complicate the mechanical design. This challenge can, however, be solved by using a double flow turbine design. These turbines are generally characterized by ultra-low (below unity) aspect ratios, whereby the secondary losses increase rapidly and lead to decreased aerodynamic performance. The key reason for the increased secondary losses is the interaction between the hub- and shroud-side secondary structures, as shown, for example, by Perdichizzi and Dossena (1992) and Grönman et al. (2020b). In addition, the relative tip-clearance is often increased in double flow turbines, with expected higher losses.

The required turbine power and design operating conditions vary between different applications, and this can affect the aerodynamic design. The turbine design may also be affected by, for example, the rotational speed limitations due to critical frequencies or high rotor stress levels. In addition, the minimum allowable blade height of 2 mm, following Grönman et al. (2020a), and the blade passage opening angle limit of 30 degrees, as suggested by Persico et al. (2013), can pose a challenge.

The one-dimensional design path of radial outflow turbines typically uses axial turbine loss correlations. Both Pini et al. (2013) and Casati et al. (2014) employed the correlation of Craig and Cox (1971) for their genetic algorithm-based design process for a single flow turbine. Meanwhile, Al Jubori et al. (2017) employed the loss correlation of Kacker and Okapuu (1981) in their single flow design. The accuracy of the axial turbine correlations has been shown to be at an acceptable level by Persico et al. (2013), with a slight over-prediction of primary losses; later, Grönman et al. (2017) presented partially similar results. A study by Grönman et al. (2020a) also showed that the loss model of Soderberg, as presented in Dixon (2005), could predict the loss trend at ultra-low aspect ratios reasonably well. There are also studies on single flow designs, such as Luo et al. (2018) and Song et al. (2017), that do not explain how the turbine losses are calculated, although the design process generally follows the axial turbine methodology.

The presented background shows that although there are studies related to radial outflow turbine design methodology, the specific design challenges of the double flow turbine have not been widely examined. There is also limited information available about the effects of turbine design conditions on the resulting geometries and performance levels. This lack of scientific information motivates this work to study how the required power level, flow layout, and varying design conditions affect the design of radial outflow steam turbines. The study employs a 1/1.5-dimensional design approach to examine the different designs and compares the resulting turbines with the others available.

The article is organized as follows: It begins with the presentation of turbine design and design methods. In the following subsection, the studied cases are explained and justified. The results are presented and analysed in the penultimate section prior to the conclusions.

TURBINE DESIGN AND MODELLING

Turbine Layout

The turbines studied here are based on a hermetic high-speed turbogenerator design, whereby the turbine and electric generator are on the same shaft and no gear box is used. The turbogenerator has magnetic bearings and the rotor shaft and the generator stator are water cooled. The general layout is presented in Fig. 1. The examined turbine is a double flow design, with a symmetric turbine rotor designed to compensate for the axial thrust. The presented layout was mechanically tested and the concept was verified by Grönman et al. (2020a). To demonstrate the specific design challenges and benefits arising from utilizing a double flow turbine instead of the commonly used single flow machine, single flow turbines are also designed for the same radial outflow turbine layout.

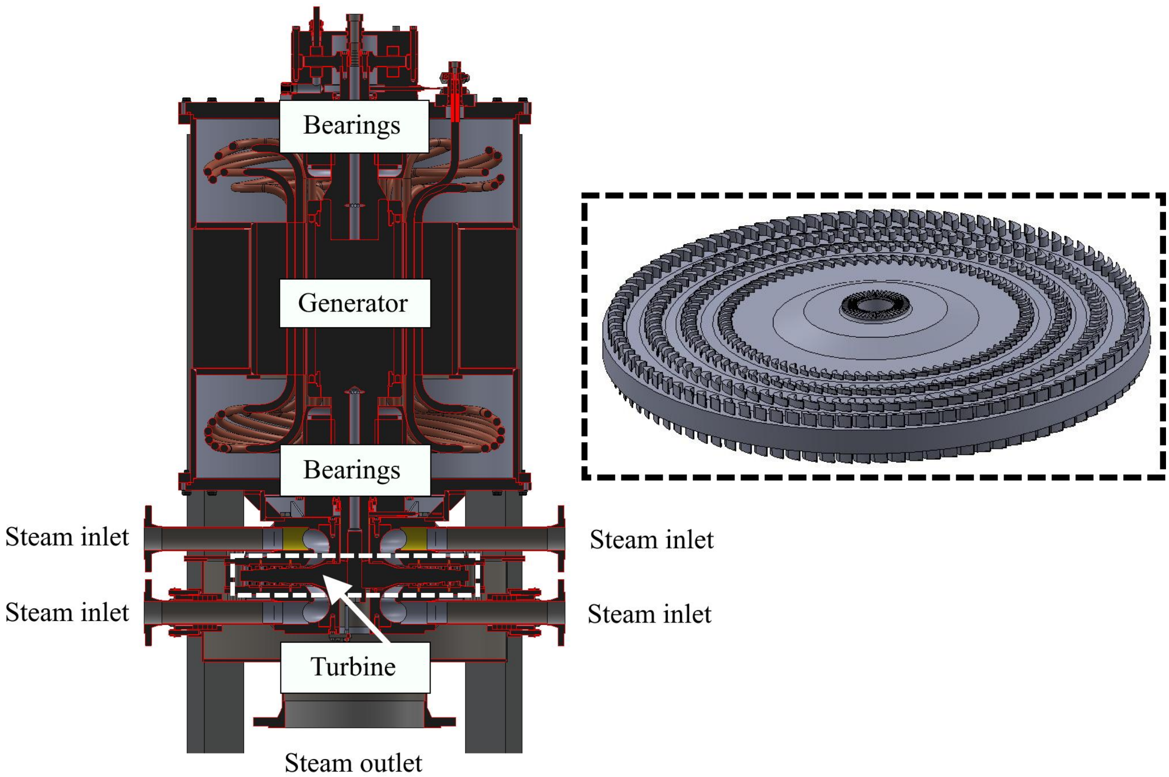


Figure 1: Schematic layout of the double flow radial outflow turbine and the turbine rotor wheel.

Design Methodology and Turbine Loss Model Accuracy

The design of high-speed radial outflow turbines typically has a rotational speed limit that cannot be exceeded due to the material strength and rotor critical speed limits if a sub-critical design is desired. Therefore, the design is often intended to maintain the rotational speed as high as possible since a higher speed allows the use of greater blade heights in the first stages. This approach leads to higher aspect ratios and decreases secondary losses. In addition, the tip-clearance losses are often decreased due to smaller relative clearances. In this study, all examined cases have rotational speeds that are chosen based on the limits from the FEM (Finite Element Method) analysis of the whole rotor.

The design is an iterative process, whereby the turbine inlet temperature and pressure and the outlet pressure (all static values) are given as input data. In addition, the mass flow and rotational speed are fixed. The fixed rotational speed is justified since higher values are typically beneficial from the performance point of view, but the mechanical constraints limit the speed as previously discussed. The calculation follows mostly the typical axial turbine design path by using radial velocity instead of axial velocity. The general design principles were presented by Grönman et al.

(2020a). During the iteration, the inlet radius of each rotor, turbine stage efficiency, and radial velocity are solved for each stage in the flow direction according to the design constraints. The last stage degree of reaction can change freely so that the exit pressure is automatically reached. Typically, a higher degree of reaction than the constant 0.25 of the other stages is aimed to increase the last stage efficiency. The blade height and the passage opening angle are adjusted by modifying the turbine inlet radius and radial velocity. The passage opening angle should not exceed 30 degrees according to Persico et al. (2013). In addition, the number of stages can be adjusted between one and six stages. The decision about the number of stages is usually made based on avoiding supersonic Mach numbers and non-smooth passage opening. From a manufacturing point of view, it is, for example, of paramount importance that the turbine inlet radius has enough radial clearance between the shaft and the first stage stator to allow the flow to turn smoothly for the first stage vanes. The following assumptions and constraints are set for the design:

1. The flow is fully radial at the inlet and outlet of each stage.
2. The incidence is zero at each blade row.
3. The last stage degree of reaction changes freely and in other stages it is maintained at 0.25.
4. A tip-clearance of 0.5 mm is used.
5. The radial clearance between successive blade rows is 25-35% of the radial chord.
6. The blade passage opening angle is below 30 degrees.
7. The minimum vane/blade height is 2 mm.

The properties of the steam are calculated using the Refprop database (Lemmon et al. (2010)). The optimal pitch-chord ratio is solved using Traupel's (1977) method, which uses the vane/blade inlet and outlet angles as input values. The axial thrust of the rotor is estimated by applying Nguyen-Schäfer's (2015) method and by neglecting the effects of impulse and back disk due to their negligibly low values. Therefore, the thrust is calculated only based on the inlet and shroud forces. The turbine loss calculation is made using Soderberg's correlation (Dixon (2005)), which is a simple method and is considered here as it was developed during a time when turbine aspect ratios were relatively low compared to the majority of today's designs; furthermore, Grönman et al. (2020a) demonstrated its promising performance. Soderberg's correlation defines the stator loss coefficient as follows:

$$\zeta_s = \left(\frac{10^5}{Re}\right)^{1/4}\left[(1+\xi)\left(0.993 + 0.021\,\frac{c_x}{H}\right) - 1\right], \tag{1}$$

and the rotor loss is calculated as

$$\zeta_r = \left(\frac{10^5}{Re}\right)^{1/4}\left[(1+\xi)\left(0.975 + 0.075\,\frac{c_x}{H}\right) - 1\right], \tag{2}$$

where the turning loss is defined as a function of deflection

$$\xi = 0.04 + 0.06\left(\frac{\epsilon}{100}\right)^2. \tag{3}$$

The loss model can be interpreted so that the secondary loss is affected by the aspect ratio (H/c_x) and the primary loss by the blade deflection ε. It also assumes that the Reynolds number affects both the primary and secondary losses. The Reynolds number is defined based on the vane/blade throat hydraulic diameter, which differs from the commonly used vane/blade chord. The correlation does not include tip-clearance effects, whereby Dixon (2005) recommends that these are calculated by multiplying the stage efficiency without the tip-clearance loss by the ratio between the blade area and the blade area + clearance area.

A study by Zhdanov et al. (2013) compared Soderberg's loss correlation with other commonly used correlations. Based on their work, Soderberg's correlation seems to predict the turbine efficiency so that most of the predictions fall inside the error of ±3.0%. This difference is approximately twice as high than for the other compared correlations. It should, however, be noted that the investigated turbines had varying aspect ratios that were mostly outside the ultra-low aspect ratio range, and thus they do not represent results that are directly comparable with the current work.

Figure 2 presents a comparison between Benner's (Benner et al. (2006a) and Benner et al. (2006b)) and Soderberg's loss models and the numerical results of Grönman et al. (2017) with aspect ratios comparable to those used in the current study. These results give a strong indication that Soderberg's correlation better follows the trend of the numerical predictions than Benner's model. The presented information provides a good foundation to use the chosen loss correlation in further designs.

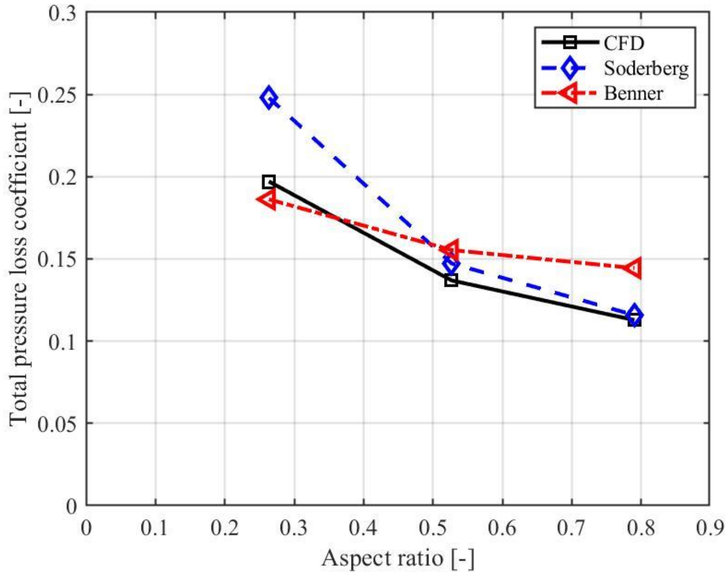


Figure 2: Comparison between the predicted cascade loss coefficients from Grönman et al. (2020a) and Grönman et al. (2017).

Studied Cases

Three different radial outflow turbine design cases are examined with different pressure and temperature levels, as presented in Table 1. The first case is a low-temperature back-pressure turbine that has been designed for biomass power plants during low load conditions or for utilizing excess steam in marine applications. The low temperature back-pressure design means that the expansion ends at atmospheric pressure, no condenser is used, and the inlet temperature is the lowest from the studied ones. The second case is a condensing turbine that has operating conditions that represent steam conditions in a power plant that utilizes waste as a fuel with an overall power plant efficiency target of 20%. The third case is a small-scale turbine that has design conditions fulfilling the requirement of 15% power plant efficiency. All cases have rotational speeds that are determined based on the maximum allowed values derived from the whole rotor FEM analysis.

Table 1: Studied design conditions for single and double flow turbines.

Case	Inlet mass flow	Inlet pressure	Inlet temperature	Outlet pressure	Rotational speed
1DF/SF	2.44 kg/s	10 bar	330 °C	1.01 bar	12 500 rpm
2DF/SF	1.80 kg/s	20 bar	405 °C	0.47 bar	12 500 rpm
3DF/SF	0.30 kg/s	20 bar	416 °C	0.47 bar	45 000 rpm

RESULTS

This section presents the results by comparing the performance of different designs. The loss sources of each turbine are further examined to highlight the differences resulting from the flow arrangement, design conditions and power levels.

Turbine Performance and Axial Thrust

The double flow layout influences the performance of the turbine, as seen in Fig. 3. The efficiency level is clearly below the level found in the literature. However, by changing to a single flow design, the performance level grows closer to typical values. The reasons for this behaviour are discussed further in the later sections, but, in general, the biggest differences between the single and double flow turbines are the aspect ratio and the relative tip-clearance, which both have a negative influence on the double flow turbine performance. In terms of power penalty, the back-pressure design of Case 1 loses approximately 84 kW, and the condensing turbine of Case 2 loses 105 kW, whereas in Case 3 a total of 8 kW is predicted to be lost from the turbine power due to the turbine flow arrangement. It is also worth noting that the typical Reynolds number effects, as seen in the Figure 3, show generally decreasing performance when moving towards lower turbine power levels (below 200 kW).

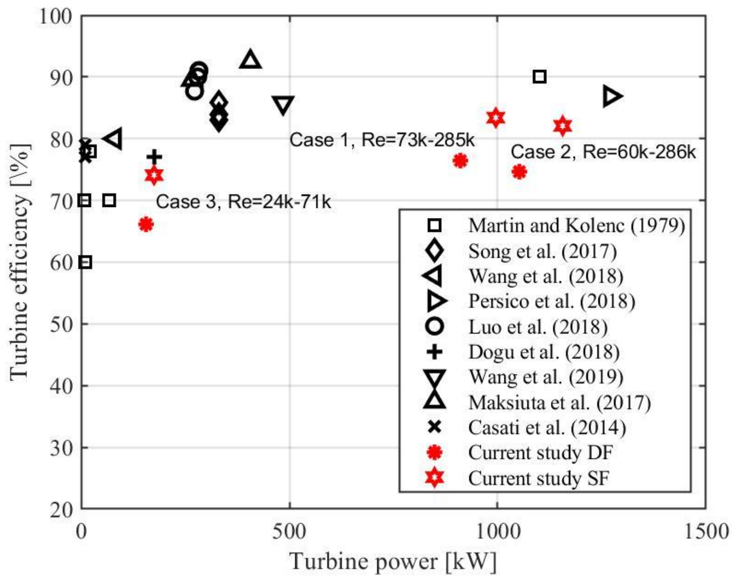


Figure 3: Comparison between the predicted turbine performance in this study and turbines in the literature. Also, the variation of vane/blade Reynolds numbers is presented for each case.

The compensated axial thrust changes significantly between the designs as functions of pressure difference over the rotor (is equal to the value over the whole turbine) and turbine inlet and outlet diameters, as seen in Tables 1 and 2. The effect of pressure difference is demonstrated when comparing Case 1 and Case 2, since they have reasonably similar disk diameters, but considerably different pressure differences over the rotor. Whereas a comparison between Case 2 and Case 3 demonstrates the effect of disk diameter since the turbine pressure differences between the two cases are equal.

Since the axial thrust compensation would require additional mass to be added to the rotor shaft, the rotational speed would most likely have to be decreased with the single flow designs. This design change would be required due to expected lower critical speed of the modified shaft. Therefore, further problems with the low turbine blade heights (and aspect ratios) may occur, which makes the double flow arrangement an even more attractive choice.

Table 2: Estimated axial thrust for different single flow turbines.

Case	Axial thrust [kN]	Inlet diameter [m]	Outlet diameter [m]
1SF	72.5	0.263	0.546
2SF	172.3	0.285	0.636
3SF	21.1	0.070	0.199

Comparison Between Different Turbine Geometries

As a result of the design process, the turbine of Case 2 has five stages and those of Cases 1 and 3 have four stages. Figs. 4 (a) and (b) illustrate the side profiles of the double and single flow turbine designs of Case 1. It can be observed that the inlet radius is slightly smaller with the double flow design due to the smaller mass flow and the minimum blade height requirement. This limitation also leads to differences in the outlet radius. The aspect ratios increase considerably while moving from a double flow to a single flow turbine, as is also shown in Table 3. The increase of the aspect ratio is a result of the mass flow change, changing radius, improved stage efficiencies and varying radial chords. It is clear that the aspect ratios are mostly ultra-low with double flow designs, but the single flow values are slightly more often above unity.

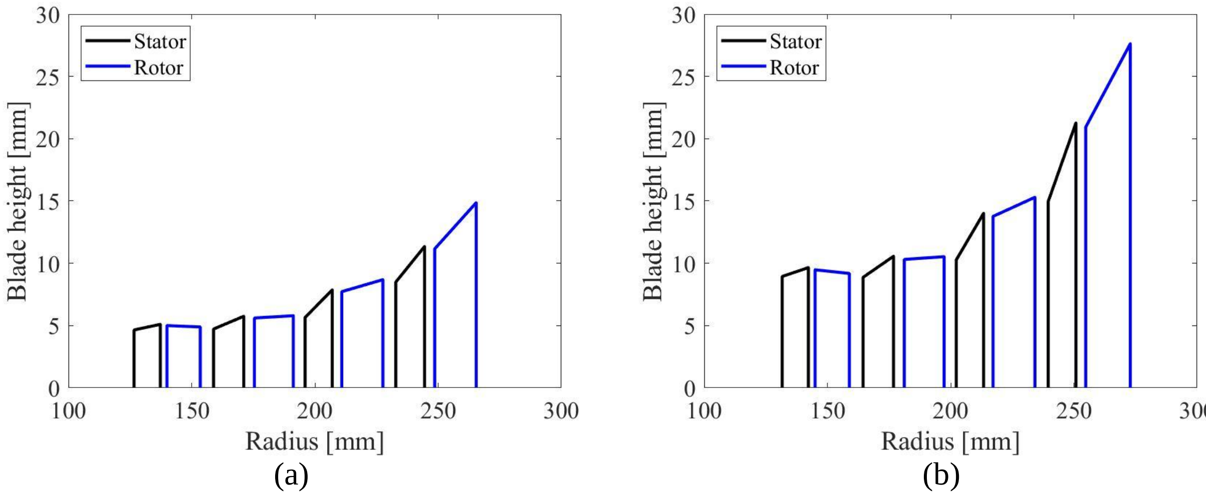


Figure 4: Comparison between the Case 1 double flow (a) and single flow designs (b).

Table 3: Vane and blade aspect ratios for double flow and single flow turbines.

	Stage 1		Stage 2		Stage 3		Stage 4		Stage 5	
Case	Vane	Blade	Vane	Blade	Vane	Blade	Vane	Blade	Vane	Blade
1DF	0.45	0.37	0.43	0.36	0.61	0.49	0.85	0.77		
1SF	0.87	0.67	0.79	0.65	1.10	0.86	1.61	1.34		
2DF	0.27	0.24	0.32	0.27	0.43	0.31	0.58	0.46	1.30	1.09
2SF	0.51	0.43	0.57	0.45	0.73	0.50	0.98	0.75	1.48	1.11
3DF	0.36	0.64	0.37	0.61	0.56	0.80	0.86	0.55		
3SF	0.69	1.17	0.64	0.97	0.83	1.12	1.57	0.83		

The comparison between the designs presented in Table 4 shows that the degree of reaction typically increases towards the last stage, which generally leads to higher efficiency. Since the power of the last stage is typically the highest, this approach is considered beneficial from the perspective of whole turbine performance. The lower reaction in the first stage also allows the blade heights to be kept at acceptable levels, and even lower values would be more likely to reach supersonic conditions. In a broader perspective encompassing the existing literature, the most common first stage degree of reaction is approximately between 0.2 and 0.3, which corresponds well with the current designs. Also, the maximum values are close to the typical values in the literature.

Table 4: Stage degree of reaction for single and double flow turbines.

Case	Stage 1	Stage 2	Stage 3	Stage 4	Stage 5	Flow type
1DF	0.25	0.25	0.25	0.39		Double
1SF	0.25	0.25	0.25	0.33		Single
2DF	0.25	0.25	0.25	0.25	0.37	Double
2SF	0.25	0.25	0.25	0.25	0.38	Single
3DF	0.25	0.25	0.25	0.47		Double
3SF	0.25	0.25	0.25	0.27		Single
Casati et al. (2014)	0.31	0.36	0.39	0.40	0.40	Single
Luo et al. (2018)	0.30	0.49	0.58	0.62		Single
Li and Huang (2017)	0.32	0.45	0.47			Single

The turbine operating conditions affect the vane outlet flow angles and their distribution between the stages, as seen in Fig. 5 (a). The flow angles are in general lower when the inlet temperature and pressure are lower, and the expansion ends in atmospheric conditions. With Cases 2 and 3, the absolute flow angles easily reach over 80 degrees, which is not a desirable result due to possible challenges in vane design. One of the known benefits of radial outflow turbines is the low Mach number level. This specific feature seems to be affected by the design conditions, but the flow arrangement (between single and double flow) may also have an effect due to changes in the inlet and outlet radii and the constant rotational speed assumption. The same explanation is also given for the observed vane outlet flow angle differences. Fig. 5 (b) shows that, in general, the Mach numbers stay below unity. However, the risk of transonic or supersonic velocities increases when the expansion takes place at lower outlet pressures, as in Cases 2 and 3.

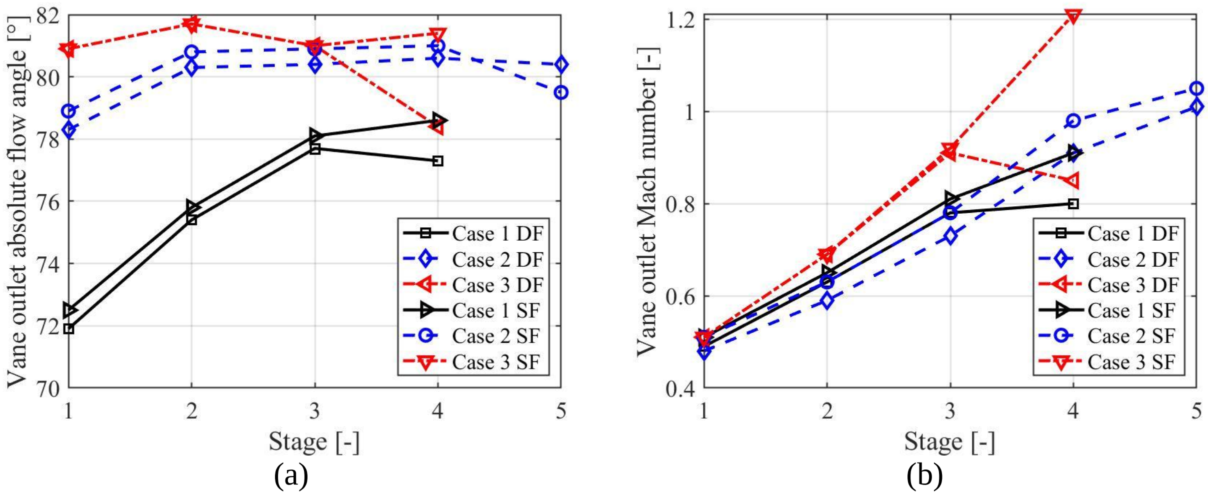


Figure 5: Vane outlet absolute flow angle (a) and vane outlet Mach number (b). Stage-to-stage comparison between single and double flow designs.

Turbine Loss Sources

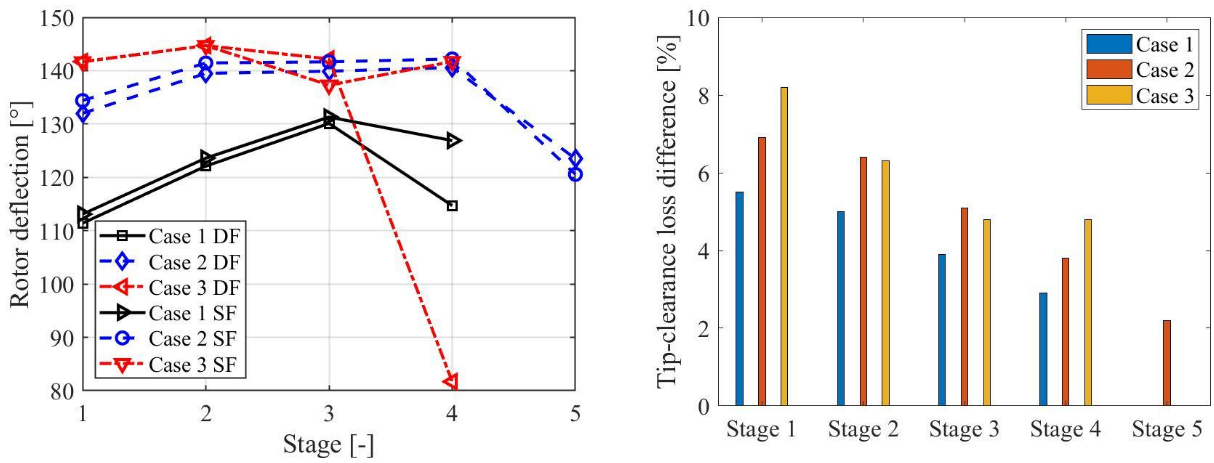
The losses according to Soderberg's model at each stage are presented in Table 5, showing that, in general, the vanes have less losses than the rotors. The reason for this finding is primarily connected to the higher blade deflection, although the aspect ratios are mostly lower for the rotors (as presented in Table 3), which causes additional losses. The aspect ratios become lower in Case 2 compared to Case 1, mainly due to increased turbine inlet pressure and temperature. At the same time, the rotor deflection increases (see Fig. 6 (a)); together, these explain the observed increasing trend in loss development.

In general, the vane and blade losses decrease when moving from double flow to single flow designs, as was expected based on the previous understanding. This phenomenon is due to the increasing aspect ratio (tip-clearance loss is not included); however, opposite to the observed trends, the turbine of Case 3 has higher rotor losses at the last stage rotor. This increase is due to significantly higher blade deflection, which is presented in Fig. 6 (a), and it can be explained by the fixed rotational speed and varying inlet and outlet radii. Similar behaviour is also found for Case 1, although the deflection difference is considerably lower, and the positive effect of the higher aspect ratio overcomes the negative deflection effect.

In overall loss breakdown, the tip-clearance related losses also play a significant role. The effects are more pronounced at lower turbine power levels or at increased inlet pressure and temperature levels due to smaller blade heights (and larger relative clearances), as highlighted in Fig. 6 (b). It can also be observed that the differences between the tip-clearance related losses become smaller between the single and double flow designs when moving towards the outlet of the turbine since the blade heights are closer to each other.

Table 5: Vane and blade loss coefficients for double flow and single flow turbines.

	Stage 1		Stage 2		Stage 3		Stage 4		Stage 5	
Case	Vane	Blade	Vane	Blade	Vane	Blade	Vane	Blade	Vane	Blade
1DF	0.10	0.29	0.11	0.27	0.10	0.28	0.09	0.22		
1SF	0.07	0.18	0.08	0.18	0.08	0.19	0.07	0.18		
2DF	0.13	0.43	0.13	0.36	0.12	0.40	0.11	0.33	0.10	0.27
2SF	0.09	0.27	0.09	0.25	0.09	0.27	0.09	0.23	0.08	0.19
3DF	0.16	0.56	0.16	0.51	0.15	0.46	0.13	0.28		
3SF	0.12	0.36	0.12	0.36	0.12	0.35	0.11	0.31		



(a) (b)

Figure 6: Rotor blade deflection (a), and tip-clearance loss difference between double flow and single flow turbines (b).

CONCLUSIONS

This study examined the effects of different operating conditions, turbine flow arrangements, and power levels on the design and performance of small-scale radial outflow steam turbines. The predicted performance was found to be comparable with other available single flow designs; however, the double flow turbine was found to deliver lower efficiency and power mainly due to larger relative tip-clearances and lower aspect ratios. The low Reynolds number effects also impacted the performance at the lowest studied power levels with additional losses.

The study also demonstrated the effects that the double flow layout and design conditions have on the velocity triangles and Mach number levels. It was illustrated that the flow layout affects the vane outlet flow angles, vane outlet Mach numbers and rotor deflection. This behaviour was found to be due to the changes in the inlet and outlet radii as well as the constant turbine rotational speed. The study also showed that higher inlet pressure and temperature and lower outlet pressure have an increasing effect on vane outlet flow angles and Mach numbers. This also increased the rotor deflections and decreased the aspect ratios, which led to higher aerodynamic losses.

In general, the examined designs showed relatively high rotor losses compared to vane losses due to high blade deflections and low aspect ratios. It was also highlighted that by accepting a performance penalty, the double flow design precludes the need for considerable axial thrust compensation, which keeps the whole machine design simpler.

It is suggested that future studies make more full-scale turbine operational data available. This would help with the development of the design process and accurate off-design performance evaluation methods.

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