ONE DIMENSIONAL DESIGN FOR CARBON DIOXIDE AXIAL FLOW TURBINE

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ABSTRACT

Carbon dioxide (CO_2) power cycle has many advantages for future energy utilization. Turbine is a vital component in this system, and its efficiency has a great impact on system performance. In general, the efficiency of the turbine expander is assumed to be a fixed value based on experiences when analysing the performance of carbon dioxide power cycle. However, the thermodynamic properties of working fluid have a great influence on turbine efficiency, so design parameters are critical to ensure a high efficiency of carbon dioxide turbine. On the other hand, most of the carbon dioxide turbines for MW-class are radial turbines at present whereas axial turbines take advantages in terms of fluidity and a simple structure compared with radial turbines. In this study, one dimensional design is carried out for a single-stage carbon dioxide axial-flow turbine by a mean-line design procedure, and the geometric parameters of the flow passage are designed and the turbine efficiency is obtained. The results can provide a reference for the practical design of turbine expander of carbon dioxide power cycle system

1. INTRODUCTION

Organic Rankine cycle and CO_2 power cycle systems are two main technologies to utilize industrial waste heat and renewable energy sources, such as solar energy, geothermal energy, for saving energy, protecting environment and enhancing thermal efficiency of industrial processes in the field of energy. CO_2 is an ideal circulating working fluid resulting from superior physicochemical properties, for example non-toxicity, cheapness, easily acquisition, abundance, non-flammability, stabilization, weak causticity to metal materials and moderate critical parameters (Brun et al.,2017). Off-design performance of a trans-critical CO_2 cycle system was studied for the utilization of geothermal energy, and the results indicated that there was an optimal value of geothermal resource mass flow rate to maximize the thermal efficiency (Li et al.,2018). Energy and exergy analysis are conducted for solar power plant with CO_2 power cycles, and the energy and exergy efficiencies were found to be 66.35% and 38.51%, respectively (Alzahrani and Dincer, 2018)

Expander as a key component, its performance affects the thermal efficiency of the system. In most cases, it is one of the most important components of CO_2 power cycle. Furthermore, selecting the expander is vital (Qiu et al.,2011) for the cycle power system where its performance depends on the working conditions, types of working fluid, available space, weight limit, range of output power, cost and noise (Wang et al.,2011; Bao and Zhang, 2013) . Usually, expander can be classified as scroll expander, rotary vane expander, screw expander, reciprocating piston expander , radial and axial flow turbine (Rahbar et al.,2017). Besides, in general, turbine has advantages in converting output work to electrical energy while reciprocating expander has advantages in directly coupling work to crankshaft due to its flexibility in operation (Qiu et al.,2011). Also, volumetric expanders are generally more bulkier and heavier than turbine (Latz et al.,2013). In addition, the lower rotating speed makes the output power of volumetric expanders lower, and the lubrication system which increases the complexity of the system is required in most applications for volumetric expanders (Wang et al.,2011). Turbine expanders are preferred due to their higher power outputs, efficiencies, simpler design and the non-requirement for a lubrication system. Turbine expanders are mainly classified into axial and radial turbines (Alshammari et al.2018). Axial turbines are commonly used at high mass flowrates and low pressure ratios. If the mass flow rates are low, small axial blades will be designed with relatively high tip clearance which lead to a significant drop in isentropic efficiency (Saravanamuttoo et al.,2009). For the axial turbine, the maximum isentropic efficiency of 73.4% was reached in the ORC test bed, and the micro-turbines could compete with volumetric expanders in the studied power range (Weiss et al.,2018). A designed super-critical CO_2 three-stage axial turbine was used to solar power generation system, and the results showed that the total-total efficiency was 91.6%, and this turbine could operate efficiently and steadily with the output power in the range of 16.2% to 155.9% (Shi et al.,2019).

Evaluating the efficiency of axial turbine is very important for preliminary and precise designing ORC cycles. Detailed CFD simulation is widely used to predict the performance of axial and radial turbines. However, three-dimensional simulation costs a lot of computing resources and needs long time period (Al Jubori et al.,2017). One-dimensional mean-line design procedure requires a smaller resource and a shorter time period than a three-dimensional simulation. According to similarity rules, a published paper provided an empirical efficiency correlations for axial turbine which is no more than three stages with organic working fluid by the mean-line procedure (Astolfi and Macchi,2015). However, the difference lies in thermodynamic properties between CO_2 and organic fluid, so this empirical efficiency relation is not so useful for turbine with CO_2 as working fluid. Therefore, in this paper, the preliminary design is conducted for a turbine using CO_2 as working fluid in order to obtain the turbine efficiency, rotational speed, velocity triangle at the designed operation condition. The mean diameter and geometric dimensions of flow passage are also obtained. Those parameters can be used for the detail design of the turbine. The obtained turbine efficiency can also be used as a reference for system analysis. Then, all of the total pressure losses are analyzed to pointing out the direction of next optimization for this turbine. At last, CO_2 and R245fa turbines are comparative analyzed to further explore the characteristic of turbine with CO_2 as working fluid.

2. PHYSICAL MODEL AND MATHEMATICAL METHOD

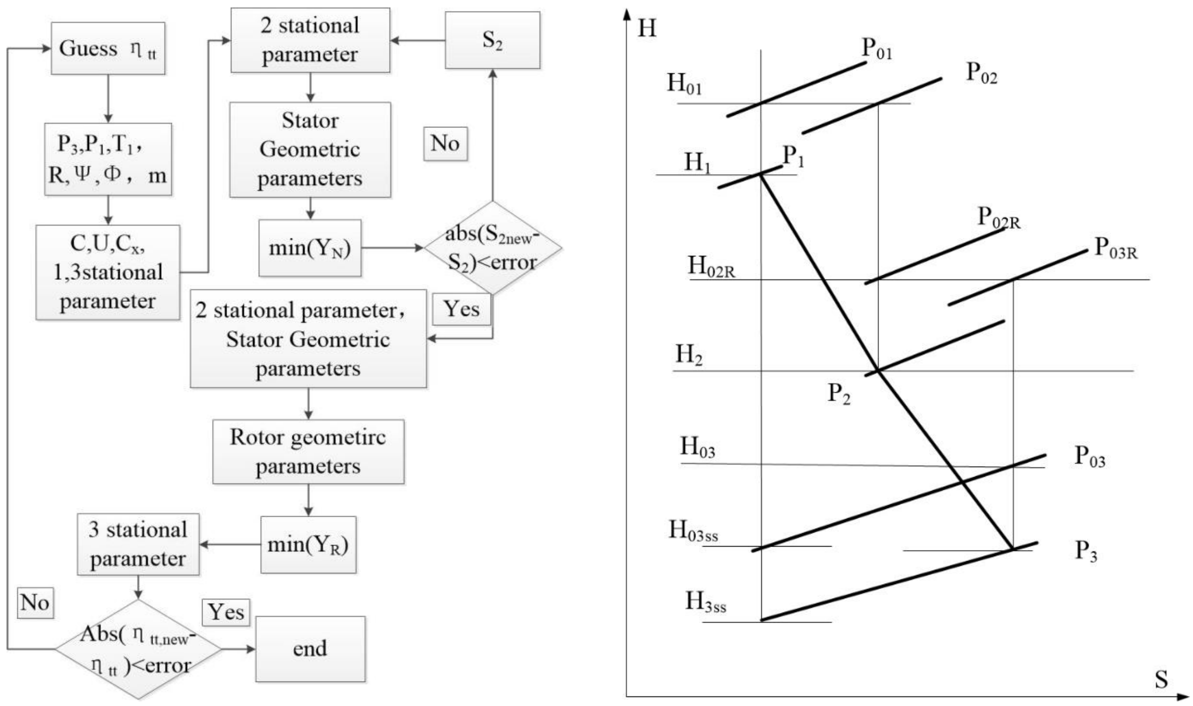

Figure 1: Model’s flow chart and enthalpy-entropy expansion process in a turbine stage

The turbine expenders used in organic Rankine cycle (ORC) power system or carbon dioxide (CO_2) power cycle system will be specially designed within operating conditions. Therefore, the power system will arrive in an optimum efficiency. So in this paper, a single-stage axial turbine used in trans-critical is designed by one dimensional mean-line method, and the vital parameters are presented such as the size of flow part, the mean diameter, the rotating speed and the efficiency. The stage within this turbine is assumed as a “normal ” stage (named also “repeating stage”) (Hall and Dixon,2013), which requires three conditions: $D_m = \sqrt{2(r_h^2 + r_t^2)}$, C_x= constant, $\alpha_1 = \alpha_3$.

Figure 1 shows the design procedures and the main optimization process. The optimal goal is the total pressure loss coefficient, and variables within the limits are the stator hub-to-tip radius ratios and the blade number of stator and rotor. When the minimum total pressure loss coefficient is calculated, the main parameters of this turbine and the efficiency in the designed operation condition are also calculated. The most widely used models to evaluate the total pressure loss are presented by Ainley and Mathieson (Ainley and Mathieson, 1951) and Dunham and Came (Dunham and Came,1970) , then refined by Kacker and Okapuu (Kacker and Okapuu, 1981). In this paper, the latest model derives from the method by Aunger (Aungier,2006),but with several modifications is applied. This applied model is more accruable to estimate the total pressure losses (Klonowicz et al., 2014) and the empirical equations is developed to apply the graphical models. Profile loss, secondary loss, trailing-edge loss, shock loss, supersonic expansion loss and blade clearance loss are involved for single stage axial turbine. The working fluid properties are taken from Refprop (Lemmon). The optimal calculation is implement by Matlab program, and the results is show in Table 2. The results of this paper are validated the published results (Da Lio et al.,2014).

3. RESULTS AND DISCUSSION

Firstly, we show the results of the designed CO_2 turbine at the designed operating condition.Then, we analyse the total pressure loss terms at the designed operating condition. At last, comparison is conducted between the designed CO_2 turbine and a R245fa turbine. It is assumed that 50% of kinetic energy in the outlet of the rotor is recovered (Macchi and Perdichizzi, 1981). The design parameters are listed in Table 1, and the resulted parameters are listed in Table 2. Figure 1 shows the turbine stage meridional channel and the velocity triangles .The turbine efficiency is 0.852 at the designed operating conditon, and the corresponding output power is 6.29MW. The blade chord and the blade numbers for the stator and the rotor are 46mm and16 ,29mm and 25, respectively. The rotating speed is 18615.5 rpm.

Table 1: Design parameters

Parameters	**Value**
Inlet static pressure (P_1)	12870 kPa
Inlet static temperature (T_1)	385 K
Inlet static pressure (P_3)	6430 kPa
Mass flow rates ($\dot{m}$)	213.86 kg/s
Loading coefficients	1.05
Flow coefficients	0.4
Degree of reaction	0.45

Table 2: Design results

Parameters	**Results from reference (Da Lio et al.,2014)（R245fa）**	**My results（R245fa）**	**My results（CO_2）**
VR	1.7	1.73	1.66
Φ	0.4	0.4	0.4
ψ	1.05	1.05	1.05
sp	0.16	0.21	0.09

Specific speed	0.95		0.95		0.91	
R	0.45		0.45		0.45	
η	0.891		0.891		0.852	
C_x	37.5		37.3		66.9	
D_m	0.4		0.4		0.172	
Angular velocity	469（rad/s）		468（rad/s）		1949.4（rad/s）	
U	94		93		167.4	
	Stator	**Rotor**	**Stator**	**Rotor**	**Stator**	**Rotor**
Mach inlet	0.28	0.28	0.28	0.28	0.24	0.27
Mach outlet	0.79	0.73	0.79	0.73	0.75	0.73
Deflection	66	78	66	78	66	78
Blade span/blande chord	1.2	2.28	1.16	2.26	0.6 1	1.30
Trailing edge thickness/blade throat	7.25%	9.35%	7.31%	9.67%	9.05%	13.11%
Rotor radial clearance/ Blade span		1.14%		1.14%		2.68%
Flaring tip	12	16	11	12	5	7
Flaring hub	17	25	16	19	7	11
Blade axial chord	0.038(m)	0.031(m)	0.041(m)	0.031(m)	0.033(m)	0.023(m)
Total pressure loss coefficient						
Profile	0.015	0.016	0.015	0.015	0.016	0.016
Secondary	0.034	0.027	0.034	0.026	0.054	0.041
Trailling	0.006	0.011	0.006	0.012	0.010	0.023
Shock	0	0	0	0	0	0
Post expansion	0	0	0	0	0	0
Clearance	0	0.071	0	0.072	0	0.157
Total	0.056	0.124	0.055	0.125	0.079	0.238

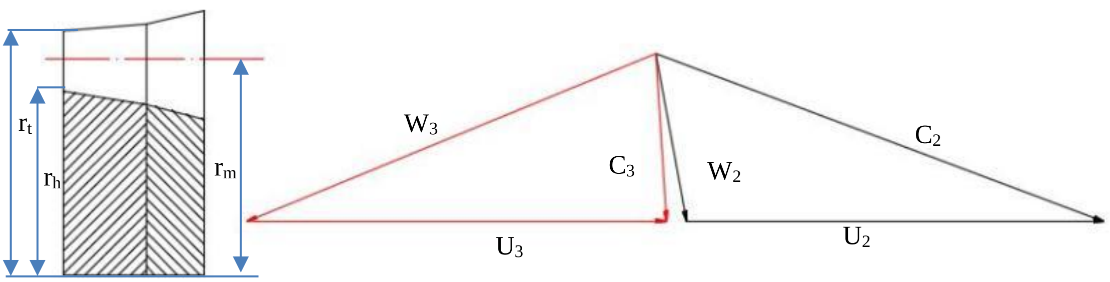

Figure 2: Turbine stage meridional channel and velocity triangles

Figure 3 depicts the total pressure losses coefficients and their percentages at design operating condition. The total pressure losses of the static blade includes the profile loss, the secondary flow loss and the trailing-edge loss. Their proportions are 20.3%, 68.4%, and 12.7% , respectively. The secondary loss is more than the sum of the others, so reducing the secondary flow loss is the main method to improve the static blade efficiency. The total pressure losses of rotor blade includes the profile loss, the secondary flow loss, the trailing edge loss, and the blade clearance loss, which respectively account for 6.7%, 17.2%, 9.7% and 66.0%. The blade clearance loss is the largest, which is followed by the secondary loss. The percentage of secondary loss in the stator is nearly 4 times than that of the rotor. The rotor blade clearance loss accounts for 49.5% of the turbine stage's total pressure loss, and the secondary flow loss accounts for 30%.These two losses account for nearly 80% of the turbine stage's total pressure loss. Therefore, it can be conclude that the main loss of the stator row comes from the secondary flow loss, and the main total pressure loss of the rotor row is the blade clearance loss. These two losses are also the main losses of the turbine stage. In summary, reducing the clearance loss and the secondary loss are the most effective methods to improve the turbine efficiency.

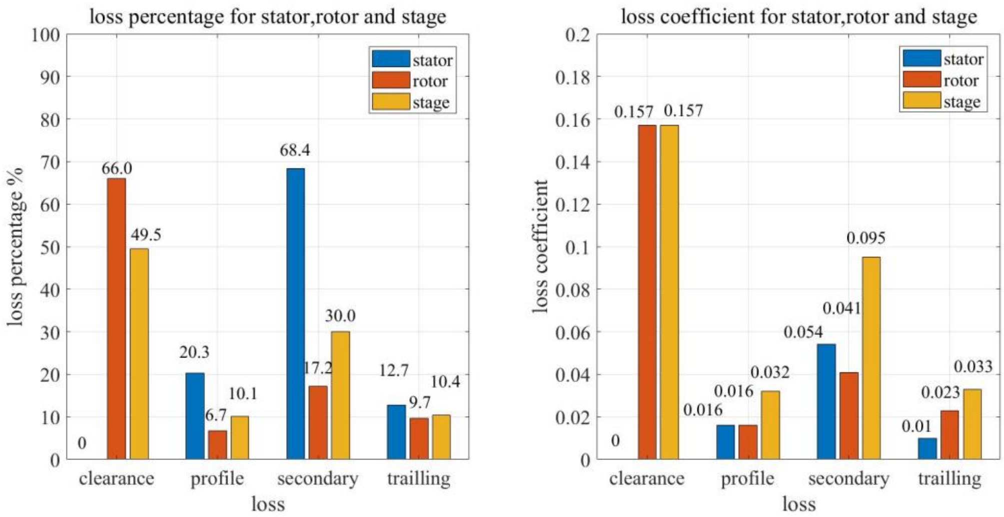

Figure 3: Total pressure loss of turbine with CO2 as working fluid

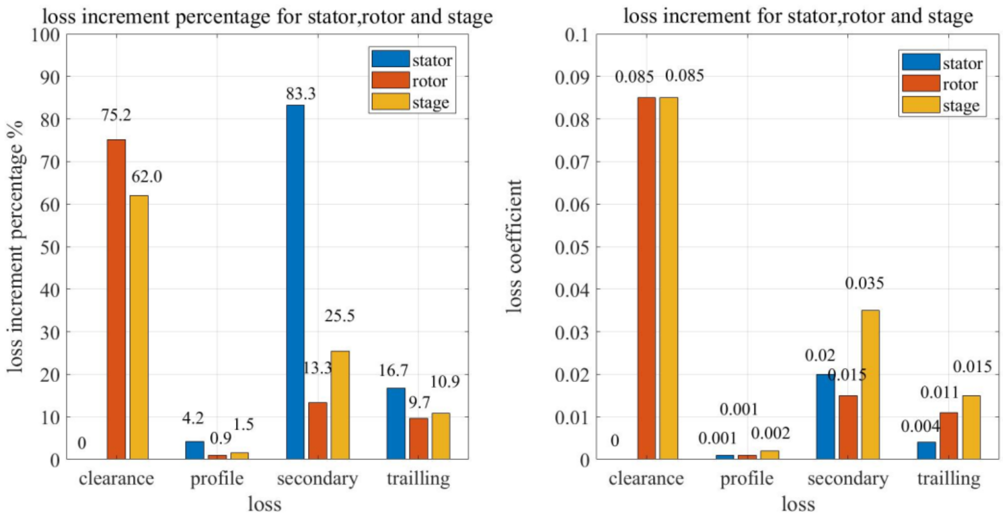

Figure 4: Compared with R245fa turbine, the total pressure loss increment of the turbine with CO_2 as working fluid

Figure 4 shows that all of the total pressure loss coefficients of the CO_2 turbine are higher than that of R245fa turbine. The secondary flow loss of the static row is increased by 0.02, accounting for 83% of the increase in the static row loss. The trailing-edge loss of the static row is increased by 0.004, accounting for 16.7% of the increase in the static row loss. There is little difference in the profile loss between the CO_2 turbine and the R245fa turbine, which is mainly results from the same dimensionless coefficient, R, ϕ, ψ . The increase in the loss of the rotor row are quite significant, except for the profile loss. The most important increase in the total pressure loss coefficient is blade clearance loss coefficient, which is 0.085, accounting for 75.2%. The loss increase of the trailing-edge loss and secondary loss account for 27% among the rotor blade row, so these pressure loss increase should not be ignored. The pressure loss increase due to the trailing-edge loss is as important as that of the secondary loss because there is a little difference between their percentages among the rotor blade row. The increasing clearance loss, the secondary loss and the trail-edge loss mainly lead to a lower efficiency of the CO_2 turbine than that of the R245fa turbine, and the most significant increase in total pressure loss is the clearance loss, which is the most vital reason for a lower turbine efficiency.

The dimensional parameter, $sp = \dot{V}_3^{0.5} \big/ (H_{01} - H_{3s})^{0.25}$, represents for the actual turbine dimensions (Macchi and Perdichizzi, 1981) . As we can see from Table 2, the size parameter of the CO_2 turbine is smaller than that of the R245fa turbine; accordingly, there are smaller aspect ratio (blade span/ blade chord) and ratio of blade clearance to span and a higher ratio of blade trailing-edge thickness to throat width. The ratio of blade clearance to span of the CO_2 turbine is more than twice that of the R245fa turbine, which is constrained by technical level and limited the decrease of blade radial clearance with the decrease of the blade span and significantly increase the rotor blade clearance loss. It is pointed out that when the flow deflection angle is the same, as aspect ratio decreases, the proportion of secondary vortex in the blade increases, so that the secondary flow loss increases rapidly (Moustapha et al., 2003). An increase in the ratio of the trailing edge thickness to the throat width in the CO_2 turbines increases the trailing-edge loss. So, it confirms that a smaller actual turbine dimension increases all of the loss terms and decreases the efficiency. The density of CO_2 is more than 10 times that of R245fa at the outlet of the turbine, which results in a smaller volume flow rate; even though, the mass flow rate of the CO_2 turbine is nearly 4 times than that of the R245fa turbine. The smaller volume flow rate needs to a smaller flow area, so it decreases the turbine size, such as blade span, blade throat width and mean diameter.

4. CONCLUSIONS

A single stage axial CO_2 turbine is designed by one dimensional mean-line method, and the resulted turbine efficiency is 0.852 with the corresponding output power being 6.29MW. The parameters about the flow passage and the rotating speed are obtained. Then, the total pressure losses are analyzed to acquire a detailed knowledge about the loss proportion for the CO_2 turbine and to label the optimum director for the turbine design. At last, the comparison is conducted between the CO_2 turbine and the R245fa turbine to understanding the characteristics of the CO_2 turbine. The corresponding results are summarized as below.

- The maximum efficiency of the designed CO_2 turbine is 0.852 at the designed operating condition, and this efficiency can be further optimized by regulating the loading and flow coefficients and degree of reaction.
- The main total pressure loss of the stator blade row origins from the secondary flow loss, accounting for 68.4%, and the main total pressure loss of the rotor blade row is the blade clearance loss, accounting for 66.0% . Those two losses are the main losses for the CO_2 turbine stage.
- The density of CO_2 is considerably higher than R245fa, so CO_2 turbine possesses a smaller actual dimension (sp) which increases the overall turbine total pressure loss and decreases the turbine efficiency. The most important increases for the rotor row arises from the blade clearance loss, accounting for 75.2% , and the most important increases for the static row origins from the secondary loss, accounting for 83.3% .

NOMENCLATURE

A	area	(m^2)
C_x	meridional velocity	(m/s)
C	absolute velocity	(m/s)
D_m	mean diameter	(m)
H	enthalpy	(J/kg)
$\dot{m}$	mass flow rate	(kg/s)
P	pressure	(Pa）
r	radius	(m)
R	degree of reaction	(-)

S	entropy	(J/(kg/K))
sp	size parameter	(m)
T	temperature	(K)
U	peripheral velocity	(m/s)
$\dot{V}$	volume flow rate	(m^3/s)
W	relative velocity	(m/s)
Y	Total pressure loss coefficient	(-)
α	absolute flow angle	(°)
β	relative flow angle	(°)
η	efficiency	(-)
ρ	density	(kg/m^3)
ϕ	flow coefficient	(-)
ψ	loading coefficient	(-)

Subscripts

0	total state
1	stator inlet
2	stator exit and rotor inlet
3	rotor exit
h	hub
m	mean direction
N	stator
R	rotor
s	isentropic
t	tip
ts	total-static
tt	total to total

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ACKNOWLEDGEMENT

This research was funded by the National Natural Science Foundation of China, grant number 51876009.