TORQUE RESEARCH OF SINGLE SCREW EXPANDERS

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ABSTRACT

In recent years, people have paid much attention to single screw expanders. Many experiments have been carried out on the performance of single screw expanders in fundamental applications such as industry waste heat recovery, compressed air power system, and pressure energy recovery. This paper presents a theoretical torque model of single screw expander under the conditions of ideal adiabatic expansion and air as working fluid. It was found that the torque ratio is independent of inlet pressure of single screw expander and is close to 1. It indicates that the single screw expander runs smooth. Besides, the calculated output power based on the theoretical torque model was close to the measured output power by experiments, if the shaft efficiency from experiments results and output work loss under over-expansion process were considered. And it demonstrates that this model can be used to estimate the output power of single screw expander before designing it according to the diameter of main rotor, volume ratio, inlet and back pressure of single screw expander.

1. INTRODUCTION

In recent years, with heavy fog and haze dominating our life, people realized the bad effects of PM2.5, such as coughs, asthma, sore throats and other respiratory illnesses. Many researchers and industries have realized the importance of using low-grade energy so that we can contribute to a permanent “Beijing Blue”. Without doubt, ORC system is a great technology to save energy and keep the energy clean. A whole ORC system is basically composed of evaporator, expander, condenser and working fluid pump. Among these components, the selection of expander type is crucial to this efficiency of the system because the expander decides how much pressure energy is converted to power.

Among piston expanders, turbo expanders, scroll expanders and screw expanders, single screw expanders (SSEs) have received much attention in recent years due to its unique advantages, such as balanced loads, long working life, simple structure, low vibration and so on. In 2008, Ma Chongfang and his team [1] first reported the SSEs with 10kW and 40kW in Science Times. He et al. [2] carried out the study on the power system of compressed air based on three-stage SSEs. By programming and mathematical modeling results, it is shown that the single screw expander (SSE) has a good potential to be used in power systems. Liu et al. [10] verified the feasibility of SSE on ORC system with heat source of flue gas. Wang et al. [3] used compressed air as working fluid to verify the performance of SSE prototype. The designed flow rate of SSE is 1.1Nm^3/min. The performance tests were conducted under different conditions including different intake flow, different humidity, constant torque and constant rotational speed. According to the experimental results, it is shown that the output power is 5kW at rotary speed of 2850r/min, discharge temperature is -45℃, the maximum temperature drop was about 62℃, and the maximum of adiabatic efficiency and total efficiency were 59% and 32.5% respectively. And it also shows that this SSE prototype has good part-load performance. And lubrication may be a factor to cause low adiabatic efficiency. He et al. [4] carried out experiments on the performance of the SSEs with compressed air as working fluid under different intake pressures and showed that the measured torque has a large increase as the intake pressure increases and has a slight decrease as the rotational speed increases. Desideri et al. [5] evaluated the SSE modified from a standard compressor and developed a steady-state model of the whole ORC unit. In this experiment, the working fluid is Solkatherm (the azetropic mixture of HFC365mfc and YR-1800), the volume ratio of the expander is 5 and the diameter of main rotor is 155mm. The results show that the maximum expander isentropic efficiency and generated power are 64.78% and 7.8 kWe respectively. The whole cycle efficiency peaked at 9.8% with the evaporating temperature of 108℃. Ziviani et al. [6] established a detailed model of SSE based on the geometric parameters, heat transfer model and governing equations. And then verified the calculated data using experiment data between mass flow rates and output power.

Although the performance of SSE have been carried out in the ORC system and power system, little attention has been paid to the theoretical torque model of SSE which can be used to predict if the SSE works in a good state and also can be used to estimate how much output power are produced. The paper presents a detailed torque model of single screw expander and gives out comparison of calculated torque and experimental data.

2. Theoretical torque calculation of single screw expander

2.1 Main Parameters of Single Screw Expanders

The main rotor and a pair of gate rotors are the key components in SSE (shown in Figure 1). In this paper, the SSE is designed by our laboratory team. The single screw expander belongs to CP type. It means that the main rotor is machined by a cutting tool with straight line like a single tooth of gate rotor. The gate rotor is generated by envelope of main rotor. The main rotor and gate rotor are machined by special purpose machine developed by our team (shown in Figure 2). Their meshing relations are just like worm gears. The main structure parameters of meshing pair are the foundation to calculate the basic volume, the volume ratio and the whole design of SSE. The main parameters are illustrated in Figure 3.

Figure 1: Single screw expander

Figure 2: Main rotor and gate rotor

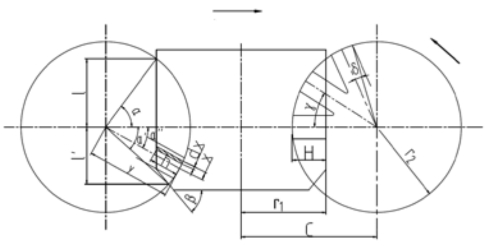


Figure 3: Structure of single screw meshing pair

$$i = \frac{\theta_1}{\theta_2} = \frac{z_2}{z_1} \tag{1}$$

$$\gamma = \frac{2\pi}{z_2} \tag{2}$$

$$k_0 = \frac{r_2}{r_1} \tag{3}$$

$$k = \frac{H}{r_1} \tag{4}$$

$$C = r_1(1 + k_0 - k) \tag{5}$$

$$l = \mathrm{r}_1 \sqrt{2k_0 k - k^2} \tag{6}$$

$$\alpha = \arcsin\left(\frac{\sqrt{2k_0 k - k^2}}{k_0}\right) \tag{7}$$

$$\alpha' = \arcsin\left(\frac{l'}{r_1 k_0}\right) \tag{8}$$

$$l' = 0.7l \tag{9}$$

$$\xi = \frac{\mathrm{e}}{2\mathrm{r}_1} \tag{10}$$

$$b = 2r_1[(k_0 - k)\sin(\frac{\gamma}{2}) - \xi\cos(\frac{\gamma}{2})] \tag{11}$$

$$b_s = \frac{b}{2r_2} = \frac{(k_0 - k)\sin(\frac{\gamma}{2}) - \xi\cos(\frac{\gamma}{2})}{k_0} \tag{12}$$

$$\delta = \arcsin(b_s) \tag{13}$$

2.2 Swept Volume Equation

In this paper, based on the method used by Sun Guangsan (1988), we calculate the whole single groove basic volume of the main rotor. The method mainly refers to the area of the gate rotor and the centroid of the gate rotor when meshing with the main rotor. The whole single groove basic volume is divided into two parts. One part is the volume of the gate rotor sweeping the groove of the main rotor from the front edge of the gate rotor just coming into the groove to the back edge of the gate rotor just coming to the groove (in Figure 4). The other part is the volume of the gate rotor sweeping the groove from the back edge of the gate rotor just coming into the groove to just beginning discharge. However, there is some difference between the volume equation inferred by Sun Guangsan and this volume equation in this paper. This volume equation of this paper is mainly expressed by non-dimensional coefficients of k and k_0 and the radius (r_1) of main rotor. The formulae are as follows:

$$V = V_1 + V_2 = \int_{\alpha-\delta}^{\alpha+\delta} A \cdot i\bar{R}_1 d\theta_2 + \int_{\alpha_{ds}}^{\alpha-\delta} A \cdot i\bar{R}_2 d\theta_2 \tag{14}$$

$$A_{\alpha-\delta} = k_0^2 r_1^2 \arcsin(b_s) + b_s\sqrt{1-b_s^2} - 2b_s k_0 r_1^2 (k_0 - k)\sec(\alpha - \delta) \tag{15}$$

$$\begin{cases} A = A_{\alpha-\delta} \cdot \left(\dfrac{\alpha + \delta - \theta_2}{2\delta}\right)^2 ; \alpha - \delta \le \theta_2 \le \alpha + \delta \\ A = \int_{-\frac{b}{2}}^{\frac{b}{2}} (\sqrt{k_0^2 r_1^2 - x^2} - (k_0 r_1 - H)\sec\theta_2 + x\tan\theta_2) dx ; \theta_{db} \le \theta_2 \le \alpha - \delta \end{cases} \tag{16}$$

(a) Cross-section **(b)** Primary view

Figure 4: Movement relations of single screw meshing pair

For the single screw expander, the whole expansion process is divided into three phases. It is suction, closed expansion and discharge respectively. The first part volume (V_1) is an inherent suction volume for every main rotor when the angle of gate rotor is between $\alpha - \delta$ and $\alpha + \delta$. So when we calculate the expander suction volume at any time during the suction phase, the formula is as follows:

$$V=\begin{cases}V_1=\int_{\alpha-\delta}^{\alpha+\delta}A\cdot i\bar{R}_1d\theta_2,\alpha-\delta\le\theta_2\le\alpha+\delta\\ V_1+V_2=\int_{\alpha-\delta}^{\alpha+\delta}A\cdot i\bar{R}_1d\theta_2+\int_{\theta_2}^{\alpha-\delta}A\cdot i\bar{R}_2d\theta_2,\theta_{se}\le\theta_2<\alpha-\delta\end{cases}\tag{17}$$

When we calculate the closing expansion volume at any time during the closing expansion phase, the formula is as follows:

$$V=\int_{\alpha-\delta}^{\alpha+\delta}A\cdot i\bar{R}_1d\theta_2+\int_{\alpha-\delta}^{\theta_2}A\cdot i\bar{R}_2d\theta_2,\theta_{db}\le\theta_2<\theta_{se}\tag{18}$$

When we calculate the discharge volume at any time during the discharge phase, the formula is as follows:

$$V=\begin{cases}\int_{\alpha-\delta}^{\alpha+\delta}A\cdot i\bar{R}_1d\theta_2+\int_{\alpha-\delta}^{2\theta_{db}-\theta_2}A\cdot i\bar{R}_2d\theta_2,2\theta_{db}-\alpha+\delta\le\theta_2<\theta_{db}\\ \int_{\alpha-\delta}^{\alpha+\delta}A\cdot i\bar{R}_1d\theta_2,2\theta_{db}-\alpha-\delta\le\theta_2<2\theta_{db}-\alpha+\delta\end{cases}\tag{19}$$

2.3 Calculation of Instantaneous and Average Torque

For the single screw expander, the instantaneous torque of main rotor is equal to product of the force on the gate rotor tooth area and the distance from the centroid of the gate rotor tooth area to the center of main rotor. Assuming that the pressure on the lower teeth surface of gate rotor is equal to the discharge pressure and the instantaneous torque is divided into three phases according to the operation process of single screw expander. Assuming that the expansion is a perfect adiabatic process and that the working fluid is compressed air, the formulae of the instantaneous torque are as follows:

$$\begin{cases}F=(p_{in}-p_{out})A,\alpha-\delta<\theta_2\le\alpha+\delta\\ F=(p_{in}-p_{out})A,\theta_{se}<\theta_2\le\alpha-\delta\\ F=p_{in}\left(\frac{V_2}{V_1}\right)^k A,\theta_{db}<\theta_2\le\theta_{se}\\ F=(p_{out}-p_{out})A=0,2\theta-\alpha-\delta\le\theta_2\le\theta_{db}\end{cases}\tag{20}$$

$$\begin{cases}T=(p_{in}-p_{out})A\bar{R}_1,\alpha-\delta<\theta_2\le\alpha+\delta\\ T=(p_{in}-p_{out})A\bar{R}_2,\theta_{se}<\theta_2\le\alpha-\delta\\ T=p_{in}\left(\frac{V_2}{V_1}\right)^k A\bar{R}_2,\theta_{db}<\theta_2\le\theta_{se}\\ T=(p_{out}-p_{out})A\bar{R}_2=0,2\theta_{db}-\alpha-\delta\le\theta_2\le\theta_{db}\end{cases}\tag{21}$$

Therefore, the average torque is given by the following equation (22):

$$T_{a1}=\frac{\int_{t_1}^{t_2}Tdt}{t_2-t_1}\tag{22}$$

The theoretical output power can be written as

$$P_{e1}=\frac{T_{a1}n}{9550}\tag{23}$$

2.4 Output Power Loss Percentage of Over or Under Expansion Process

It is known that the single screw expander is a positive displacement expander without discharge valve. As a result, there is a built-in volume ratio ($\eta_{\mathbf{iv}}$) for the single screw expander. If the inlet pressure of expander (P_{in}) is certain, the internal outlet pressure ($P_{\mathbf{iout}}$) is a certain value when the expansion ends and the expander begins discharging. But the internal outlet pressure finally needs to be equal to the back pressure (P_d) of the discharge pipe, so that the working fluid can be discharged. The internal and external expansion ratio (ε_i and ε_d) can be obtained in the following equation (24-25). If the internal outlet pressure $P_{\mathbf{iout}}$ is equal to the external outlet pressure P_d, the expansion process is called as 'full expansion' (shown in Figure 5a). If the internal outlet pressure $P_{\mathbf{iout}}$ is less than the external outlet pressure P_d, the expansion process is called as 'over expansion' (shown in figure 5b). If the internal outlet pressure $P_{\mathbf{iout}}$ is greater than the external outlet pressure P_d, the expansion process is called as 'under expansion' (shown in Figure 5c). For the over and under expansion, there is additional power consumption and it will decrease the output power. The output work loss Δw is shown in the triangle with shadow part. The output power loss percentage η_w can be calculated in equation 26.

$$\varepsilon_i = \frac{p_{in}}{p_{iout}} \tag{24}$$

$$\varepsilon_d = \frac{p_{in}}{p_d} \tag{25}$$

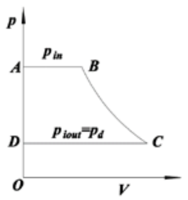
$$\eta_w = \frac{\Delta w / t}{w_d / t} = \frac{\Delta w}{w_d} = \frac{\left(\frac{\kappa-1}{\kappa}\varepsilon_d^{-1}\varepsilon_i^{\frac{1}{\kappa}} - \varepsilon_d^{\frac{1-\kappa}{\kappa}} + \frac{1}{\kappa}\varepsilon_i^{\frac{1-\kappa}{\kappa}}\right)}{1-\varepsilon_d^{\frac{1-\kappa}{\kappa}}} \tag{26}$$

Hence, the output power P_{e2} after considering the output power loss under different expansion type can be written as

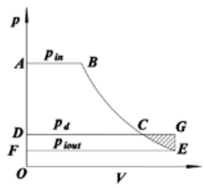
$$P_{e2} = P_{e1}(1-\eta_w) \tag{27}$$

Furtherly considering the shaft efficiency η_s, the output power P_{e3} is given by

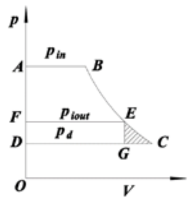
$$P_{e3} = P_{e1}(1-\eta_w)\eta_s \tag{28}$$



(a) Full expansion



(b) Over expansion



(c) Under expansion

Figure 5: Expansion type

3. Results and Discussions

3.1 Experimental Results

The single screw expander designed by our laboratory is installed in a compressed air power system. This whole system includes five parts: air intake and exhaust circuit, oil lubrication circuit, power testing system, water cooling system and data acquisition system. The air source comes from a compressed air storage tank. The rotary speed of single screw expander is 3000r/min. And the inlet air measured pressure was varied from 4.97bar to 9.975bar. The main parameters of SSE are listed in

Table 1. The measured variables and their uncertainties are shown in Table 2. And the experimental results are shown in Table 3.

The built-in volume ratio of single screw expander is 5. This experiment uses the air as working fluid. And the adiabatic exponent of air is assigned as an ideal value of 1.4. Assuming that the expansion process is an adiabatic process, according to the adiabatic equation of air, the ideal adiabatic expansion ratio is 9.518. As can be seen from Table 3, the expansion ratios calculated by experiments data are less than that of ideal adiabatic expansion ratio. So this single screw expander was in an over-expanded working state. This inlet pressure in this experiment just sets less than 10 bar, because there is an upper end of measured output power for the eddy current dynamometer (0~10 kW). That's the reason that experiments with higher inlet pressure cannot be carried out.

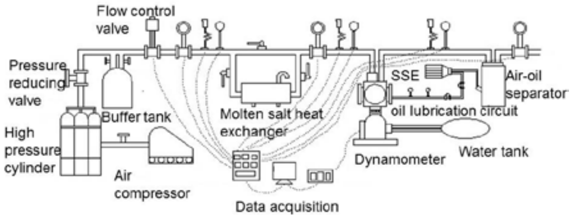


Figure 6: General layout of experimental setup with air compressor

Table 1: Main parameters of single screw expander

Parameters	
Diameter of main rotor	155mm
Diameter of gate rotor	155mm
Transmission ratio	11/6
Center distance	124mm
Volume ratio	5
Tooth width of gate rotor	23.4mm

Table 2: Measured variables and their uncertainties

Variable	**Units**	**Uncertainties**
Temperature	℃	0.5%
Pressure	bar	0.5%
flow	m^3/h	1.5%
Torque	N.m	±0.2~0.4%FS
Rotate speed	r/min	±1%

Table 3: Experimental results of single screw expander

Inlet pressure P_{in} **(bar)**	**Rotate speed** n **(r/min)**	**Torque** T_{ex} **(N.m)**	**Output Power** P_{ex} **(kW)**	**Outlet Pressure** P_d **(bar)**	**External expansion ratio** ε_d	**Shaft efficiency** η_s **(%)**
4.971	2999	11.464	3.600	0.500	3.981	48.189
5.976	3000	15.785	4.958	0.600	4.340	51.387
7.033	2999	20.166	6.333	0.700	4.725	54.046
8.010	2999	24.028	7.457	0.800	5.005	56.393
8.975	3000	27.776	8.72	0.967	5.073	57.679
9.975	3000	31.583	9.92	1.100	5.226	58.274

3.2 Calculated Torque and Torque Ratio of Single Screw Expander

Based on the equation presented in this paper, the Matlab codes are programmed and developed to calculate the instantaneous torque of single screw expander with the rotary angle changes of main rotor. Assuming that the inlet pressure is equal to the measured values by experiments, the expansion process is a perfect adiabatic expansion, and that there is no mechanical and frictional loss. From the Figure 7, the output torque value appears again with period 60 degrees. In a period, no matter how much the inlet pressure is equal to, all the relative highest point and lowest point appears at the same degrees (46degrees and 30degrees respectively). In Figure 7, it is shown that the output torque of single screw expander increases with the increase of inlet pressure. This trend is in accordance with that of experimental results.

Torque ratio is defined as theoretical torque at different angle of main rotor in to average torque of single screw expander and can be calculated in equation (shown in equation 29). Torque ratio can reflect the operation performance of single screw expander. The torque ratio is closer to 1, the single screw expander runs smoother. It can be observed in Figure 8 that the torque ratio is independent of inlet pressure, and that there is a small fluctuation for torque ration around at the horizontal line (Torque ratio of the horizontal line is equal to 1). It could be concluded that the single screw expander runs steadily.

$$\lambda = \frac{T}{T_{a1}} \tag{29}$$

3.2 Calculated and Measured Output Power of Single Screw Expander

According to the trapezoid rule of numerical integral formulas, the average torque in one period is obtained by equation (22), and then by equation (23-28), the calculated output power can be calculated (shown in Table 5). The relative error can be obtained by the following equation (30). As listed in table 5, it reveals that the over-expansion would cause great output power loss and that the output power loss percentage will decrease sharply with a slow increase of external expansion ratio.

$$Er = \frac{P_{e3} - P_{ex}}{P_{ex}} \tag{30}$$

It can be observed in Figure 9 that the output power of single screw expander increases linearly with growth of inlet pressure. The relative error of output power is given in Table 4. The difference value of output power is around at 1 and there is almost no change for ΔP, but the relative error is bigger compared to the change of difference value ΔP of it. Because the output power value itself is not a big value. Although the relative error is not much small, the output power by this mathematical model still can be accepted. That's because in this model, the leakage and friction loss are not considered.

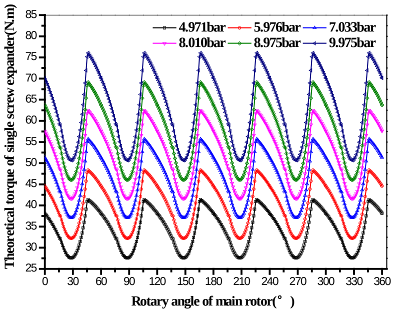


Figure 7: Theoretical torque of main rotor at different rotary angle of main rotor

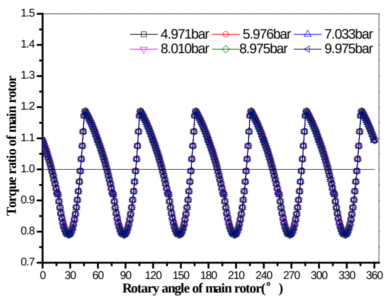


Figure 8: Torque ratio at different angle of main rotor

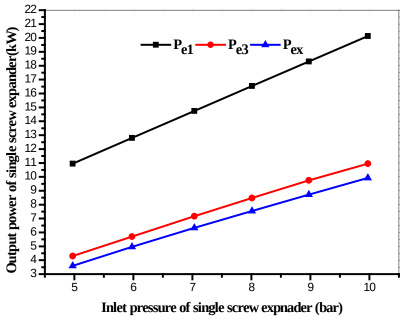


Figure 9: Output power of single screw expander with changes of inlet pressure

Table 4: The relative error, the theoretical output power, the output power after considering shaft efficiency and output power loss percentage and the measured output power

Inlet Pressure(bar)	P_{e1} (kW)	η_w (%)	P_{e3} (kW)	P_{ex} (kW)	ΔP (kW)	Er (%)
4.971	10.949	18.46	4.301	3.600	0.701	19.5
5.976	12.800	13.48	5.704	4.958	0.746	15.0
7.033	14.739	10.01	7.168	6.333	0.834	13.2
8.010	16.537	7.98	8.479	7.547	0.932	12.4
8.975	18.301	7.57	9.748	8.725	1.023	11.7
9.975	20.135	6.67	10.951	9.920	1.031	10.4

4. Conclusions

In this paper, based on the modified mathematical model of basic volume for main rotor, the theoretical model torque model of single screw expander is established under ideal adiabatic expansion process and air as working fluid. And this paper presents the output power loss percentage equation during under or over expansion process. According to the present analysis, the following results are concluded:

(1) From the torque equation, it can be found that the theoretical torque mathematical model is independent of rotation speed of single screw expander. The instantaneous torque and the torque ratio reflect that the single screw expander runs steadily.

(2) The output power loss percentage equation can also be applied to polytropic process in twin screw expanders and single screw expanders. The κ value in equation (26) is replaced by the polytropic exponent n .

(3) By comparison between calculated output power by mathematical model and measured torque by experiments, this relative error is 10%~19% while the differential value ΔP between calculated and measured output power is just around at 1. Hence, this model can be used to estimate the output power of SSE under given diameter of main rotor, inlet pressure, built-in volume ratio and back pressure when design the SSE. And there is a rapid increase for η_w with a slight decrease of external expansion ratio. So the SSE should avoid over-expansion process when design it in order to lower great output power loss.

This theoretical mathematical model can just be used to estimate the operation state and output power of SSE under ideal adiabatic expansion process and air as working fluid, if the diameter of main rotor, inlet pressure, volume ratio and back pressure are given. But there is still some room to improve. On the one hand, the inlet pressure loss should be included in the future study in order to make the calculated torque be closer to the measured torque. On the other hand, in order to calculate the torque of SSE in ORC system, the state equation of organic fluid need to be studied. With the continuous improvement of this model, this model will be more accurate.

NOMENCLATURE

Variable	definition	units
i	transmission ratio	(–)
z	number of grooves or teeth	(–)
γ	indexing angle of gate rotor	(rad)
r	radius	(mm)
k_0	ratio of the main rotor radius in to the gate rotor radius	(–)
k	meshing depth coefficient	(–)
H	the maximum meshing depth	(mm)
C	Center distance of single screw meshing pair	(mm)
l	axial length of the discharge side	(mm)
α	meshing angle of the discharge side	(–)
α'	meshing angle of the suction side	(–)
l'	axial length of the suction side	(–)
b_s	tooth width coefficient	(–)
b	tooth width of the gate rotor	(mm)
δ	half angle of the tooth width	(rad)
e	the minimum width of the groove wall	(mm)
ξ	coefficient of the groove wall	(−)
A	area of gate rotor tooth meshing with main rotor	(mm^2)
V	volume of main rotor groove	(mm^3)
θ	rotary angle	(rad)
p	inlet or outlet pressure of single screw expander	(Mpa)
p_d	back pressure	(Mpa)
p_{iout}	internal expansion pressure of single screw expander	(Mpa)
T	instantaneous output torque of single screw expander	(N.m)
T_{a1}	theoretical average output torque	(N.m)
P	output power of single screw expander	(kW)
η_s	shaft efficiency of single screw expander	(−)
η_w	output power loss percentage of single screw expander	(−)
λ	torque ratio	
Er	relative error	(−)
ε_i	internal expansion ratio	(−)
ε_d	external expansion ratio	(−)
ΔP	differential value of output power	(kW)

Subscript

1	main rotor
2	gate rotor
e1	theoretical
e2	after considering output power loss percentage
e3	after considering shaft efficiency

ex	experimental results
in	inlet
out	outlet
se	suction ending
db	discharge beginning

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