Abstract: The energy loss of the multi-stage centrifugal pump was investigated by numerical analysis using the entropy generation method with the RNG k-ε turbulence model. Entropy generation due to time-averaged motion and velocity fluctuation was mainly considered. It was found that the entropy generation of guide vanes and impellers account for 71.2% and 23.3% of the total entropy generation under the designed flow condition. The guide vanes are the main hydraulic loss domains and their entropy generation is about 9 W/K, followed by impellers. There are vortices at the tongue of the guide vane inlet as well as flow separations in the impellers, which lead to entropy generation. The fluid impacts the outer surface of the guide vanes, resulting in the increase in entropy generation. There are refluxes near the guide vane tongues which also increase the entropy generation of this part. The entropy generation distribution of the guide vanes and impellers was investigated, which found that the positive guide vane has more entropy generation compared with the reverse guide. The entropy generation of the blade suction surface is higher compared with the pressure surface. This study indicated that the entropy generation method has distinct advantages in the assessment of hydraulic loss.

Keywords: multi-stage centrifugal pump; entropy generation; numerical analysis; energy loss

1. Introduction

Multi-stage centrifugal pumps are widely applied in industry fields, such as in power stations, refineries, petroleum extraction, transportation, and so on. With the increasing demand for pump applications, the traditional design scheme is not enough as it has a long design period and high cost, and it cannot clearly show the details of the internal flow. Therefore, new research schemes are needed. With the improvement of the computer, CFD technology is now widely applied to the fluid machinery [1,2].

Three-dimensional flow analysis in the pump is a basic means to improve the pump efficiency, predict the pump performance and analyze the energy loss. Figure 1 displays a BB5 multi-stage centrifugal pump that is composed of an inlet chamber, impellers, guide vanes, and shell. Due to the complexity of the multi-stage centrifugal pump structure and the rotation of impellers, there are many complicated flow phenomena in the pump such as secondary flow, recirculation flow, and boundary layer separation flow. Many researchers have used numerical simulation to study the flow field in the centrifugal pump. Guleren and Pinarbasi [3] analyzed the stalled flow of the centrifugal pump by the standard k-ε turbulence model. Jafarzadeh et al. [4] analyzed and compared the errors of the numerical results and experimental results in the centrifugal pump under different turbulence models. It was found that the results calculated with the RNG k-ε turbulence model are more accurate. Zhou et al. [5] used the S-A DES model and SST k-ω model to simulate and analyze the whole flow channel of the double-suction centrifugal pump, respectively. It resulted that the S-A DES model can capture more flow phenomena and it
is more suitable to solve the unsteady flow. Feng et al. [6] analyzed the mechanism of the tip-clearance flow and the motion law of the gap vortices. It was found that the leakage flow in the tip clearance affects the mechanical fluid transfer and unsteady flow field in the impeller. Zhu et al. [7] used the RNG k-ε turbulence model and CFX software to calculate the Reynolds time-averaged N-S equations and studied the interaction and impact of the multi-stage centrifugal pump between various levels.

![Figure 1. Internal structure of a BB5 multi-stage centrifugal pump.](image)

However, it is not intuitive to analyze the flow information in fluid machinery only from the perspective of hydrodynamics. Due to the fluid viscosity and velocity fluctuation, the fluid mechanical energy is transformed into heat irreversibly. Entropy is a state parameter derived from the second law of thermodynamics and reflects the irreversibility of spontaneous processes. Bejan and Kestin [8] first put forward the concept of entropy generation calculation in the flow heat transfer process. Spurk [9] derived the transport equation of the entropy in detail. Kock and Herwig [10] treated the transport equation of entropy generation by the Reynolds-averaged process and classified entropy generation. Zhang et al. [11] defined the head loss coefficient by the method of entropy generation integration, which was verified by a conical diffuser case. Chu and Liu [12] studied the two-dimensional high-temperature limited jet and analyzed the primary causes of irreversible process loss by the entropy generation method. The research of entropy generation is a way to study the hydraulic loss and has been widely used in rotating machinery [13–20], which is meaningful to further optimize the structure and comprehensively explore the internal flow of the centrifugal pump. Li et al. [21] established a simplified hydraulic loss model by the entropy generation method and calculated the loss by three centrifugal pump models with different blade thickness. It was found that the calculated errors of the three models were close. Pei et al. [22] assessed the effect of the axial clearance between the impeller and diffuser cascade on energy loss in the pump by the generation method. It was found that turbulent dissipation is the primary reason causing hydraulic loss. Zhang et al. [23] assessed the energy consumption of a centrifugal pump by the local entropy generation method and accurately calculated the size and location of the hydraulic loss. It resulted that the local entropy generation calculation method is reliable. Li et al. [24] researched the head rupture characteristics that were caused by cavitation in the centrifugal pump. It was found that there is a certain correlation between entropy generation and the external characteristics of the centrifugal pump. Feng et al. [25] obtained a detailed distribution of the hydraulic loss and analyzed the distribution of energy loss of flow components during the power-off transition period in the centrifugal pump by the entropy generation method. It resulted that the energy loss of the centrifugal pump is related to the flow separation, backflow and vortex in the flow field. The internal structure and entropy generation of a multi-stage centrifugal pump are more complex than that of a general centrifugal pump. Koranteng Osman et al. [26] used the entropy generation term to evaluate the turbulent
dissipation and to characterize the flow loss, studied the flow loss in a two-stage centrifugal pump, and verified the numerical results with experiments. Ren et al. [27] studied the influence of the guide ring on the energy loss of the multi-stage centrifugal pump by the entropy generation method. The calculated entropy generation rate value was in good agreement with the actual hydraulic loss value in the pump. Wu et al. [28] study the energy loss characteristics of a multi-stage pump with a different number of diffuser vanes by the entropy generation method. It was found that with the increase in the number of diffuser blades, the flow is more stable and the entropy generation decreases, but the intensity of pressure pulsation in the diffuser increases slightly. Therefore, the entropy generation analysis with CFD can be used to reveal the energy loss position of a centrifugal pump for selective optimization. However, the research on entropy production of a multi-stage centrifugal pump is not enough.

Based on the entropy generation method, this paper further studies the complex flow and energy loss of flow passage components in the multi-stage centrifugal pump, analyzes the causes of entropy generation, and provides a theoretical basis for the optimal design of the multi-stage centrifugal pump.

2. Entropy Generation Theorem

This study assumes that the temperature of the multi-stage centrifugal pump is constant, thus only considered the entropy generation rate concerning viscous dissipation.

The transportation equation of specific entropy for single-phase incompressible flow is as follows [29]:

\[ \rho \left( \frac{\partial s}{\partial t} + u_1 \frac{\partial s}{\partial x_1} + u_2 \frac{\partial s}{\partial x_2} + u_3 \frac{\partial s}{\partial x_3} \right) = div \left( \frac{\partial q}{\partial T} \right) + \Phi + \Phi_\Theta \] (1)

where \( s = \Phi + s' \), \( \Phi \) is the viscous dissipation term, and \( \Phi_\Theta \) is the entropy production term.

Equation (1) is changed as follows:

\[ \rho \left( \frac{\partial s}{\partial t} + u_1 \frac{\partial s}{\partial x_1} + u_2 \frac{\partial s}{\partial x_2} + u_3 \frac{\partial s}{\partial x_3} \right) = div \left( \frac{\partial q}{\partial T} \right) - \rho \left( \frac{\partial u_1' s'}{\partial x_1} + \frac{\partial u_2' s'}{\partial x_2} + \frac{\partial u_3' s'}{\partial x_3} \right) + \frac{\Phi}{T} + \frac{\Phi_\Theta}{T^2} \] (2)

For incompressible flow:

\[ \Phi = 2\mu \left[ \left( \frac{\partial u_1'}{\partial x_1} \right)^2 + \left( \frac{\partial u_2'}{\partial x_2} \right)^2 + \left( \frac{\partial u_3'}{\partial x_3} \right)^2 \right] 
+ \mu \left[ \left( \frac{\partial u_2'}{\partial x_1} + \frac{\partial u_1'}{\partial x_2} \right)^2 + \left( \frac{\partial u_3'}{\partial x_1} + \frac{\partial u_1'}{\partial x_3} \right)^2 + \left( \frac{\partial u_3'}{\partial x_2} + \frac{\partial u_2'}{\partial x_3} \right)^2 \right] \] (3)

The specific entropy generation rate consists of two parts in turbulent flow: one is the mean term caused by average velocity, the other is the fluctuating term caused by the fluctuating velocity [30]:

\[ S_{pro,D} = S_{pro,VD} + S_{pro,TD} \] (4)

\[ S_{pro,VD} = 2\mu \left[ \left( \frac{\partial u_1'}{\partial x_1} \right)^2 + \left( \frac{\partial u_2'}{\partial x_2} \right)^2 + \left( \frac{\partial u_3'}{\partial x_3} \right)^2 \right] 
+ \mu \left( \left( \frac{\partial u_2'}{\partial x_1} + \frac{\partial u_1'}{\partial x_2} \right)^2 + \left( \frac{\partial u_3'}{\partial x_1} + \frac{\partial u_1'}{\partial x_3} \right)^2 + \left( \frac{\partial u_3'}{\partial x_2} + \frac{\partial u_2'}{\partial x_3} \right)^2 \right) \] (5)

\[ S_{pro,TD} = 2\mu \left[ \left( \frac{\partial u_1'}{\partial x_1} \right)^2 + \left( \frac{\partial u_2'}{\partial x_2} \right)^2 + \left( \frac{\partial u_3'}{\partial x_3} \right)^2 \right] 
+ \mu \left( \left( \frac{\partial u_2'}{\partial x_1} + \frac{\partial u_1'}{\partial x_2} \right)^2 + \left( \frac{\partial u_3'}{\partial x_1} + \frac{\partial u_1'}{\partial x_3} \right)^2 + \left( \frac{\partial u_3'}{\partial x_2} + \frac{\partial u_2'}{\partial x_3} \right)^2 \right) \] (6)
For a numerical analysis, the former can be calculated by the known velocity and temperature, whereas the following is related to the mean temperature and turbulent dissipation rate [31,32]:

\[ S_{pro,TD} = \frac{\rho e}{T} \]  

(7)

The total entropy generation rate of different parts can be calculated by the volume integration of each local entropy generation term:

\[ \Delta S_{pro} = \Delta S_{pro,VD} + \Delta S_{pro,TD} \]  

(8)

\[ \Delta S_{pro,VD} = \int_V S_{pro,VD} dV \]  

(9)

\[ \Delta S_{pro,TD} = \int_V S_{pro,TD} dV \]  

(10)

3. Experimental and Numerical Methods

3.1. Experimental Model

A seven-stage centrifugal pump was used as the experimental object, as shown in Figure 2a. Its main performance parameters are shown in Table 1. The main equipment of the experimental device includes the water storage tank, multi-stage centrifugal pump, motor, inlet and outlet pressure sensor, electromagnetic flowmeter, photoelectric velocimeter, control cabinet, and computer, as shown in Figure 2b. The models of the main equipment for the performance test are shown in Table 2.

![Figure 2](image-url)

(a) (b)

Figure 2. Multi-stage centrifugal pump and its experimental device: (a) entity; (b) experimental device. 1—outlet pipe valve, 2—electromagnetic flowmeter, 3—inlet pipe valve, 4—inlet pipe pressure transmitter, 5—outlet pipe pressure transmitter, 6—control cabinet, 7—computer, 8—motor, 9—photoelectric tachometer, 10—multi-stage centrifugal pump, 11—reservoir.

| Rated Flow (m³/h) | Rated Speed (rpm) | Rated Head (m) | Cavitation Allowance (m) | Shaft Power (kw) | Motor Power (kw) |
|-------------------|-------------------|----------------|--------------------------|-----------------|-----------------|
| 30                | 2980              | 252.2          | 30                       | 30              | 30              |
Table 2. The models of main equipment for performance test.

| Equipment                  | Model/Type                       |
|----------------------------|----------------------------------|
| Electromagnetic Flowmeter  | E-mag EJA430E                    |
| Resonant Pressure Transmitter | Variable-Frequency Adjusted-Speed Three-Phase Roller |
| Motor                      | Laser Doppler Velocimeter        |
| Photoelectric Tachometer   | BB5 Seven-Stage Pump             |
| Multi-Stage Centrifugal Pump |                                |

The inlet and outlet of the test device were connected with the reservoir. The inlet and outlet pipelines of the centrifugal pump were equipped with pressure transmitters, the couplings of the pump and motor were equipped with photoelectric tachometers, and the outlet pipelines were equipped with electromagnetic flow meters. The water pump comprehensive performance test system was installed on the computer. During the test, the centrifugal pump was tested for external characteristics within the range of $0 Q_d \sim 1.4 Q_d$ ($1 Q_d = 30 \text{ m}^3/\text{h}$), and the flow was adjusted by the pipe valves.

3.2. Grid Generation and Grid Independence Investigation

The original design drawings were three-dimensional and two-dimensional drawings of the whole machine and parts of the BB5 multi-stage centrifugal pump supplied by the pump plant. The main flow passages were the inlet chamber, seven impellers, seven guide vanes, and shell, as shown in Figure 3. The structure and axial section of the pump stage part is shown in Figure 3b.

![Figure 3. Fluid domain of pump: (a) fluid domain; (b) the interior of fluid domain assembly.](image)

Because of the complexity of the physical model, the unstructured tetrahedral grids were adopted in the computational zone, and were created by the software ANSYS Meshing 18.2, as shown in Figure 4. Only the grids of impeller 2 and guide vane 2 are shown due to their similarities. The mesh of the surface with the large curvature and the small gap in the pump were refined with the Curvature and Proximity size functions included in ANSYS Meshing. The grid number of each flow passage component is shown in Table 3. The total number of grids was about 9.83 million. The grid quality was above 0.2 and the average grid quality was about 0.83. The average $y^+$ of the key surfaces was reasonable for a safe limit according to the Fluent User’s Guide. Four sets of the grid scheme under the designed flow condition are shown in Table 4. The computational domain keeps proportional similarity to the grid number. The head was selected to assess the impact of the grid number on the final solution. When the total grid number was greater than 9,833,274, the change of the head was little and the change of relative error between the simulated head and test head became smaller. Thus, the total grid number of 9,833,274 was used for the calculation.
Because of the complexity of the physical model, the unstructured tetrahedral grids were adopted in the computational zone, and were created by the software ANSYS Meshing 18.2, as shown in Figure 4. Only the grids of impeller 2 and guide vane 2 are shown due to their similarities. The mesh of the surface with the large curvature and the small gap in the pump were refined with the Curvature and Proximity size functions included in ANSYS Meshing. The grid number of each flow passage component is shown in Table 3. The total number of grids was about 9.83 million. The grid quality was above 0.2 and the average grid quality was about 0.83. The average $y^*$ of the key surfaces was reasonable for a safe limit according to the Fluent User’s Guide. Four sets of the grid scheme under the designed flow condition are shown in Table 4. The computational domain keeps proportional similarity to the grid number. The head was selected to assess the impact of the grid number on the final solution. When the total grid number was greater than 9,833,274, the change of the head was little and the change of relative error between the simulated head and test head became smaller. Thus, the total grid number of 9,833,274 was used for the calculation.

### 3.3. Turbulence Model and Boundary Conditions

Considering the large curvature of the internal overflow interface of the centrifugal pump, the governing equations selected in this research were the three-dimensional Reynolds time-averaged N–S equation and RNG k-$\varepsilon$ turbulence model [33]. Compared with other turbulence models, the RNG k-$\varepsilon$ turbulence is more effective for determining the flow with a high strain rate and large streamline curvature.

The three-dimensional Reynolds-averaged N–S equation is as follows:

$$\frac{\partial \rho u_i}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} = -\frac{1}{\rho} \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left( \nu \frac{\partial u_i}{\partial x_j} - u_i' u_j' \right)$$

(11)

The RNG k-$\varepsilon$ model is identical to the standard k-$\varepsilon$ model in form. However, the difference is that the value of the constant is not based on experimental data, but on theoretical analysis. Therefore, in the calculation of complex flows, the RNG k-$\varepsilon$ model has higher accuracy [34]. Its equations are as follows:

$$\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left( \alpha_k \mu_{eff} \frac{\partial k}{\partial x_j} \right) + G_k + G_b - \rho \varepsilon - Y_M + S_k$$

(12)
\[
\frac{\partial (\rho e)}{\partial t} + \frac{\partial (\rho e u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left( \alpha_k \mu_{\text{eff}} \frac{\partial e}{\partial x_j} \right) + C_1 \frac{\varepsilon}{k} (G_k + C_3 G_b) - C_2 \frac{\varepsilon^2}{k} - R_\varepsilon + S_\varepsilon
\]  

(13)

where \( C_1 = 1.42 \), \( C_2 = 1.68 \), \( C_3 = 1.39 \), \( \alpha_k = \alpha_\varepsilon = 0.7179 \), \( \mu_{\text{eff}} = \mu + \mu_t \) and \( R_\varepsilon \) are terms to adapt to the rapid flow with variable rate and streamline curvature, and its expression is:

\[
R_\varepsilon = \frac{C_u \rho \eta^3 (1 - \frac{\eta}{\eta_0})^2}{1 + \beta \eta^3}
\]

(14)

The turbulent flow field of the pump was calculated by ANSYS Fluent 18.2. It was assumed that the flow was steady. The RANS equations were closed with the Reynolds stress turbulence model and standard wall function.

The inlet boundary was set as the pressure inlet according to the actual pressure. Furthermore, the hydraulic diameter was set at 40 mm and the turbulence intensity was set at 5%. The mass flow outlet was set as the outlet boundary.

All overflow areas were set as the fluid domain, and the fluid was liquid water. The impeller area was set as the rotating basin in the rotating reference system, where the rotating speed was 2980 rpm (note that the rotating direction of the impeller follows the right-hand rule), and the inlet section, guide vanes, middle section, and outlet section were the static areas.

In addition to the inlet and outlet interfaces, all walls were set as the no-slip boundary. The near-wall region was solved by the standard wall function.

The SIMPLE algorithm was used to couple the pressure and velocity. A residual of \( 1 \times 10^{-4} \) was set as the convergence criteria for the numerical analysis.

### 3.4. Validation of the Numerical Results

This study calculated the hydraulic performance under different flow conditions from 0.6 \( Q_d \) to 1.4 \( Q_d \) (1 \( Q_d \) = 30 m\(^3\)/h). The corresponding equations of head, shaft power, and efficiency are as follows:

\[
H = \frac{p_{\text{out}} - p_{\text{in}}}{\rho g}
\]

(17)

\[
N = M n \pi / 30
\]

(18)

\[
\eta = \eta_h \eta_m \eta_v
\]

(19)

where \( \eta_h \), \( \eta_m \), and \( \eta_v \) are hydraulic efficiency, mechanical efficiency, and volumetric efficiency, respectively. Their formulas and empirical expressions are as follows [35]:

\[
\eta_h = \frac{\rho g Q H}{M n \omega}
\]

(20)

\[
\eta_m = 1 - 0.07(0.01 n_s)^{-1.18}
\]

(21)

\[
\eta_v = 1 - 0.028(0.01 n_s)^{-0.6}
\]

(22)

The main flow field simulation results at 2980 rpm were plotted to obtain the curves of \( H-Q \), \( N-Q \), and \( \eta-Q \) and were compared with the experimental results, as shown in Figure 5. The simulated head, shaft power, and efficiency are basically the same as the experimental results. Under the design condition \( (Q = 30 \text{ m}^3/\text{h}) \), the head error is 6.7%, the shaft power error is 5.7%, and the efficiency error is 5.2%. The pump efficiency is the product of hydraulic efficiency, mechanical efficiency, and volumetric efficiency, as shown
in Equation (13), but only hydraulic losses were used in the determination of the simulated pump efficiency; thus, there is a deviation between the simulation and the experimental results. When the flow rate is greater than 36 m$^3$/h, the error increases gradually. The reason is that the compressibility of the system increases as the high-speed rotating fluid speed in the multi-stage centrifugal pump increases. The compressibility of fluid in the multi-stage centrifugal pump has a certain impact on the mechanical loss and volume loss of the pump. In addition, air bubbles generated by cavitation also cause energy loss of the pump. However, only the incompressible system was considered and the cavitation effect of the pump was also not considered in this paper, thus, the deviation was larger. However, the deviation was still within 10%. These results show that the simulation results agree well with the experimental results.

![Graph showing comparison of numerical and experimental results.](image)

**Figure 5.** Comparison of numerical and experimental results.

4. Results and Discussion

4.1. Entropy Generation of Different Components under the Rated Flow Condition

The entropy generation of each channel component under the rated flow condition (30 m$^3$/h) is shown in Figure 6. Figure 6a shows that the guide vanes of the centrifugal pump are the primary areas of entropy generation, accounting for 71.2%. Entropy generation of guide vane 1 to guide vane 6 is roughly the same, and entropy generation of the last-stage guide vane is relatively low compared with others because there are no middle components in the last-stage guide vane. Entropy generation of the inlet chamber, shell, and impeller is roughly the same. Impellers account for 23.3% of the total entropy generation and impeller 1 has the highest entropy generation. It is suggested that the primary reason for energy loss is the guide vanes, followed by the impellers. Figure 6b shows that entropy generation of the guide vanes is about 9 W/K. Guide vane 1 has the largest entropy generation value, which is about 1.6 W/K. Additionally, the average entropy generation of the impellers is about 0.5 W/K, but its total value cannot be ignored.

Figure 7 displays the entropy generation distribution of the axial section under the rated flow condition. The figure suggests that the entropy generation is mainly concentrated in the impeller inlet, impeller outlet, and positive guide vane, which suggests the hydraulic loss mainly occurs in these places and the entropy generation method is helpful to accurately calculate the location and size of the energy loss.
The smaller the entropy production, the higher the efficiency. Secondly, guide entropy generation in the inlet chamber decreases with the increase in flow, and increases in the outlet chamber. The variation amplitude of the outlet chamber is larger than that of other overcurrent parts.

Additionally, the entropy generation in the inlet chamber decreases with the increase in flow rate. It illustrates that the bigger the flow rate, the smaller the entropy generation and impeller 1 has the highest entropy generation. It is suggested that the entropy generation of guide vane cannot be reduced by simply changing the flow conditions, and it needs to be optimized. Besides, entropy generation of impeller 1 and impeller 2 descends gradually as the flow rate increases. The minimum value is achieved at 1.1 \( Q_d \), which is opposite to the variation trend of the efficiency curve in the characteristic curve. This shows that there is a correlation between entropy generation and efficiency of the centrifugal pump.

4.2. Entropy Generation of Different Components under Different Flow Conditions

The typical overcurrent parts of a multi-stage centrifugal pump include the inlet chamber, outlet chamber, impellers, and guide vanes, which are shown in Figure 8. The results of impellers 3–6 and guide vanes 2–6 are not shown because there are similarities in their structures.

Figure 9 displays the entropy generation of typical overcurrent parts under different flow conditions (0.6 \( Q_d \) ~ 1.4 \( Q_d \)). Firstly, the total entropy generation first descends and then ascends as flow rate increases. The minimum value is achieved at 1.1 \( Q_d \), which is the total entropy generation ratio; (b) entropy generation value.

Figure 7 displays the entropy generation distribution of the axial section under the rated flow condition. The figure suggests that the entropy generation is mainly alleviation under different flow conditions. When the flow rate is greater than 1.1 \( Q_d \), the entropy production increases slightly. It shows that the energy loss of the centrifugal pump are the primary areas of entropy generation, accounting for 71.2%. Entropy generation of overcurrent parts is about 0.5 W/K, but its total value cannot be ignored.

Figure 9 displays the entropy generation of typical overcurrent parts under different flow conditions. The entropy generation of impeller 1 to guide vane 6 is roughly the same, and entropy generation ratio of the last-stage guide vane is relatively low compared with others because there are no middle components in the last-stage guide vane. Entropy generation of the inlet chamber, outlet chamber, impellers, and guide vanes, which are shown in Figure 8. The typical overcurrent parts are the inlet chamber, outlet chamber, impellers, and guide vanes, which are shown in Figure 8. The entropy generation distribution of the axial section at rated flow condition.
in the outlet chamber. The variation amplitude of the outlet chamber is larger than that of other overcurrent parts.

![Figure 8](image.png)

**Figure 8.** The typical overcurrent parts.

4.3. Entropy Generation of the Impellers and Guide Vanes

Only the results of impeller 2 and guide vane 2 were analyzed because there are similarities in the entropy generation distribution of all impellers and guide vanes.

Figure 10 displays the entropy generation of impeller 2 and guide vane 2 in the middle section. On the whole, the entropy generation of the impeller is distributed in a central symmetry. Additionally, the impeller inlet, guide vane tongue, and the channel of the guide vane have relatively more entropy generation compared with others.

Figure 11 displays the streamlines of impeller 2 and guide vane 2. It can be observed that the streamlines in the impeller are smoother compared with the guide vane. The centrifugal pumps add mechanical work to fluid through the impellers and convert it into pressure energy. The flow along the channel is subject to an adverse pressure gradient, thus flow separations occur easily, which is one of the reasons resulting in entropy generation. The high-speed rotating fluid in the impeller is discharged from the outlet and enters the diffuser of the guide vane. The flow area increases, the flow rate decreases, and some kinetic energy is converted into static energy, which further increases the fluid pressure. When the impeller outlet section rotates near the guide vane tongue, the flow rate decreases because of the impeller pushing action and the reduction in the flow area, resulting in vortices and also resulting in entropy generation.
The partial, enlarged view of the lower left corner suggests that the fluid reflux at the guide vane tongue which also increases the energy loss. This corresponds to the information of the velocity vector in Figure 11.

Moreover, the partial, enlarged view of the upper left corner suggests that the fluid enters the guide vane area with a larger flow area and impacts the wall surface, which changes that part of the kinetic energy into heat energy and static pressure, resulting in energy loss. The high-speed rotating fluid in the impeller is discharged from the outlet and enters the guide vane. The flow area increases, the flow rate decreases, and thus flow separations occur easily, which is one of the reasons resulting in entropy generation. The forward guide vane has large entropy generation and it is the primary part where energy loss occurs. The space of the area becomes larger because the high-speed fluid flows from the impeller outlet into the guide vane, which changes the fluid kinetic energy into heat energy and static pressure. The fluid impacts the outer edge of the guide vane, resulting in an increase in entropy generation. In addition, there is the reflux near the guide vane diaphragm tongue which also increases the energy loss. This corresponds to the reflux at the guide vane diaphragm tongue enters the impeller when the fluid enters the guide vane, resulting in energy loss.

Figure 10 displays the entropy generation distribution of impeller 2 and guide vane 2 in the middle section. Figure 11 displays the streamlines of impeller 2 and guide vane 2. It can be observed that the streamlines in the impeller are smoother compared with the guide vane. The central symmetry. Additionally, the impeller inlet, guide vane tongue, and the channel of the guide vane have relatively more entropy generation compared with others.

Figure 12 displays the velocity vectors of guide vane 2 in the middle section. The fluid at the impeller outlet enters the positive guide vane, then passes through the reverse guide vane, and finally flows to the next-stage impeller. Due to the diffuser deceleration, the flow velocity of the reverse guide vane is lower compared with the positive guide vane. Moreover, the partial, enlarged view of the upper left corner suggests that the fluid enters the guide vane area with a larger flow area and impacts the wall surface, which changes that part of the kinetic energy into heat energy and static pressure, resulting in energy loss. The partial, enlarged view of the lower left corner suggests that the fluid reflux at the guide vane diaphragm tongue enters the impeller when the fluid enters the guide vane, resulting in energy loss.

Figure 13 displays the entropy generation distribution of guide vane 2. It shows that the forward guide vane has large entropy generation and it is the primary part where energy loss occurs. The space of the area becomes larger because the high-speed fluid flows from the impeller outlet into the guide vane, which changes the fluid kinetic energy into heat energy and static pressure. The fluid impacts the outer edge of the guide vane, resulting in an increase in entropy generation. In addition, there is the reflux near the guide vane tongue which also increases the energy loss. This corresponds to the information of the velocity vector in Figure 11.
Figure 12. Velocity vectors of guide vane 2 in the middle section.

Figure 13. Entropy generation distribution of the blade of impeller 2.

Figure 14 displays the entropy generation distribution of the blade of impeller 2. It shows that the entropy generation of the blade pressure surface is smaller compared with the suction surface because there is flow separation at the suction surface. The entropy generation of the impeller inlet area and outlet area is larger due to vortex and flow separation.

Figure 14. Entropy generation distribution of the blade of impeller 2.

5. Conclusions

A BB5 multi-stage centrifugal pump was investigated by numerical analysis using the entropy generation method with the RNG k-ε turbulence model, and its energy loss char-
acteristics were studied. The simulation was performed at the flow rate of 0.6 $Q_d \sim 1.4 Q_d$ ($1 Q_d = 30 \text{ m}^3/\text{h}$). Additionally, the accuracy of the numerical simulation was verified with the performance comparison of the numerical results and experimental results.

The main conclusions are as follows:

- Entropy generation of the guide vanes and impellers accounted for 71.2% and 23.3% of the total entropy generation at a flow rate of 1.0 $Q_d$, respectively. The entropy generation of the guide vanes is the biggest entropy generation at about 9 W/K, thus the guide vanes are the main hydraulic loss domains, followed by impellers.
- There are many vortices at the guide vane inlet, which are located on the tongue of the guide vane. Additionally, there are flow separations in the impeller. The fluid impacts the outer edge of the guide vane, which leads to the increase in entropy generation. Furthermore, there is the reflux near the guide vane tongue, which also increases entropy generation.
- The entropy generation of the positive guide vane and blade suction surface is greater compared with the reverse guide vane and pressure surface, respectively.

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References

1. Holbein, P.; Dupont, P. CFD Calculation of a Mixed Flow Pump Characteristic from Shutoff to Maximum Flow. J. Fluids Eng. 2002, 124, 798–802. [CrossRef]
2. Cao, S.; Peng, G.; Yu, Z. Hydrodynamic Design of Rotodynamic Pump Impeller for Multiphase Pumping by Combined Approach of Inverse Design and CFD Analysis. J. Fluids Eng. 2004, 127, 330–338. [CrossRef]
3. Guleren, K.M.; Pinarbasi, A. Numerical simulation of the stalled flow within a vaned centrifugal pump. ARCHIVE Proc. Inst. Mech. Eng. Part C J. Mech. Eng. Sci. 2004, 218, 425–435. [CrossRef]
4. Jafarzadeh, B.; Hajari, A.; Alishahi, M.M.; Akbari, M.H. The flow simulation of a low-specific-speed high-speed centrifugal pump. Appl. Math. Model. 2011, 35, 242–249. [CrossRef]
5. Zhou, P.J.; Wang, F.J.; Yang, M. Internal flow numerical simulation of double-suction centrifugal pump using DES model. IOP Conf. Ser. Earth Environ. Sci. 2012, 15, 41–51. [CrossRef]
6. Feng, J.; Luo, X.; Guo, P.; Wu, G. Influence of tip clearance on pressure fluctuations in an axial flow pump. J. Mech. Sci. Technol. 2016, 30, 1603–1610. [CrossRef]
7. Zhu, D.S.; Ji, G.F.; Cheng, W.Q. Effect of Multi Stage Centrifugal Pump on the Guide Vane Based on CFD Simulation. J. North Univ. China Nat. Sci. Ed. (China) 2017, 38, 322–326, 332. [CrossRef]
8. Bejan, A.; Kestin, J. Entropy Generation Through Heat and Fluid Flow. J. Appl. Mech. 1983, 50, 475. [CrossRef]
9. Spurk, J.H. Fluid Mechanics; Springer: New York, NY, USA, 1997; pp. 67–74.
10. Kock, F.; Herwig, H. Entropy production calculation for turbulent shear flows and their implementation in CFD codes. Int. J. Heat Fluid Flow 2005, 26, 672–680. [CrossRef]
11. Zhang, H.C.; Schmandt, B.; Herwig, H. Determination of Loss Coefficients for Micro-Flow Devices: A Method Based on the Second Law Analysis (SLA). In Proceedings of the ASME 2009 2nd Micro/Nanoscale Heat & Mass Transfer International Conference MNHMT, Shanghai, China, 18–21 December 2009; Volume 2.
12. Chu, S.X.; Liu, L.H. Entropy generation analysis of two-dimensional high-temperature confined jet. Int. J. Therm. Sci. 2009, 48, 998–1006. [CrossRef]
13. Hou, H.; Zhang, Y.; Li, Z. A numerical research on energy loss evaluation in a centrifugal pump system based on local entropy production method. Therm. Sci. 2016, 2016, 143. [CrossRef]
14. Hou, H.; Zhang, Y.; Li, Z.; Jiang, T.; Zhang, J.; Xu, C. Numerical analysis of entropy production on a LNG cryogenic submerged pump. *J. Nat. Gas Sci. Eng.* **2016**, *24*, 36–85. [CrossRef]

15. Li, X.; Jiang, Z.; Zhu, Z.; Si, Q.; Li, Y. Entropy generation analysis for the cavitation head-drop characteristic of a centrifugal pump. *Proc. Inst. Mech. Eng. Part C J. Mech. Sci.* **2018**, *232*, 4637–4646. [CrossRef]

16. Liang, Z.; Xiang, L.; Wei, X.; Chen, S.; Liu, J.; Hao, Z. Numerical study on the flow characteristics of centrifugal compressor impeller with crack damage. *Adv. Mech. Eng.* **2021**, *13*, 1–15. [CrossRef]

17. Yu, A.; Tang, Y.; Tang, Q.; Cai, J.; Zhao, L.; Ge, X. Energy analysis of Francis turbine for various mass flow rate conditions based on entropy production theory. *Renew. Energy* **2022**, *183*, 447–458. [CrossRef]

18. Zhang, F. Study on Flow Loss Characteristics of Side Channel Pump Based on Entropy Production. *J. Mech. Eng.* **2018**, *54*, 137. [CrossRef]

21. Li, S.; Wu, P.; Wu, D. Hydraulic Optimization and Loss Analyses of a Low Specific-Speed Centrifugal Pump with Variable-Thickness Blades. In Proceedings of the Fluids Engineering Division Summer Meeting, Washington, DC, USA, 10–14 July 2016; p. V01BT27A003.

22. Pei, J.; Meng, F.; Li, Y.; Yuan, S.; Chen, J. Effects of distance between impeller and guide vane on losses in a low head pump by entropy production analysis. *Adv. Mech. Eng.* **2016**, *8*, 11. [CrossRef]

23. Zhang, Y.; Hou, H.; Xu, C.; He, W.; Li, Z. Application of entropy production method to centrifugal pump energy loss evaluation. *Paiguan Jixie Gongcheng Xuebao/J. Drain. Irrig. Mach. Eng.* **2017**, *35*, 277–282. [CrossRef]

24. Li, Y.; Feng, G.; Li, X.; Si, Q.; Zhu, Z. An experimental study on the cavitation vibration characteristics of a centrifugal pump at normal flow rate. *J. Mech. Sci. Technol.* **2018**, *32*, 4711–4720. [CrossRef]

25. Feng, J.; Zhang, Y.; Zhu, G.; Li, Y.; Li, W.; Luo, X. Transition process characteristics of centrifugal pump with power-off based on entropy production theory. *Trans. Chin. Soc. Agric. Eng.* **2020**, *36*, 10–17.

26. Koranteng Osman, M.; Wang, W.; Yuan, J.; Zhao, J.; Wang, Y.; Liu, J. Flow loss analysis of a two-stage axially split centrifugal pump with double inlet under different channel designs. *Proc. Inst. Mech. Eng. Part C J. Mech. Eng. Sci.* **2019**, *233*, 5316–5328. [CrossRef]

27. Ren, Y.; Zhu, Z.C.; Wu, D.H.; Li, X.J. Influence of Guide Ring on Energy Loss in a Multistage Centrifugal Pump. *J. Fluids Eng.-Trans. Asme* **2019**, *141*, 061302. [CrossRef]

28. Wu, T.X.; Wu, D.; Zhang, T.; Huang, H.M.; Wu, Y.Z.; Mou, J. Influence of Diffuser Vane Number on Energy Loss of Multistage Centrifugal Pump. *Int. J. Fluid Mach. Syst.* **2021**, *14*, 373–382. [CrossRef]

29. Spurk, J.H.; Aksel, N. *Strömungslehre*; Springer: Berlin/Heidelberg, Germany, 1989; Volume 4.

30. Herwig, H.; Kock, F. Direct and indirect methods of calculating entropy generation rates in turbulent convective heat transfer problems. *Heat Mass Transf.* **2007**, *43*, 207–215. [CrossRef]

31. Callenaere, M.; Franc, J.-P.; Michel, J.-M.; Riondet, M. The cavitation instability induced by the development of a re-entrant jet. *J. Fluid Mech.* **2001**, *444*, 223–256. [CrossRef]

32. Kock, F.; Herwig, H. Local entropy production in turbulent shear flows: A high-Reynolds number model with wall functions. *Int. J. Heat Mass Transf.* **2004**, *47*, 2205–2215. [CrossRef]

33. Sheng-Bo, H.U.; Mo-Ming, S.U. Comparison of Different Turbulence Models for Numerical Simulation of Internal Flow of Centrifugal Compressor. *Fluid Mach.* **2010**, *38*, 22–26.

34. Zhang, M.; Shen, Y. Application of 3-D RNG k-ε turbulence model of meandering river. *J. Hydroelectr. Eng.* **2007**, *26*, 86–91.

35. Zhang, J.; Xu, C.; Zhang, Y.; Zhou, X. Quasi-3D hydraulic design in the application of an LNG cryogenic submerged pump. *J. Nat. Gas Sci. Eng.* **2016**, *29*, 89–100. [CrossRef]