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# Investigation on inner flow quality assessment of centrifugal pump based on Euler head and entropy production analysis

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**Abstract.** The quality of flow field inside centrifugal impeller is significant for the performance of centrifugal pump. The hydraulic performance and pump stability can be improved by means of optimizing the flow field. Numerical simulations are made to study the flow field using SST  $k-\omega$  as turbulent model. In this study the inner flow quality of the model centrifugal pump is assessed in the light of energy distribution and entropy production. The Euler head is employed to measure energy of fluid obtained from impeller. The fluid energy obtaining process is characterized by Euler head distribution in streamwise location. The distribution uniformity of Euler head is also a measure to evaluate flow field quality which can be used to predict pump fluctuation performance. Meanwhile, entropy production analysis is also introduced to investigate the model pump in the study. The energy loss inside pump is denoted as entropy production and its distribution can be predicted based on numerical results. Analysis on entropy production can help provide basis for improving pump efficiency from another perspective.

## 1. Introduction

Centrifugal pump is a complex flow system with a non-homogeneous spatial distributed and transient flow field. As is influenced by Coriolis force and rotor-stator interaction etc., the inner flow field of centrifugal pump is exhibiting a strong non-uniformity and instability. Its feature is difficult to be described by conventional parameters. In recent years researchers have introduced various ways to describe the key parameters in centrifugal pump then assess the performance of impeller in a different perspective. Yan [1,2] employed Euler head to describe the process that fluid obtains energy from centrifugal impeller and gave a pattern of Euler head distribution along streamwise location to ensure a good performance on efficiency and fluctuation. Entropy production rate can be extracted from CFD simulation result and it can help locating energy loss in pump accurately. Sciubba [3] stated the advantages of entropy generation analysis over traditional methods in the energy loss assessment of fluid machinery. Yoon [4] displayed the energy loss in different positions of turbine flow passage by entropy generation rate. Copeland [5] used entropy analysis to investigate energy loss of a double-entry turbine. Newton [5,6] divided impeller zone of a turbocharger into different parts and investigated the contributions of each part on entropy generation rate. Zhang [7] employed entropy generation rate to study the energy loss of a centrifugal fan under stall condition. Li [8] applied entropy analysis method on measuring and locating energy loss inside a low specific speed centrifugal pump.



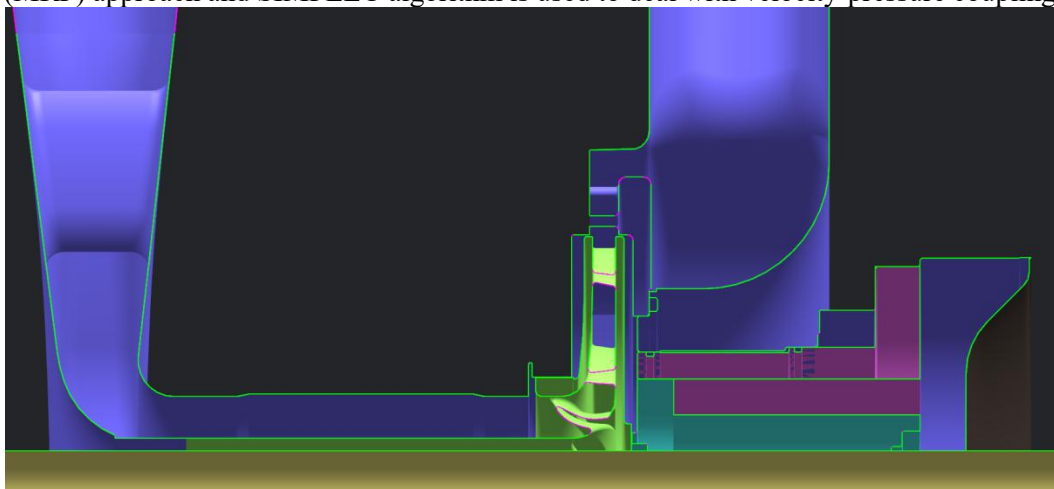
In this study, the inner flow quality of the model pump is investigated by based on the methods mentioned above. Euler head is employed to describe energy of fluid inside the impeller and its distribution is interpreted through further analysis. Entropy generation rate analysis is another method used to locate the energy loss in the impeller. The numerical simulations are conducted by the popular commercial CFD package, Fluent. The entropy production equation using to analyze the energy loss in the present study is simplified to fit the pump case.

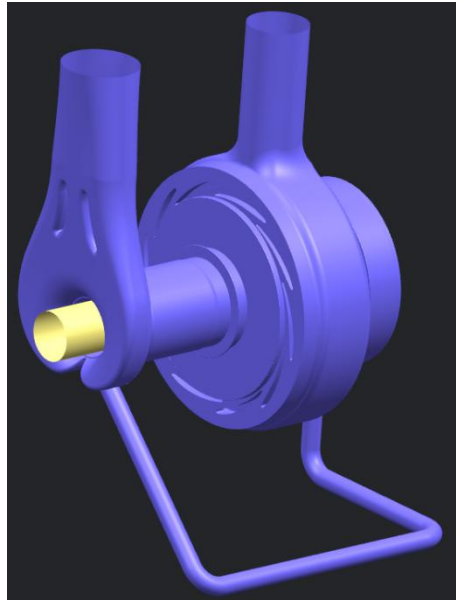
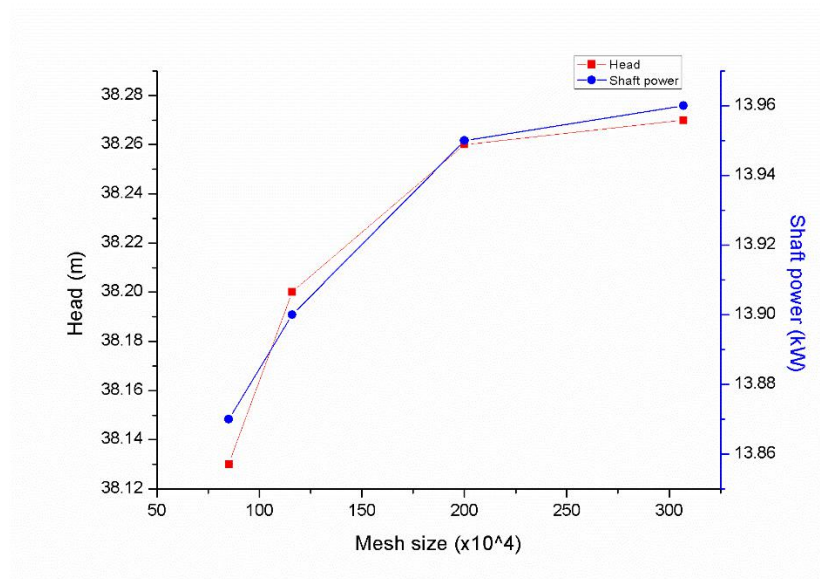
## 2. Model pump and numerical methods

The pump model used in the study is provided by the workshop Single- & Multi-Stage Pump Flow Prediction of IAHR 2018, Kyoto. Figure 1 shows the overview of the model pump investigated in this study. It was a three stages centrifugal pump consisting of a suction, three identical impellers and diffusers, reflux tube, balancing drum and discharge. In order to simplify numerical calculation, the former two impellers and diffusers are removed with a tube instead. The third impeller and diffuser of the three stages pump is used as model pump in the study. The specific speed of the pump is  $n_s = \frac{3.65n\sqrt{Q}}{H^{0.75}} = 60$  (m, m<sup>3</sup>/s, min<sup>-1</sup>). The number of impeller blades is 7 and the number of diffuser vanes is 10. The design flow rate of pump is 90 (m<sup>3</sup>, h<sup>-1</sup>), the design head is 38 (m), and the rotational speed is 1600 (min<sup>-1</sup>).

The grids of the model are generated using the commercial software package ANSYS ICEM. The suction, impeller, diffuser, balancing drum and reflux tube are filled with tetrahedral grids. The side wall gaps are filled with hexahedron grids. Considering the computing power of server, the total number of grid cells in the study is 6.8 million with 3 million grids in the impeller region. The mesh sensitivity is examined in fig. 2. When impeller grid number reaches 2 million, the head and shaft power barely change. The grids around the blades and other important walls are dealt carefully to get the reasonable boundary layer results.

The steady SST k- $\omega$  model is selected to simulate the incompressible flow in this case. The inlet is set as velocity inlet and the outlet is set as pressure outlet for boundary conditions. The multiple reference frame (MRF) approach and SIMPLEC algorithm is used to deal with velocity-pressure coupling.



**Figure 1.** Pump model**Figure 2.** Mesh sensitivity

### 3. Results and discussions

#### 3.1. Euler head distribution analysis

The fluid receives energy from impeller blade when running through the flow passage. Its theoretical head is defined by Eq. (1) [9].

$$H_{th} = \frac{u_2 c_{2u} - u_1 c_{1u}}{g} \quad (1)$$

$u_1$  and  $u_2$  are circumferential velocity of fluid at impeller inlet and outlet.  $c_{1u}$  and  $c_{2u}$  are circumferential component of fluid absolute velocity at impeller inlet and outlet. The definition of Euler head is simplified based on Eq. (1). It is assumed that the fluid flow to the impeller inlet is mostly axial. The circumferential component of fluid absolute velocity is therefore  $c_{1u} = 0$ . When local Euler head is applied to describe local energy the term  $u_1 c_{1u}$  can be removed, so the local Euler head is defined as Eq. (2)

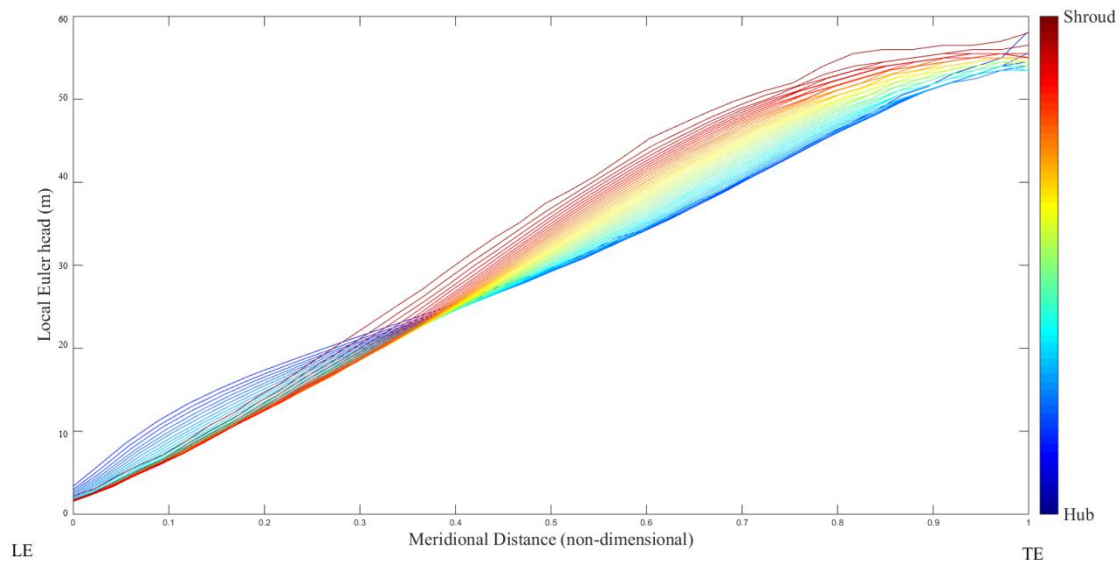
$$\text{Local Euler Head} = \frac{UV_\theta}{g} \quad (2)$$

$U$  is the local circumferential velocity of fluid and  $V_\theta$  is the circumferential component of local absolute velocity.

The simulation is conducted at the design flow rate and the post analysis are made based on the simulation result. Figure 3 shows the local Euler head distribution from leading edge (LE) of impeller blade to trailing edge (TE) in s1 stream surfaces (the surfaces in the impeller with constant value of span). The difference of distribution from impeller hub and shroud is also displayed in fig. 3. The relation between the pressure difference of two surfaces, pressure surface and suction surface, of blade and stream derivative of local Euler head is shown in Eq. (3) [10]. The variables  $p^+$  and  $p^-$  relate to the pressure on pressure and suction side of the blade,  $\omega$  is the angular velocity of the impeller,  $\bar{V}_m$  is the circumferential averaged streamwise component of the absolute velocity. LEH is the abbreviation of local Euler head and its streamwise derivative of the is the slope of local Euler head curve in fig. 3. The equation is based on some assumptions but still acceptable for predicting the tendency of pressure change on the blade. As is observed in fig. 3, the LEH continues rising along the whole range of streamwise location. The slope of the ribbon is generally constant, which indicates that the blade loading distribution is generally even. While there are differences of LEH distribution on the spanwise location. The LEH near hub is higher than that near shroud in the streamwise location between 0 to 0.4 and the situation completely reverses in the range between 0.4 to 1. The slopes of different span curves are different. Slopes of curves near hub is high in the first 10% meridional distances and slightly slide down from 0.1 to 0.5 area and then raise again in the last 50% part. The slopes of curves from shroud to even span 0.5 area is rather different that they remain high level in the first 60% and then gradually decline. The energy distributions of the s1 stream surfaces are displayed in this way and optimizations can be conducted by improving the uniformity of energy distribution. The ribbon also tends to diverge in the last half of the streamwise position, it can be explained that in this area the flow passage becomes larger on circumferential direction when it approaches TE. The flow field is more difficult to be controlled by the blades in the wider passage.

Figure 4 gives a more direct perspective of Euler head distribution contours at span of 0.2, 0.5 and 0.8. The hub and shroud are at span of 0 and 1 respectively. The Euler head distributions near hub and shroud are less uniform than that in the span 0.5 surface. It is because the influence of hub and shroud wall. There are areas of low Euler head near the outlet of impeller. It may be caused by the wake flow near blade trailing edge as well as the interaction between impeller blades and diffuser blades. These low Euler head areas will aggravate the non-uniformity of circumferential energy distribution at impeller outlet, which result in the pump pressure fluctuation and the vibration performance will be affected. Measures can be taken to improve hydro induced vibration by improving circumferential distribution uniformity of Euler head.

$$p^+ - p^- = \frac{2\pi \rho g}{Z_b \omega} \bar{V}_m \frac{\partial LEH}{\partial m} \quad (3)$$



**Figure 3.** Local Euler head distribution on s1 stream surfaces

### 3.2. Entropy production analysis

The hydraulic loss of pump is usually evaluated by total pressure distribution analysis, which is difficult to reach a further concept of energy loss assessment. While energy loss in pumps can be explained as entropy production. Therefore it is better to describe energy loss with entropy production based on the assumption that centrifugal pumps are almost adiabatic [8].

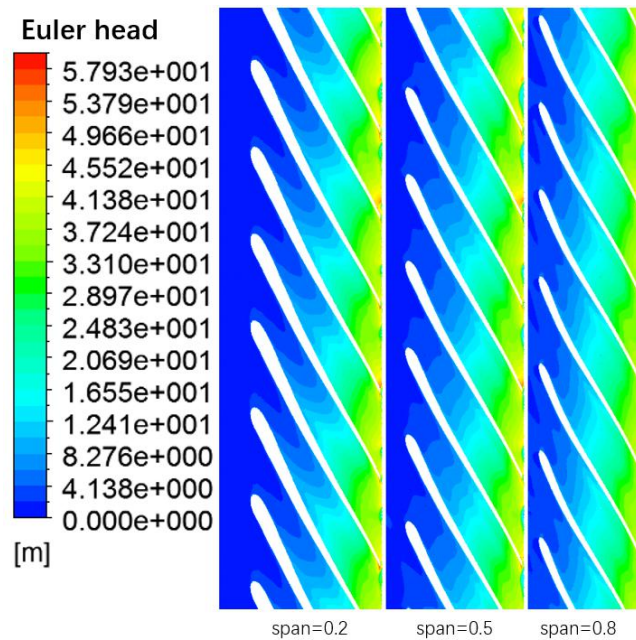
Woods [11] gave the equation for entropy production rate per unit volume with gradients of temperature and velocity, which is shown in Eq. (4).  $T$  is the fluid temperature and  $\sigma$  is the entropy production rate. Given that the fluid compressibility is negligible, the first term on the right side of Eq. (1) is the entropy produced by heat flux, which can be ignored because of the adiabatic assumption. The second term is entropy from the dissipation of mechanical energy because of the viscous effects. However, the variables in the equation are instantaneous value which are not compatible with RANS models.

$$T\sigma = -\frac{q}{T} \cdot \nabla T + \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial u_i}{\partial x_j} \quad (4)$$

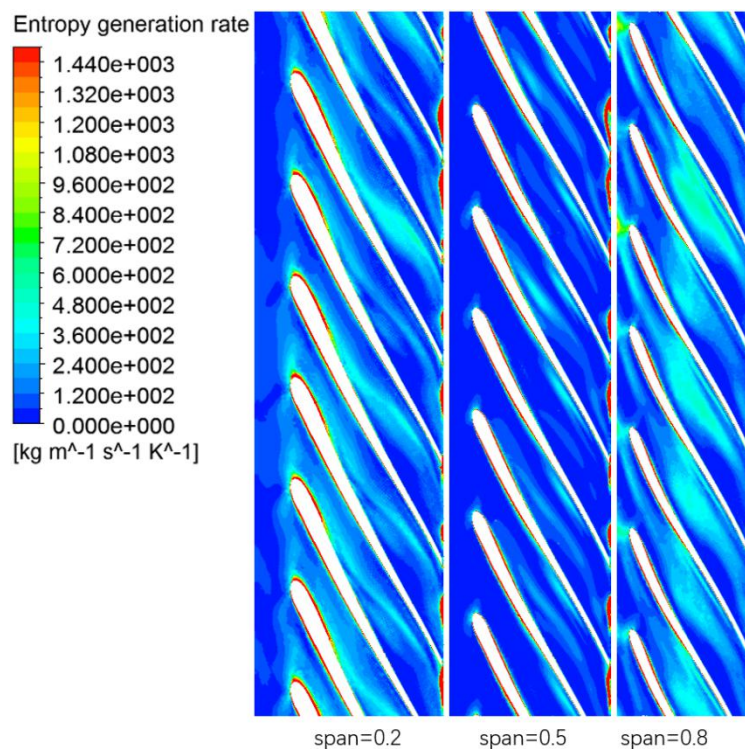
Moore [12] decomposed the second term and provided a processed equation time-averaged variables and eddy viscosity, making it compatible and practicable with RANS models. The equation is shown in Eq. (5).  $\Psi$  is the energy loss rate,  $\bar{T}$  and  $\bar{\sigma}$  are time-averaged temperature and entropy production rate,  $\mu_t$  is eddy viscosity and  $\mu$  is kinematic viscosity. In pumps the temperature is constant and the entropy production rate is easy to be extracted from CFD software.

$$\Psi \equiv \bar{T}\bar{\sigma} = (\mu_t + \mu) \left( \frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) \frac{\partial \bar{u}_i}{\partial x_j} \quad (5)$$





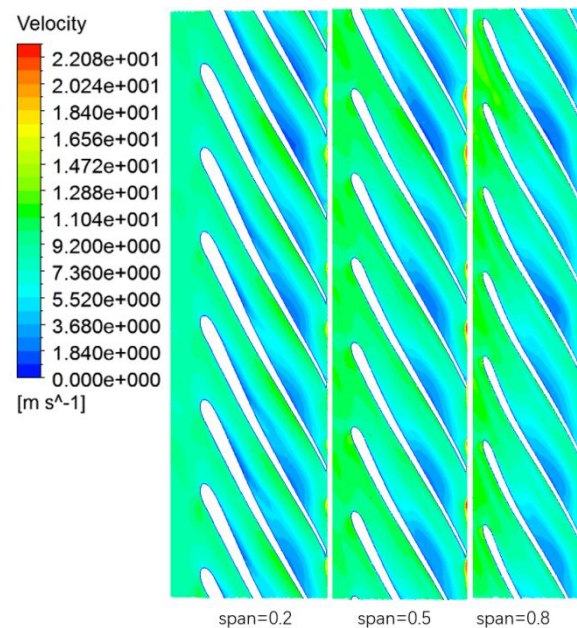
**Figure 4.** Local Euler head distribution contour on different span surfaces



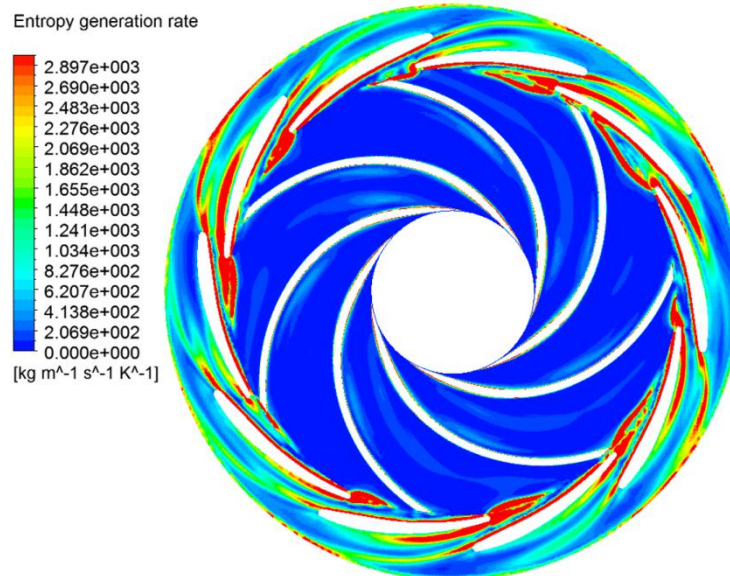
**Figure 5.** Entropy generation on different span surfaces

The entropy generation rate distribution on sl stream surfaces with different span is shown in fig. 5. It is noticeable that the entropy generation rate near blade wall is obviously higher, which is due to wall friction. The region near impeller outlet is sprinkled with high entropy production rate areas, which coincide with low Euler head areas in fig. 4. This corresponding evidence helps proving that the uneven distribution of Euler head near impeller outlet results in extra energy loss. The cause of energy loss in these areas is the wake flow of trailing edge and rotor-stator interaction. With the entropy generation rate distribution given in fig. 5, it should be noted that the pump efficiency performance

can be optimized by improving the Euler head distribution uniformity. There are also areas with energy loss in the middle part of the flow passage in impeller. They can be explained by the definition of energy loss rate in Eq. (5). There is more energy loss where the velocity gradient change is more obvious. As is shown in fig. 6, the relative velocity contours are related to the distribution of entropy generation in fig. 5. The areas of middle flow passage with energy loss at span of 0.2 and 0.8 are larger than that at span of 0.5. It is because that the influence of hub and shroud wall help expanding the area of strong velocity gradient.



**Figure 6.** Relative velocity contour on different span surfaces



**Figure 7.** Entropy generation in vaned diffuser

The data of entropy generation rate can also be extracted from other parts of pump. Figure. 7 shows the entropy generation in vaned diffuser as well as impeller. It can be noticed that the high entropy generation rate areas in vaned diffuser are larger, which means that the energy loss in vaned diffuser is apparently more compared with impeller. According to fig. 7, the energy loss mainly occurs near the diffuser vane wall and its wake flow area because of the wall friction and vortex generated there.



#### 4. Conclusion

In this study, Euler head distribution analysis and entropy production analysis are employed to assess the inner flow quality of the model centrifugal pump. The distribution of local Euler head in impeller can reflect the pattern of energy growth directly. Its circumferential distribution uniformity at impeller outlet should be concerned because uneven energy distribution brings extra energy loss and hydro-induced vibration. In order to locate the energy loss in the pump, entropy production analysis is also introduced. The energy loss mainly occurs near blade wall, diffuser vane wall, rotor-stator interaction influenced area and diffuser vane wake flow area. The strong velocity gradient in the middle part of flow passage also brings energy loss.

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