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CFD analysis of the hydrodynamic characteristics of an eccentric shaft in a canned motor

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Abstract. Boiler water circulation pump is the core of the 600MW and above supercritical boiler start system and the subcritical boiler furnace forced circulation system. A pressure fluctuation with the same frequency as shaft rotation was found during the use of a boiler water circulation pump at its outlet. Based on an engineering problem, the effects of eccentricity with axial flow on the hydrodynamic characteristics of the Taylor-Couette system is numerically studied in a simplified shaft system of canned motor, by using RNG k- ϵ turbulence model to solve the near-wall region more accurately. Sliding mesh is used to calculate the hydrodynamic balance of the shaft. Results show that the fluctuation rule of pressure in the entire shaft system is quite consistent. The pressure loss increases along the axial flow, and finally there is no pressure fluctuation at the outlet.

1. Introduction

Boiler water circulation pump is the core of 600MW and above supercritical boiler start system and the sub-critical boiler furnace forced circulation system. The boiler water circulation pump in this study is a single-stage centrifugal pump, which is composed of a pump casing, a heat barrier, a motor, connecting pipelines, etc. As shown in figure 1, the motor is connected by bolts directly under the pump casing. There is a heat barrier between the pump part and the motor part to separate the hot pump part from the cold motor part so that the heat conduction between the two parts is minimized. Then the cooling water carries out heat through the cooling chamber.

The boiler circulation pump has following features[1]:

- The inner cavity of the motor and the inner space of the pump housing are linked along the gap between the motor rotor and the pump impeller. Pressure in the motor chamber varies with that in the pump casing. The impeller of the pump is fixed on the same shaft with the rotor of the motor, which makes the two parts a single unknit and is housed in a sealed case. The stator of the motor is immersed in water and is under the operating pressure of the boiler.
- The motor is cooled by the drive of an auxiliary impeller mounted on the bearing. The circulating water enters the motor from the rear bearing of the motor, and then passes through the gap between the rotor and the stator. Finally, it flows into the external cooler. The whole cooling system forms a closed loop system.

A pressure fluctuation with the same frequency as the shaft rotation was found during the use of a boiler water circulation pump at its outlet. Preliminary examinations suggest the possibility of



multiple causes. One possible reason, which is the main topic of this paper, is described as follows. The structure of the motor shaft is shown in figure 2. Gap between the rotor and the stator is actually composed of three parts. The surface of the rotor is not in direct contact with the cooling water. Instead it is attached to the sleeve by a layer of industrial glue. Since industrial adhesives do not guarantee absolute uniformity during application, a certain degree of eccentricity may exist in the rotor, which may be the cause of pressure fluctuations. Based on the characteristics of the boiler circulation pump, it's postulated that the pressure fluctuation occurred at the motor shaft was conducted through the flow to the outlet of the pump. Based on this assumption, discussion of pressure distribution and flow field in the motor is carried out in this paper.

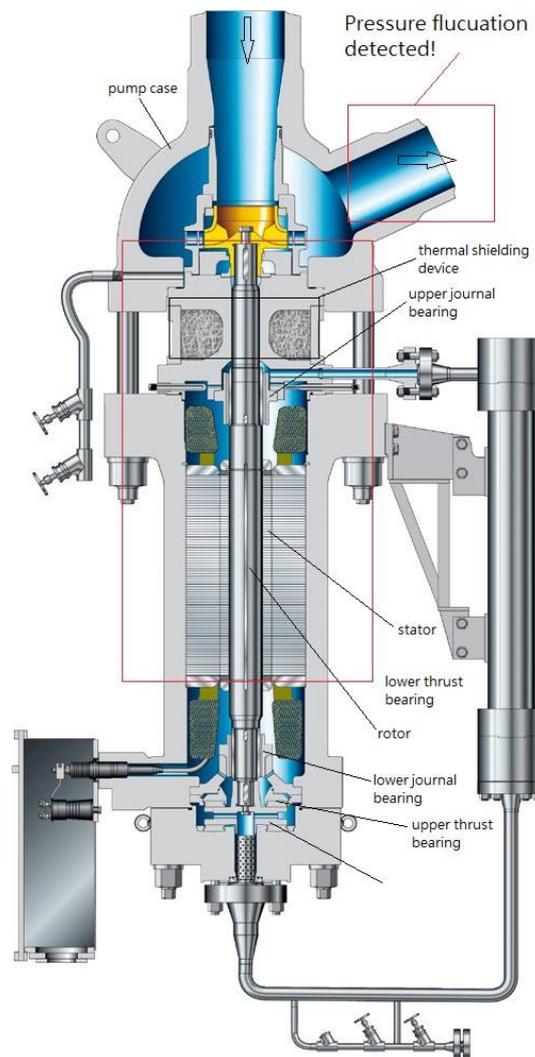


Figure 1. Structure of boiler water circulation pump.

As shown in the figure 3, the actually hydraulic model of the motor shaft is complicated, which brings certain difficulties to the numerical modeling. Based on the following criteria, the model is reasonably simplified:

- The core issue of this study is to investigate whether eccentric shaft causes pressure fluctuation.
- Previous studies have shown that the geometry of some small corners can be ignored and simplified [2].

- Meanwhile, the focus and difficulty of this issue is the study of pressure fluctuation and division of the gap or the hydro bearing.

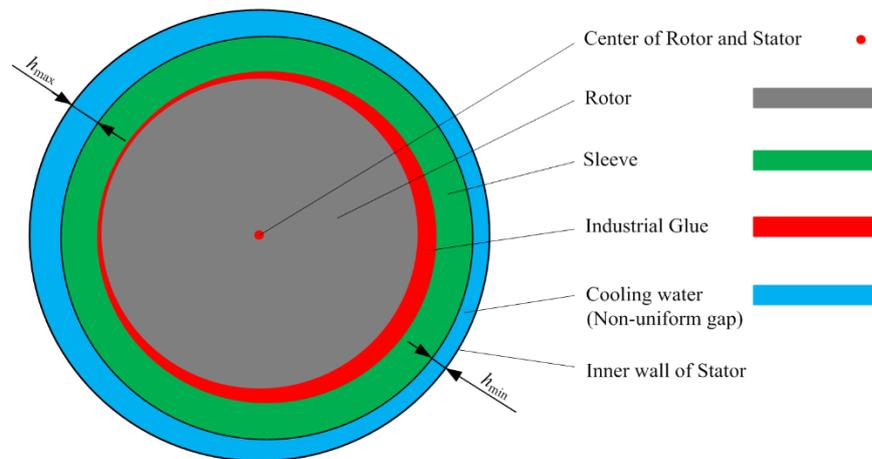
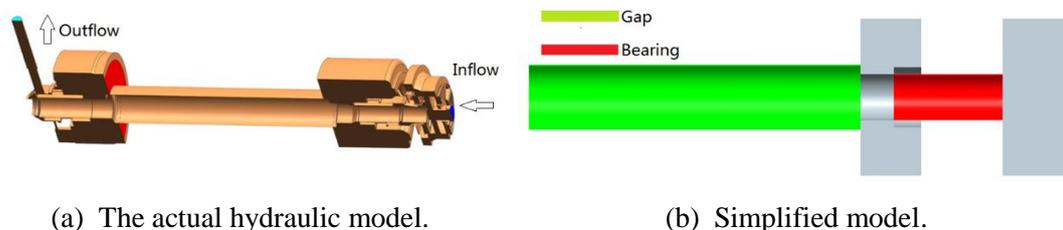


Figure 2. Cross section of the shaft.

The remaining sections of this article are organized as follows. In section Methodology, the original model of the motor, and the numerical methods adopted are described. RNG $k-\varepsilon$ turbulence model is used to solve the near-wall region more accurately. Sliding mesh is used to calculate the hydrodynamic balance of the shaft because the shape of the gap is changed as the shaft rotates. In section Results and Analysis, numerical calculations of the eccentric Taylor-Couette system with axial flow in the canned motor are presented. Finally, in section Conclusion, the summary and conclusion of this study are provided.



(a) The actual hydraulic model.

(b) Simplified model.

Figure 3. Model of the motor.

2. Methodology

To describe the complicated motor flow field accurately, the computational domain is divided into following five parts as shown in figure 4: (1) inlet housing; (2) gap; (3) hydro bearing; (4) hydro bearing housing; (5) outlet housing. The main parameters of the canned motor are listed in table 1.

Numerical calculation was carried out in two models with different eccentricities (h_{max} equals 1.6mm ($e=0.3\text{mm}$) and 1.8mm ($e=0.1\text{mm}$)) shown in figure 5. The inlet was set as pressure inlet with a pressure value of 200,000Pa. The outlet was directly connected to the atmosphere. Non-slip boundary conditions were used at the solid wall surface. The rotor wall rotated about the x-axis at 3000rpm. The fluid's material was set to water and the rest of the parameters were set as default.

In our study, quality of all grids is over 0.7. y^+ of the first layer grids has a mean value ranging from 40 to 60. Performance in two different eccentricities were compared. In order to ensure a smooth transition in the area where the cross-section changes, it is also necessary to select a suitable distribution of mesh density. In order to avoid distortion and deformation of the mesh, different node distribution rules like hyperbolic and exponential were use in this model.

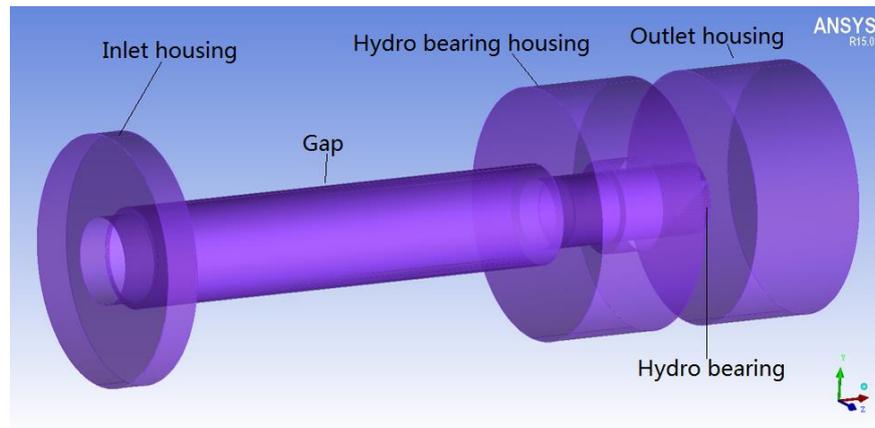


Figure 4. Computational domain of the canned motor.

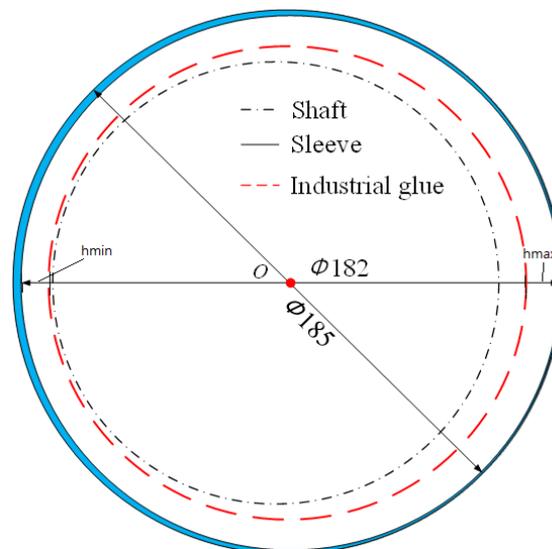


Figure 5. Instructions of parameters at the cross section.

Table 1. Parameters of the canned motor.

Parameters	Value
Axial length of inlet, hydro bearing housing and outlet	210mm
Axial length of gap	880mm
Rotor diameter	185mm
Stator diameter	182mm
Diameter of inlet outface, hydro bearing outside face or outlet	460mm
Hydro bearing outside face diameter	130mm
Hydro bearing inner face diameter	125mm
Axial length of hydro bearing	260mm
Rotating speed	3000rpm

As shown in figure 6 and 7, verification of grid independence was performed with respect to the torque along the x-axis, with total number of grid cells varied from 3 million to 10 million. It can be concluded from the figure that the degree of computational precision is satisfactory when the number of cells is more than 8 million. A mesh with about 8.10 million cells in total was chosen for the simulation. The results of time independence test were as follows in Figure 6 and 7 with respect to the average pressure of a pressure measurement point at the gap. Time step of 0.00005s was adopted in this study.

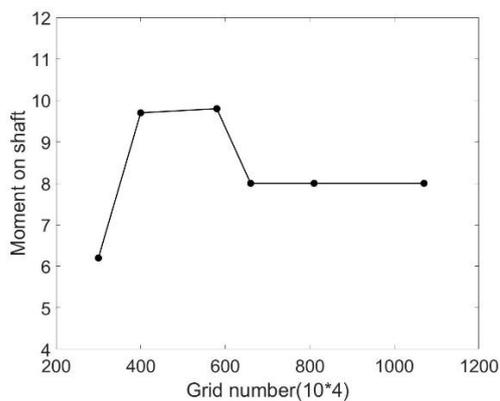


Figure 6. Grid independence verification.

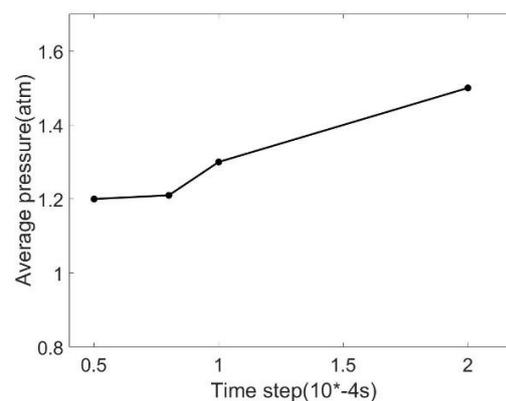


Figure 7. Time step independence verification.

While the rotor rotated around its own centre, it also revolved around the centre of the axis of rotation. Therefore, the gap would deform as the rotation of the rotor, so the grid needs to be updated at the same time or the gap and the hydro bearing must be divided in to the rotating part and the static part. Therefore, sliding mesh was adopted.

3. Result and Analysis

3.1. A subsection

The streamlines at the inlet accorded with expectation, because the cross section at the gap was much smaller than that of the inlet chamber, most of the cooling water would meet with the wall after it flows in, and then back step vortex was formed. The size of some vortex was larger than the inlet cavity, which caused the back-flow phenomenon, and the remaining cooling water flowed into the gap. Obvious step vortex phenomena also were observed at the hydro bearing housing and the outlet due to the fierce change of the cross section which can be seen in figure 8 and 9.

In figure 10, typical Taylor-Couette flow didn't appear at the gap, instead, regular spiral flow field was observed. Although Taylor number Ta in this study was about 3×10^{13} , which was larger than the critical Ta measured in Taylor-Couette system without axial flow as 3000.

The reason why Taylor vortex didn't arise may be as follows. Although Taylor number set in this calculation has far exceeded the critical Taylor number, Reynold number in the axial direction was also very large. The critical Taylor number compared in this study didn't consider the axial flow which was actually a very important parameter. At present there have been almost no studies discussing the influences of axial Reynolds number on critical Taylor number. In an article by Yuan Pingyan, it's mentioned that the valued critical Taylor number would increase when the axial flow existed. But in her article, the axial differential pressure was 1031.25Pa, much smaller than that in our study and the Taylor number studied was only at a dimension of 1000[3]. It's obviously observed that vortex would disappear when the axial flow was enhanced to a certain degree. This is because

with the increase of the differential pressure, axial flow will increase. At the same time, the role of pressure field is in the dominant position, which means the rotation effect can be neglected. Typical pressure field characteristics will be observed in the shaft. The relationship between the axial flow and the critical Taylor number is not specifically explored.

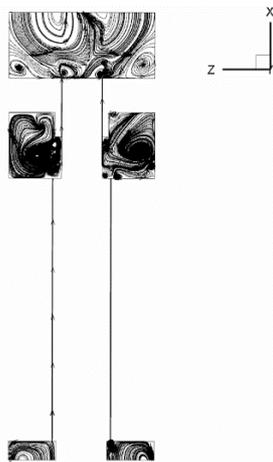


Figure 8. Streamlines(X-Z).

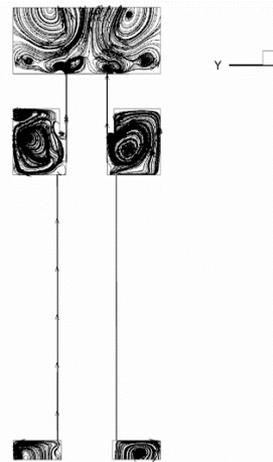


Figure 9. Streamlines(X-Y).

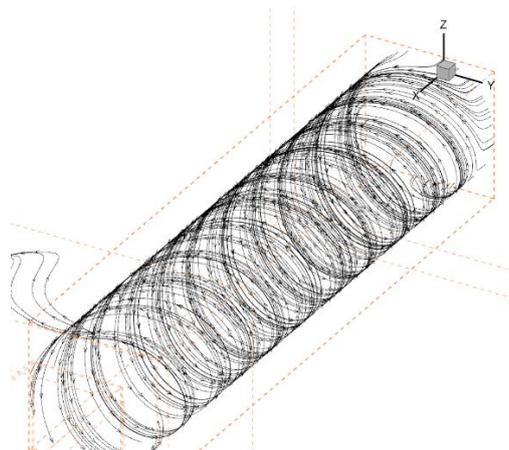


Figure 10. Flow field at the gap.

3.2. Pressure fluctuation

To better clarify pressure distribution in this problem, 12 pressure measurement points were arranged in the circumferential direction at the position of gap, hydro bearing and the outlet part. The data of the circumferential four pressure measuring points were plotted on a time domain figure 11 and 12. The law of pressure fluctuation at different measure points at gap was the same but there was a certain phase difference. This shaft, thus caused a little bit of advance or lag between the pressure signals at different points. And this was in line with the theoretical analysis, the specific phase difference is $\pi/4$.

According to the FFT, the power spectrum was obtained in figure 13 and 14. A peak appeared at $f=50\text{Hz}$ in the power spectrum, which was the same as the rotation frequency of the axis, in line with our assumption. Pressure fluctuation emerged here because of the cyclical change of the rotor gap as the rotor rotated, the entry conditions at the gap would change.

As shown in figure 15 and 16, the pressure measurement data at hydro bearing showed that pressure fluctuation also existed here. But what was different was that the frequency of pressure fluctuation was 200Hz , which was four times as that of the shaft. And there was no phase differences between

different measurement points. The reason for this phenomenon is still unexplained. More pressure measurement points on the model and data of experiments will be needed to make a reasonable explanation. The preliminary speculation is that it may be related to the complex structure of the motor cavity.

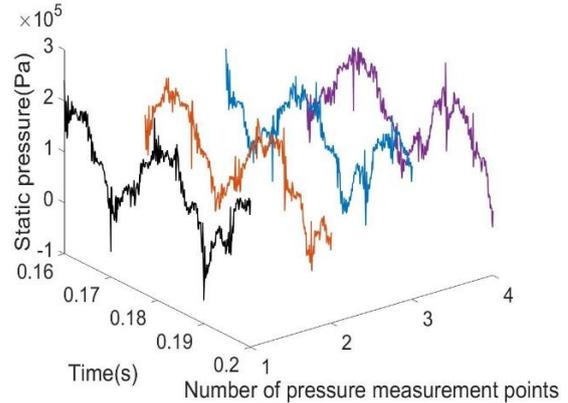
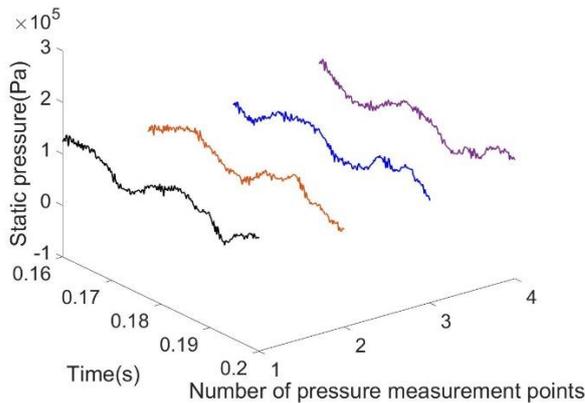


Figure 11. Pressure time history at gap ($e=0.1\text{mm}$). **Figure 12.** Pressure time history at gap ($e=0.3\text{mm}$).

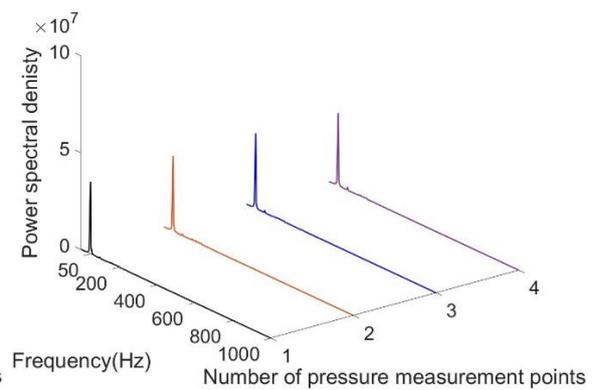
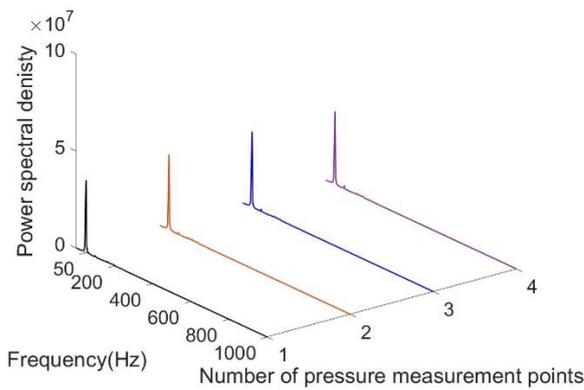


Figure 13. Power spectrum at gap ($e=0.1\text{mm}$).

Figure 14. Power spectrum at gap ($e=0.3\text{mm}$).

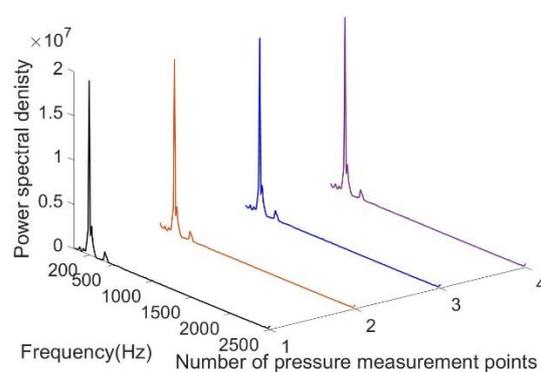
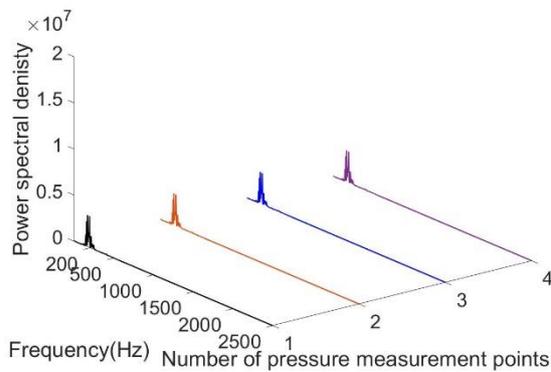


Figure 15. Power spectrum at hydro bearing ($e=0.1\text{mm}$).

Figure 16. Power spectrum at hydro bearing ($e=0.3\text{mm}$).

At the outlet, the magnitude of pressure fluctuation was already very small as seen in figure 17 and 18, and it could be assumed that there is no pressure fluctuation at this place. Therefore, the hypothesis mentioned at the beginning of the article was wrong, namely, the pressure fluctuation detected at the outlet of the pump wasn't caused by the eccentricity of shaft.

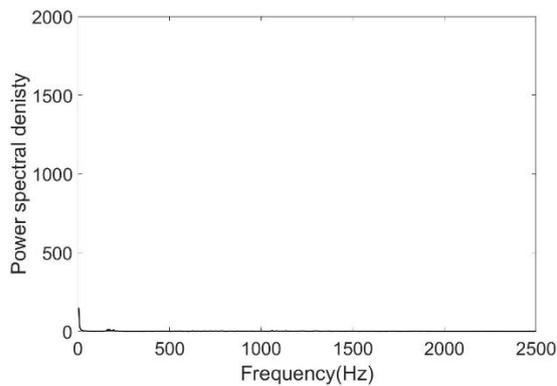


Figure 17. Power spectrum at outlet ($e=0.1\text{mm}$).

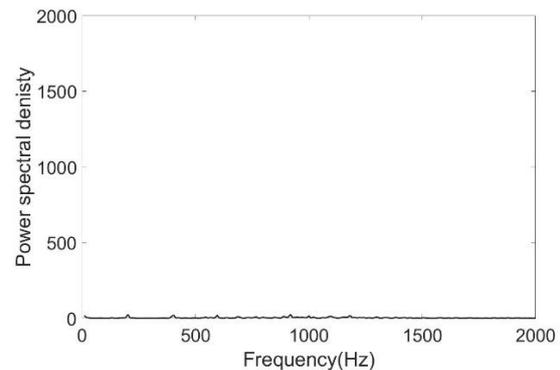


Figure 18. Power spectrum at outlet ($e=0.3\text{mm}$).

Comparing the pressure fluctuation data at different eccentricities, it's observed that at larger eccentricity, the amplitude of the pressure fluctuation was greater, indicating that the flow is more violent, but there was no difference in the frequency of pressure fluctuation.

4. Conclusion

Method of slip grid based on RNG $k-\epsilon$ under FLUENT platform was in this paper to compare and analyse flow field and pressure data of two models in different eccentricities to judge whether the hypothesis for pressure fluctuation detected at the outlet of the pump was correct.

Due to the violent transition between different parts of this model, typical back step vortex was observed. Taylor vortex didn't occur at the gap or hydro bearing as expected, although the Taylor number was much larger than the critical one given in references. This phenomenon may be due to the fact that the axial Reynolds number was too large and acted as a restraining effect for the generation of the vortex. When the axial pressure gradient is too large, the influence of the rotation of the inner wall would be ignored, the flow field in the gap exhibited typical pressure field characteristics, which indicates that axial flow can postpone the formation of Taylor vortex.

The pressure fluctuation at the gap appeared obvious cyclical, and the cycle was at the same as that of the shaft. There was a certain phase difference in the pressure fluctuation at different points on the gap. At the hydro bearing, pressure fluctuation with a frequency of four times of the rotation frequency was obtained, which is till impossible to carry out a reasonable explanation now and needs subsequent experiment results. At the outlet, the magnitude of the pressure was very small and it can be assumed that there was no pressure fluctuation here, which means that the hypothesis mentioned for the formation of pressure fluctuation detected at the outlet of the pump was wrong.

Acknowledgments

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