Investigation of the distortion suction flow on the performance of the canned nuclear coolant pump

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Keywords: canned nuclear coolant pump, distortion suction flow, head, efficiency, circumferential velocity

Abstract

In AP1000 reactor system, two canned motor pumps are directly attached to the cold side of the steam generator. In order to investigate the effect of the velocity distortion generated by the steam generator on the performance of the two pumps, CFD method is used to simulate the model pump with three different suction flows, one comes from the straight pipe and the other two come from the cold side of the steam generator. The results show that the head of the pump under the left channel head increases by 1.1% while the head under the right channel head decreases by 2.2% when compared to the straight pipe case at design point. The efficiency of the pumps under the steam generator both decreases. 3-D flow field at the inlet of the impeller and in the channel head is studied in details and the circumferential velocity induced by the complex geometry upstream in the discharge pipe of the steam generator is analysed which causes the change of the head. The results suggest that the nozzle dam brackets should be installed in the discharge pipe of the steam generator to improve the performance of the pump.

Introduction

In AP1000 reactor system, two canned motor pumps are directly attached to the cold side of the steam generator [1] (Figure 1). The pumps are identical designs and are selected based on performance under uniform inflow. But actually non-uniform sunction flow is induced in the discharge pipe due to the complex geometry in the channel head,which might influence the performance of the pumps.

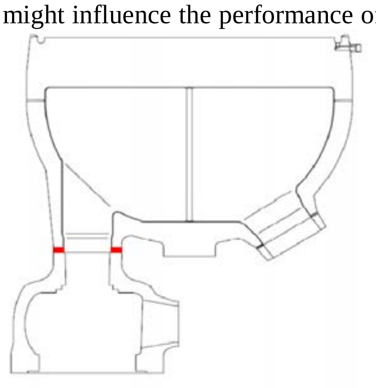

Figure 1. Schematic diagram of connection type

Up to now, only few researchers have studied the effect of the non-uniform inflow on the performce of the pump. van Esch B.P.M. et al, [2] investigated the performance and radial roading of a mixed-flow pump under non-uniform suction flow with experimental method, they find that the global performance of a pump is influenced by the type of inflow velocity profile and a considerable steady radial force appeared when the suction flow is non-uniform. Zhaofeng Xu et al, [3] studied the flow patterns of non-uniform flows in a rectangular open suction passage, such as distribution of velocity, streamline, turbulence kinetic energy and vortex, by a 2-D PIV method. Weidong Shi et al, [4] studied the effect of non-uniform sunction flow on performance and pressure fluctuation in axial flow pump. But all these non-uniform inflow conditions are quite different from that studied here. Wei Huang et al, [5] studied the flow characteristics within connection between steam generator channel head and pump suction by experimental method and found that the axial vortex is eliminated and axial velocity is uniform in the discharge pipe of the channel head with the nozzle dam brackets. It is a great discovery but the interaction between the channel head and the pump has not been investigated since no pumps are connected to the channel head in the test loop.
In this paper, CFD method is used to simulate a scaled model pump with three different suction flows, one comes from the straight pipe and the other two come from the cold side of the steam generator. 3-D flow field at the inlet of the impeller and in the channel head is studied in details. The circumferential velocity induced by the complex geometry upstream in the discharge pipe of the steam generator is analysed.

1 Numerical simulation

1.1 Geometry and mesh

Figure 2 shows the cold side of the steam generator, it's a fourth spherical cavity with two discharge pipes. Since the cold side of the steam generator is symmetrical, it is assumed that the flow field inside is also symmetrical. So the channel head could be divided into two mirror parts and the effect of each part on the performance of the pump can be investigated individually (Figure 3). The third suction flow comes from the straight pipe as shown in Figure 4.

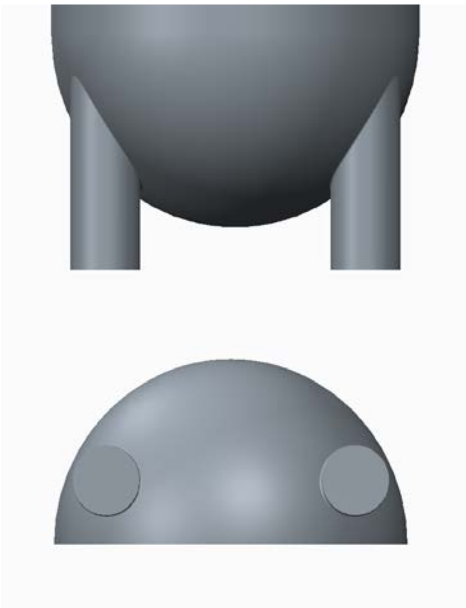

Figure 2. Front and upward view of the channel head

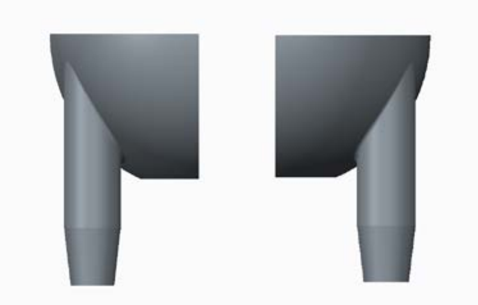

Figure 3. Left and right part of the channel head

Figure 4. Straight inlet pipe

The two model pumps are identical designs and both rotate clockwise. Each of them has 5 impeller blades and 11 guide vanes. The diameter of the 5-bladed impeller is 308mm. The computational domain includes three different inlet fluid zones, impeller, diffuser, spherical casing with straight outlet pipe as shown in Figure 5-7.

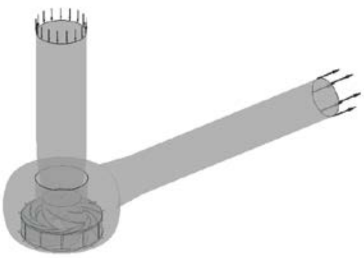

Figure 5. Computational domain with straight inlet pipe

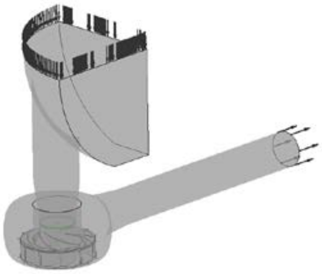

Figure 6. Computational domain with left channel head

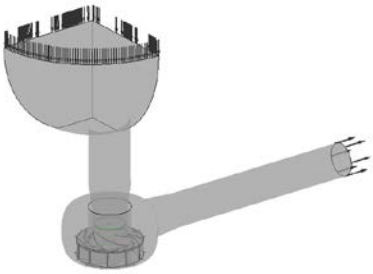

Figure 7. Computational domain with right channel head

In order to improve the grids quality, nearly all the computational domains are meshed by the hexahedral elements, except for the left and right part of the channel head with complex geometry. The grids information is show in Table 1. The calculation grids of the impeller, diffuser and casing are shown in Figure 8. The magnitude of y^+ around the blades is lower than 200.

Table 1. Mesh information of the computation domain

Domain	Element type	Number
Straight inlet pipe	hexahedral	1,214,400
Left channel head	tetrahedral	1,967,476
Right channel head	tetrahedral	1,967,476
Impeller	hexahedral	1,132,405
Diffuser	hexahedral	1,919,775
Casing	hexahedral	1,276,405

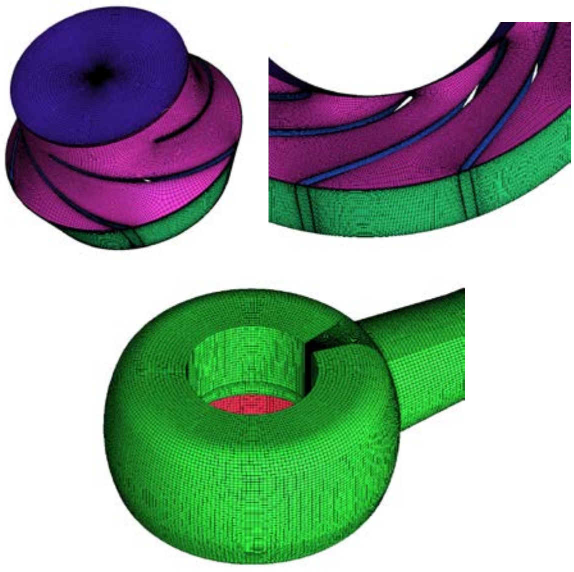

Figure 8. Calculation grids

All these grids are assembled in the commercial code CFX. The type of the interface between each part is set as frozen rotor.

1.2 Boundary Conditions

According to the practical operation condition and the similarity criterion, the design point is chosen to calculate. The working medium is water at 25℃ whose density is 997kg/m^3 and dynamic viscosity is 0.0008899kg/m/s. The angular velocity of the impeller is set to be n=1480r/min. The mass flow rate at inlet is 273.03kg/s and the outlet is set as a pressure boundary condition. Standard k-ε turbulent model [6] is chosen to solve RANS equation. This turbulent model has been used for more than 20 years and has been widely demonstrated to provide good predictions at design point for pumps. The adiabatic and no-slip boundary condition is applied to the solid walls.

2 Result and discussion

After running 2000 iteraions of each calculation, the RMS residuals have been reduced to a magnitude below 10^{-4}, which means that the calculations have converged.

2.1 Hydraulic performance

Table 2 shows the hydraulic performance of the pumps in different suction flows. It can be seen that the head of the pump under the left channel head increases by 1.1% while the head under the right channel head decreases by 2.2% when compared to the straight pipe case. Both the hydraulic efficiency of the pumps under the channel head is lower than that under the straight pipe. It can be concluded that the suction flow coming from the channel head of the steam generator has a considerable effect on the performance of the canned motor pumps.

Table 2. Hydraulic performance

Suction type	Head (m)	Head change	Efficiency
Straight pipe	15.147	-	86.71%
Left suction	15.311	1.1%	85.85%
Right suction	14.806	-2.2%	85.26%

2.2 Flow field analysis

In order to investigate the influence mechanism, 3-D flow field is analyzed. Figure 9 shows the position of the inlet of the impeller. The axial velocity contour and tangential velocity vector at the inlet are shown Figure 10-13. It indicates that the velocity distribution at the inlet is non-uniform, similar to the flow in an elbow pipe where secondly flow is induced by the centrifugal force. The axial velocity is small in the region near the center of the channel head while the secondly flow is strong there. Figure 11 and Figure 13 represent that the main swirl under left suction is counterclockwise while under right suction is clockwise.

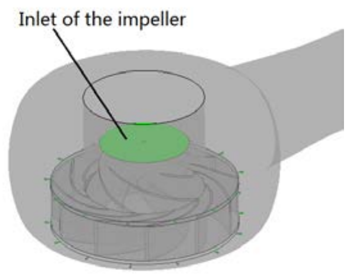

Figure 9. Position of the inlet of the impeller

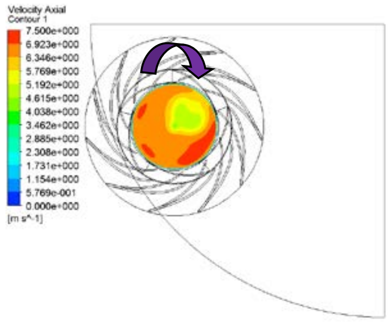

Figure 10. Axial velocity contour with left suction

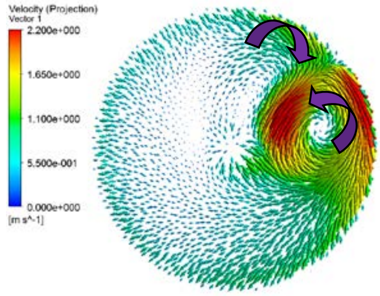

Figure 11. Tangential velocity vector with left suction

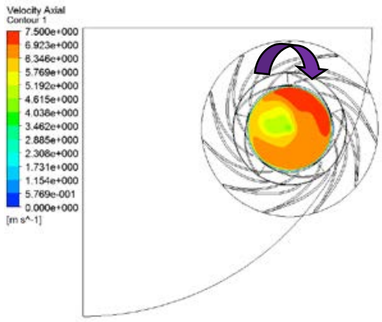

Figure 12. Axial velocity contour with right suction

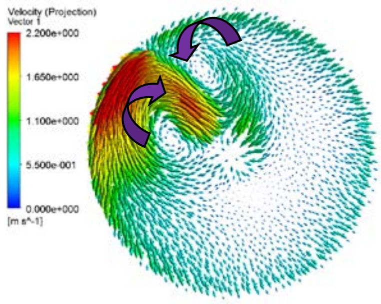

Figure 13. Tangential velocity vector with right suction

2.3 Analysis of the efficiency decrease

Figure 14-16 shows the turbulence eddy dissipation between blades at midspan in three different suction flows. It can been seen that the turbulence eddy dissipation is much larger in the non-uniform suction condition than that in the uniform suction flow. This is because the incidence angle is small when the suction flow is uniform at design point. While the incidence angle will increase when axial velocity and tangential velocity are non-uniform, which will cause the increase of impact loss and mixing loss. This is the reason that leads to the decrease of efficiency.

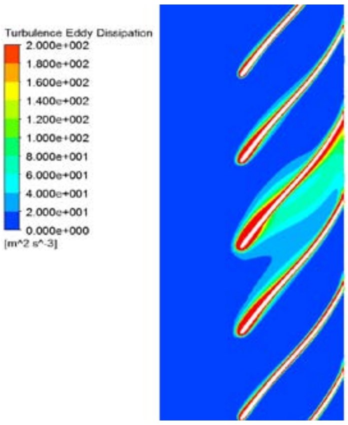

Figure 14. Turbulence eddy dissipation with left suction

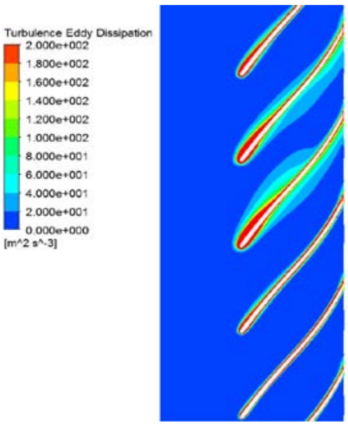

Figure 15. Turbulence eddy dissipation with right suction

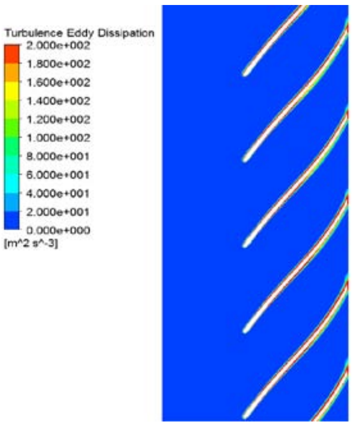

Figure 16. Turbulence eddy dissipation with straight pipe

2.4 Analysis of the head change

It is well-known that if the head of the pump decreases at a given flow rate, then a co-swirl condition should exist. If the head of the pump increases at a given flow rate, then a conterswirl condition should exist. Co-swirl is defined as rotation of the fluid at the inlet of the impeller in the same direction as the impeller rotates. Couterswirl is defined as rotation of the fluid in the opposite direction of impeller rotation.

As mentioned above, the head of the pump under the left channel head increases by 1.1% while the head under the right channel head decreases by 2.2% when compared to the straight pipe case. Since the impeller rotates clockwise while the main swirl in figure 11 is conterclockwise, which shows that a conterswirl exists and leads to the increase of the pump head. But the opposite phenomenon occurs in the right suction case in figure 13, where a co-swirl exists at the inlet of the impeller which leads to the decrease of the head.

But this is only a qualitative analysis, the net effect of the tangential velocity at the inlet of the impeller can be estimated by use of equation 1 which defines a term called average circumferential velocity.

$$V_{\mathrm{a}} = \frac{\oiint_{S} V_{\theta} \mathrm{d}S}{S} \tag{1}$$

Where:

V_{a} = average circumferential velocity [m/s]

V_{θ} = circumferential velocity at any point [m/s]

S = area of the inlet [m^2]

The calculation is done assuming that the rotating direction of the impeller is positive and the result is shown in Table 3. It clearly shows that there is no circumferential velocity component in the straight inlet case. While the value of average circumferential velocity is negative in left suction condition and leads to the increase of the head. The right suction condition is just the opposite.

Table 3. Average circumferential velocity

Suction type	V_{a} (m/s)
Straight pipe	0.00
Left suction	-0.202
Right suction	0.236

2.5 Generation of the circumferential velocity

The streamline in the left and right channel head are shown in Figure 17-18. It can be seen that the fluid firstly flows along the wall of the spherical channel head, when it reaches the bottom, the flow direction changes to the discharge pipe as shown with the arrow. It is the change of the flow direction at the bottom of the channel head that induces the circumferential velocity in the discharge pipe.

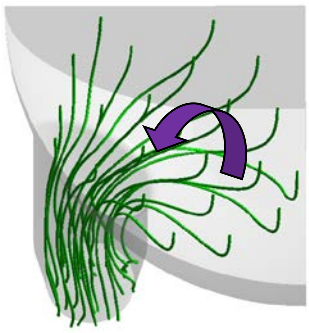

Figure 17. Streamline in the left part

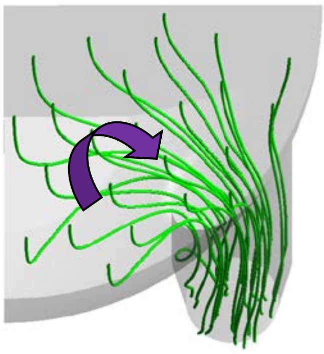

Figure 18. Streamline in the right part

3 Conclusion

In this paper, CFD method is used to simulate the model pump with three different suction flows, one comes from the straight pipe and the other two come from the cold side of the steam generator. An important observation in this study is that the head of the pump under the left channel head increases by 1.1% while the head under the right channel head decreases by 2.2% when compared to the straight pipe case. Both the hydraulic efficiency of the pumps under the channel head is lower than that under the straight pipe.

3-D flow field at the inlet of the impeller and in the channel head is studied in details. It is found that the main swirl at the inlet under left suction is counterclockwise while under right suction is clockwise. And the turbulence eddy dissipation is much larger in the non-uniform suction condition than that in the uniform suction flow which causes the decrease of the efficiency. The value of average circumferential velocity is negative in left suction condition assuming that the rotating direction of the impeller is positive and leads to the increase of the head. The right suction condition is just the opposite. The streamline in the left and right channel head explains the cause of the circumferential velocity.

The different head change of the two canned motor pumps found in this paper might induce the fluctuation of the flow rate in the primary coolant circuit and then lead to the safety problems in AP1000 reactor system. In order to mitigate the influence of the distortion suction flow on the head of the pumps, nozzle dam brackets should be installed in the discharge pipe of the steam generator to eliminate the circumferential velocity at the inlet of the impeller.

Acknowledgments

The author would like to thank Yun Long for providing the scaled model pump. The research work was supported by the National Natural Science Foundation of China (No. 51406114).

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