Numerical investigation of the effects of splitter blades on the cavitation performance of a centrifugal pump

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Abstract. For the centrifugal pump, additional splitter blades are sometimes necessary in order to improve the head and efficiency. On the other hand, the additional splitter blades will have effect on the cavitation performance due to the changes at the impeller inlet channel. In order to investigate this influence, three impeller schemes were proposed based on a model pump IS50-32-160, one without splitter blades and another two with splitter blades of different inlet diameters. Numerical simulations were carried out to investigate the characteristics of internal flow and the pump cavitation performances at different *NPSHA* with the CFD technique. The results show that the additional splitter blades will have some positive effect on the pump cavitation performance if the inlet diameter of the splitter blade is properly selected. The reason behind such improvement is that it helps to avoid the flow blocking at the impeller inlet and the vortex cavitation inside the blade passages effectively. For the pump model under our investigation, the cavitation performance reaches its best when the inlet diameter of the splitter blade is $0.725D_2$.

1. Introduction

Cavitation is one of the hazardous fluid dynamic phenomena which commonly occur in a wide range of fluid dynamic scenarios including pumps. Also, it can be used to remove burrs, rust, scratch from the cast, forged, machined parts in machinery field and broken gallstones in the medical field and so on. But so far, cavitation process is harmful in the field of hydraulic machinery. Usually when improving the efficiency of low specific speed centrifugal pumps, impeller diameter requires a significant reduction, so the larger blade outlet angle β_2 and more blades are required to achieve the required head. There will be crowding critically in the impeller inlet and cavitation is more severe by the increase of blades [1]. In this case, the splitter blades are often used to avoid the crowding at the impeller inlet and reducing the degree of diffusion inside the flow passage to stabilize the flow.

Since the 70s of the 20th century, the centrifugal pump with splitter blades has been studied by many scholars. The common conclusion is that the splitter blade can improve the head and efficiency of the pump and the cavitation performance. The former Soviet experts ХлопенковП.Р [2] and ВеселовВ.И. [3] studied the impeller with different splitter blade structures, and the experimental results showed that the pump head increased by 20% to 95% and the efficiency at high flow was increased by 12 % to 18% with the additional splitter blades. Yuan et al. [4] designed the orthogonal experiment L^9 (3^4) about the splitter offset blades and obtained an optimal design method for the shortly offset blades. Asuaje et al. [5] analysed the effects of the impeller outlet flowing by the additional splitter blades. Kergourlay et al. [6] analysed the simulation and pressure pulsation tests and

concluded that the pump head increased by 10 % to 15% with splitter blades and the pressure pulsation significantly reduced. Mustafa et al. [7, 8] showed that the splitter blades cause negative effects on pump performance in impellers with blade numbers of 6 and 7. When the splitter blade is added to the impeller with the blade number of 5, the efficiency increases with flow up to 10 L/s flow rate, after which it decreases as the splitter blade length increases. Zhang et al. [9] conducted numerical simulation and experimental studies on a centrifugal pump with additional splitter blades based on the multi-factor orthogonal method, and analysed the improvement mechanism of the "jet-wake" structure and the performance of centrifugal pumps by the splitter blade.

The current research of splitter blade mainly focused on numerical simulation of internal flow, PIV analysis and the pump performance analysis, and there is very few detailed study about the effect of the splitter blades on the pump cavitation performance. In this paper, a numerical analysis for low specific speed centrifugal pump was carried out by the commercial code ANSYS CFX 14.5. The result of the internal flow distribution and cavitation performance by the splitter blade was analysed mainly in the impeller for the design condition. In addition, some experimental data were also collected at the laboratory for the model pump to verify the accuracy of the simulation.

2. Model description and numerical methods

2.1. Physical model
A typical centrifugal pump IS50-32-160 with four blades is selected for study. The design parameters and rating hydraulic specification of the model centrifugal pump are shown in table 1.

Table 1. Parameters of the model centrifugal pump.

Suction diameter D_1	50mm	Flow rate Q_d	12.5m^3/h
Impeller out diameter D_2	160mm	Head H_d	32m
Blade out width b_2	6mm	Efficiency η	56%
Water temperature T	298.15K	Specific Speed n_s	47
Motor power P	3kW	Motor speed n	2900r/min

In order to evaluate the effect of the additional splitter blade on the internal flow distribution and cavitation performance of a centrifugal pump and its influence degree of the splitter blades with different inlet diameters. Three different impellers with main parameters as listed in table 2, are proposed and analysed in detail.

Table 2. Design schemes of the impeller.

Scheme abbr.	4	4-0.65D_2	4-0.725D_2
Number of long blades	4	4	4
Number of splitter blades	-	4	4
Inlet diameter of splitter blades D_i/mm	-	104	116
Offset angle of splitter blades θ/°	-	5	5

2.2. Computational model and boundary conditions

In this study, the three-dimensional model of the entire flow field is created by using Pro/E 5.0 software. The long straight pipes at the impeller inlet and volute outlet were properly extended to make the simulation more accurate. The entire models include the inlet and outlet extension part, the front and the back chamber, impeller and volute. The entire computational domain is shown in figure 1.
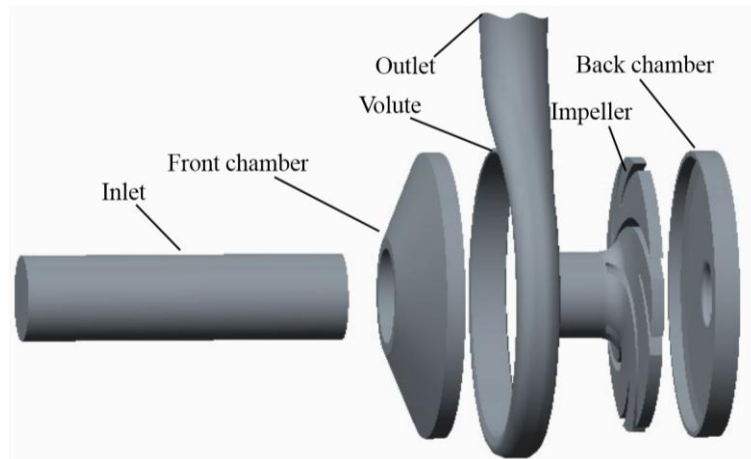

Figure 1. 3D model of the computational domain

CFD can accurately predict the head, power and efficiency of a centrifugal pump and greatly reduce the time of prototype test, development period and design cost. In this study, the mesh is generated by using ANSYS ICEM CFD 14.5. The unstructured tetrahedral mesh is chosen, this mesh has excellent adaptability and especially for the flow with complex boundary conditions. A mesh independence analysis has been carried out and the results are shown in table 3. When the mesh size is smaller than 1.65, which the mesh number is 1320620, the changes of the head and efficiency of the pump is very sight. Therefore, the mesh size 1.65 is selected to generating the computational domain, considering the configuration of the applied computer. The mesh distributions of the computational domains are partially shown in figure 2.

Table 3. Mesh independence analysis.

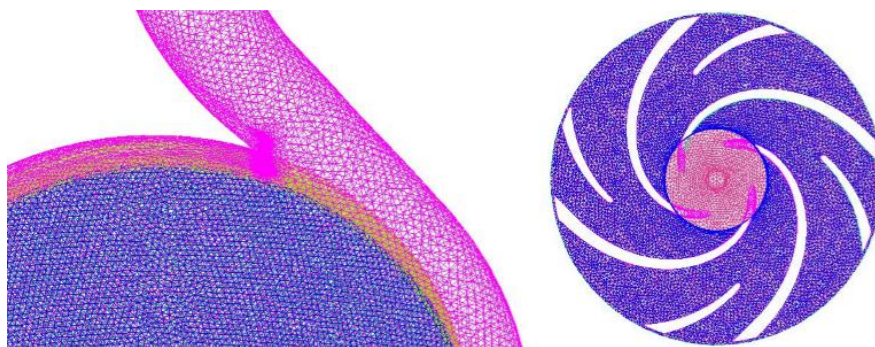| Parameters	Mesh size						
	2	1.9	1.8	1.7	1.65	1.6	1.55
Mesh numbers	815998	928835	1074409	1215410	1320620	1463019	1575341
Head H/m	37.557	37.987	38.337	37.838	37.878	37.876	37.881
Efficiency η/%	56.627	56.530	54.635	56.549	56.455	56.234	56.202

Figure 2. Mesh distributions of the computational domains.

The Reynolds-averaged SST k-ω turbulence model is used, which is a compounded model of the k-ω model and the k-ε model, so it is reliable in calculating the near-wall viscous flow as the k-ω model and accurate in calculating far-field free flow as the k-ε model. Boundary conditions are as follows: uniform velocity, isothermal temperature for the inflow plane, the averaged static pressure for the outflow plane, slip condition for the part between the inlet and the impeller and between the impeller and the volute, non-slip condition for the other walls, 12.5μm for the wall roughness (product is casting), standard wall functions for the near-wall region. The interfaces between the impeller and the connected parts (inlet and volute) are set to frozen rotor.

3. Results and discussions

3.1. Pump performance curves analysis

The design flow (Q_d=12.5m³/h) was selected to simulate the cavitation flow of the three models. The pump cavitation performance is drawn by calculating the *NPSHA* and the head at the different total pressure. The pump head is defined as the equation (1), the *NPSHA* is defined as the equation (2) and the hydraulic efficiency is described as the equation (3).

$$H = \frac{P_{in} - P_{out}}{\rho g} \tag{1}$$

$$NPSHA = \frac{P_{in} - P_v}{\rho g} + \frac{v_{in}^2}{2g} \tag{2}$$

$$\eta = \frac{\rho g Q H}{M \cdot n} \tag{3}$$

Figure 3 and 4 are the head and efficiency performance and the cavitation performance respectively of all 3 proposed schemes listed in table 2 at different flow rates. From figure 3, it can be seen that the scheme 4-0.65D_2 and 4-0.725D_2 show an obviously higher hydraulic head and relatively better efficiency at all the flow rate compared with the original scheme 4. So the additional splitter blade can improve the pump head and the pump efficiency greatly. The *NPSHR* of 3% head loss of the scheme 4-0.725D_2 is 0.4m lower than the original scheme 4 and 0.1m lower than the scheme 4-0.65D_2 as shown in figure 4, which indicates that additional splitter blade also improve the pump cavitation performance efficiently.
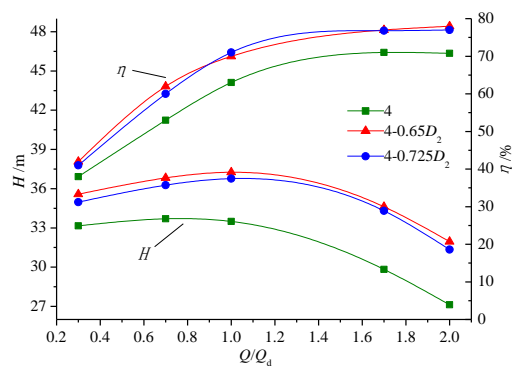

Figure 3. Hydraulic performance curves (head and efficiency) at different Q/Q_d of the three schemes.
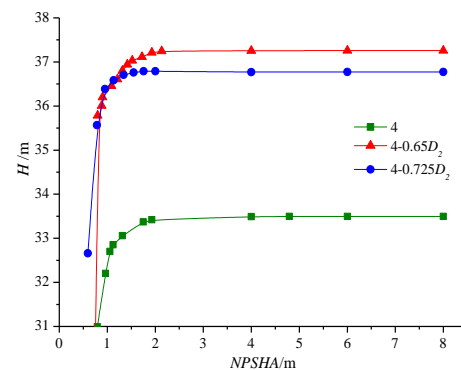
Figure 4. Cavitation performance curve (the head at different *NPSHA*) at the design flow of the three schemes.

3.2. Water vapour volume fraction analysis

In order to analyse the distribution and development of the pump cavitation bubble, figure 5 shows the water vapour volume fraction of the impeller middle section of three design schemes. Due to the impact of the blade inlet angle, the cavitation inception only occur in a small low pressure region on the blade suction side close to the impeller inlet, where the cavitation bubbles generate, shrinkage and collapse.

With the reduction of *NPSHA*, the bubbles will gradually expand from blade back low pressure area to the blade working face and the impeller outlet quickly as the impeller rotates rapidly. When *NPSHA* reached 0.6m, the cavitation of the scheme 4 is very serious since the bubbles occupy the entire impeller flow passage. While the cavitation of the scheme $4-0.65D_2$ and $4-0.725D_2$ is much better with the additional splitter blades were added, the bubbles occupy the entire impeller flow passage relatively only when *NPSHA* is down to 0.4m.
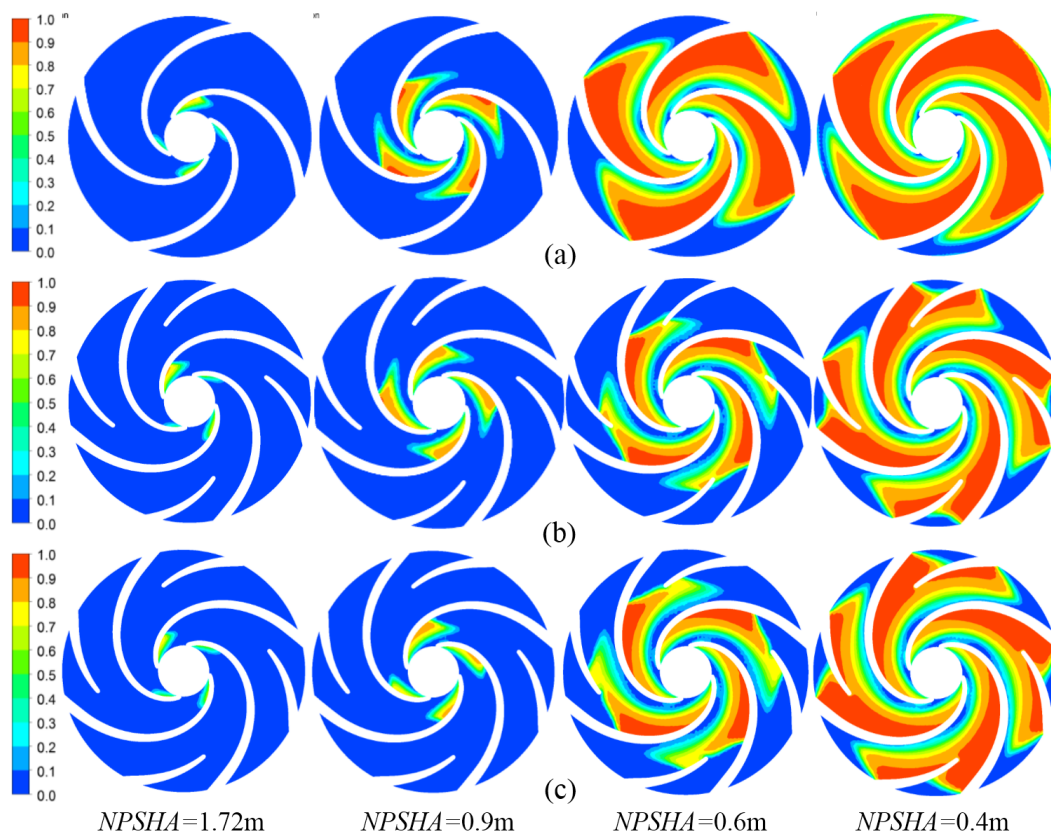

NPSHA=1.72m   *NPSHA=0.9m*   *NPSHA=0.6m*   *NPSHA=0.4m*

Figure 5. Water vapour volume fraction of the impeller middle section of the schemes 4 and $4-0.65D_2$ at different *NPSHA*. (a) numerical results for the scheme 4; (b) numerical results for the scheme $4-0.65D_2$; (c) numerical results for the scheme $4-0.725D_2$

3.3. Water velocity streamline analysis

Figure 6a shows the water velocity streamline of the impeller middle section at different *NPSHA* of the scheme 4. The water velocity streamline in the impeller passage are disorder, it has created significant cavitation vortex when the *NPSHA* is 2m, and with the reduction of *NPSHA*, the cavitation vortex area become larger and even fill the entire impeller passage, along with other small cavitation vortexes, which impede the normal flow seriously in the impeller passage, making the flow rate increased, the flow condition worsen and the flow loss increased.

From the figure 6a and figure 6b, it can be seen that the flow in the impeller with splitter blade is more stable than that of non-splitter blade impeller. The cavitation vortex begins to appear in blade back near the impeller inlet when *NPSHA* reduced to 1.5m, and when *NPSHA* reduced to 1.1m, there

are two connected cavitation vortexes in the passage between the back of the long blade and the working face of splitter blade, which make the loss of this passage increased.

As can be seen from the figure 6, the impeller flow of the scheme $4\text{-}0.725D_2$ with splitter blade is more stable than that of the scheme $4\text{-}0.65D_2$. It had created significant cavitation vortex in blade back near the impeller inlet only when the *NPSHA* reduced to 0.9m, which is decreased by 0.4m than that of the scheme $4\text{-}0.65D_2$. Similarly, the cavitation vortex move and become larger with the decrease of *NPSHA*, there are two connected cavitation vortexes in the back of the long blade only when the *NPSHA* reduced to 0.6m, which is decreased by 0.5m than that of the scheme $4\text{-}0.65D_2$.

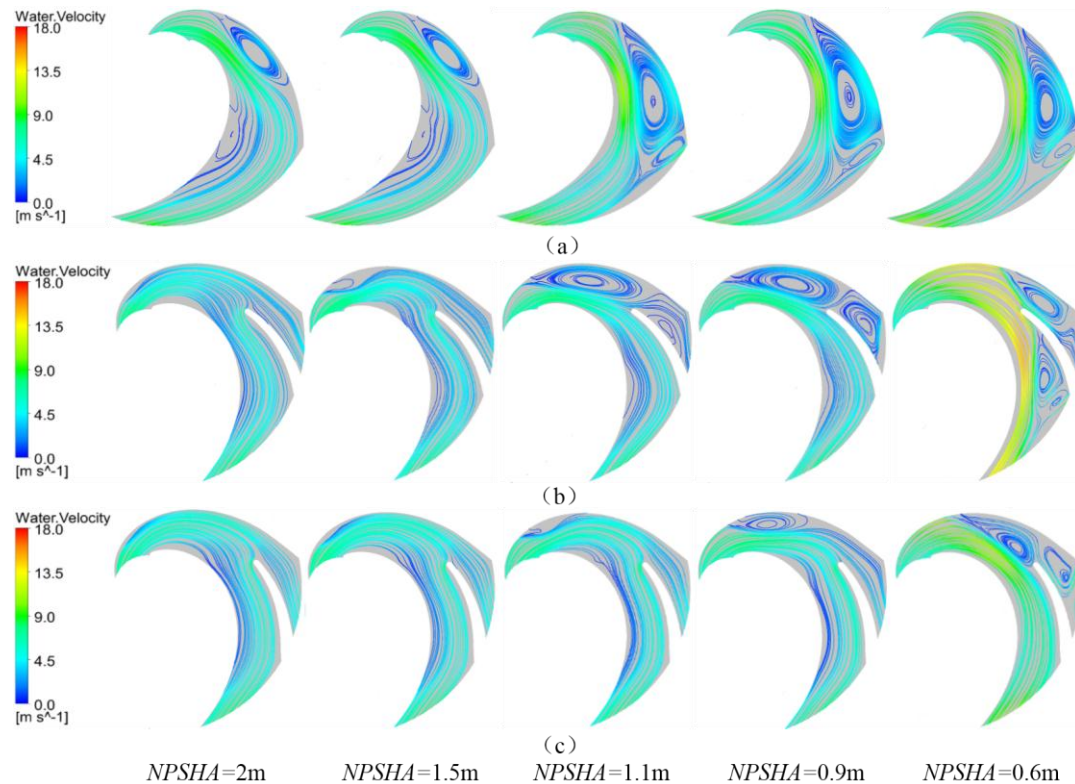|				
NPSHA=2m	*NPSHA*=1.5m	*NPSHA*=1.1m	*NPSHA*=0.9m	*NPSHA*=0.6m

Figure 6. Water velocity streamline of the impeller (One-quarter of the impeller middle section) at different *NPSHA* of the three different schemes. (a) numerical results for the scheme 4; (b) numerical results for the scheme $4\text{-}0.65D_2$; (c) numerical results for the scheme $4\text{-}0.725D_2$

4. Experimental verification

The experimental verification has been performed to certify the accuracy of the simulation. The tested impellers have been machined successfully with the rapid prototyping process and were shown in figure 7. The experiment was conducted at the test rig of open pump performance. The test rig is diagrammatically shown in figure 8.

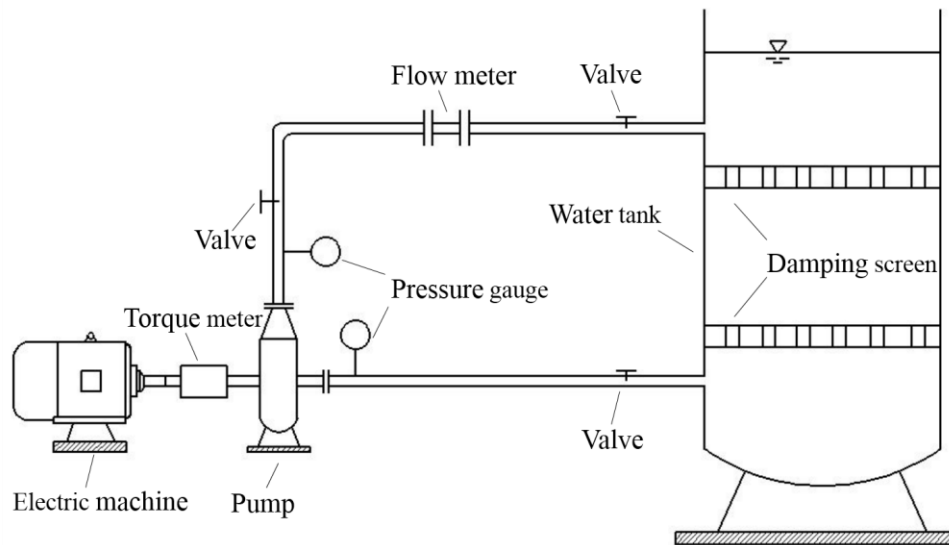|		
4	$4\text{-}0.65D_2$	$4\text{-}0.725D_2$

Figure 7. Test impellers of the three schemes.

Figure 8. Test rig of open pump performance.

Figure 9 and figure 10 show the pump head and efficiency at different flow rates (Q/Q_d) of the three models from the simulating and experimental results. The error between the simulating and experimental results may be caused by the casting of the model pump, machining errors, the test rig, etc... Moreover, the other principal part (volute, the long blades of the pump, etc...) of all three models are the same, the trends and errors of the head and efficiency performance curve are similar and consistent. Therefore, the numerical simulations are reasonable and comparable to analyse the effects of the pump cavitation performance by the splitter blades scientifically. In addition, it can be seen from the figures, the head of the additional splitter blade schemes 4-0.65D_2 and 4-0.725D_2 is increased by 2% to 12% than that of the model 4 without the splitter blade. Thus, the tested *H-Q* curve is flatter. Meanwhile, the head value of the two additional splitter blade scheme 4-0.65D_2 and 4-0.725D_2 are almost the same.
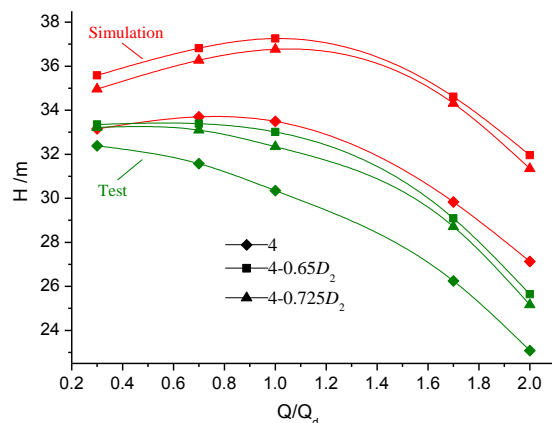

Figure 9. Comparisons of the head performance of the three schemes.
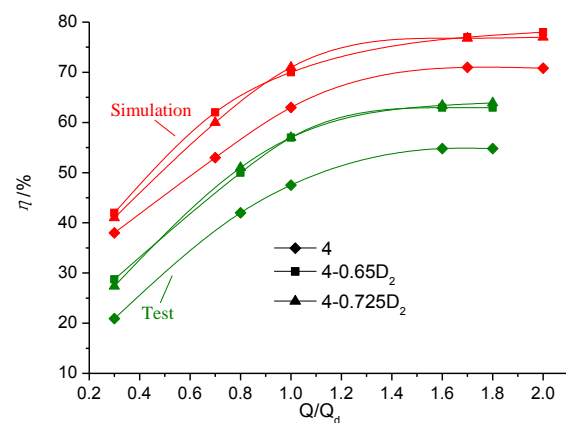

Figure 10. Comparisons of the efficiency performance of the three schemes.

5. Conclusions

The model pump IS50-32-160 is a typical centrifugal pump. In order to investigate the effect of the centrifugal pump cavitation performance with the additional splitter blades, three different impellers

schemes were proposed, numerically simulated and compared and analyzed focusing on the cavitation performance with following main conclusions:

(1) The head of the pump model is not only increased by 2% to 12% with the splitter blade, but also its *H-Q* curve is becoming flatter compared without the splitter blade. Meanwhile, the head of two models with different inlet diameters of the splitter blade $4-0.65D_2$ and $4-0.725D_2$ are almost the same.

(2) Together with the head and efficiency performance, the pump cavitation performances can be improved with the additional splitter blade. Also, the cavitation performance reaches its best when the inlet diameter of the splitter blade is $0.725D_2$ for the pump model under the investigation.

(3) The degree of improvements on the pump cavitation performance depends on the proper selection of the inlet diameter of the splitter blade, the reason behind is that it helps to avoid the blocking at the impeller inlet and cavitation vortex inside the water passage efficiently.

6. Acknowledgments

7. Nomenclature

D_1	Suction diameter[m]	Q_d	Design pump flow rate[m³/h]
D_2	Impeller out diameter[m]	Q	Pump flow rate[m³/h]
b_2	Blade out width[m]	H_d	Design pump head[m]
Z	Number of vanes	H	pump head[m]
P	Motor power[kW]	D_i	Inlet diameter of splitter blades [m]
P_{in}	Pressure of pump inlet [Pa]	v_{in}	Average speed of pump inlet[m/s]
P_v	Saturation pressure [Pa]	g	gravity acceleration [m/s²]
n	Motor speed[rpm]	θ	Skewing angel of splitter blades [°]
n_s	Specific Speed	ρ	fluid density [kg/m³]
η	Pump hydraulic efficiency[%]	NPSHA	net positive suction head available [m]
M	Impeller moment [Nm]	NPSHR	net positive suction head required [m]

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