Experimental studies on the optimization design of a low specific speed centrifugal pump

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Abstract. Start your abstract here...For a low specific speed centrifugal pump with the requirement of high efficiency of 68% and non-overload power characteristics, series experimental studies, by matching 9 volutes with 19 impellers were done. By combining the former research results about the splitters and the non-overload theory in centrifugal pump, the theoretical conditions to achieve the property of non-overload in a centrifugal pump with splitters was analyzed, and formulas to estimate the maximum shaft power and its position are derived. Based on the requirement of high efficiency and non-overload, blade outlet angle $\beta_2$, blade outlet width $b_2$, volute throat area $F_t$ and the inlet diameter of splitters $D_i$ were chosen with three levels to design a normal $L_9(3^4)$ orthogonal test scheme. Meanwhile, the optimized design scheme was determined, and corresponding test was done also, it demonstrates that the experiment purpose was reached, the design method to combine the splitters and non-overload theory is reasonable, which can get the property of high efficiency and non-overload.

1. Introduction
Yuan [1-3] revealed that the splitter blade technique is one of the techniques to solve three hydraulic problems of low specific speed centrifugal pumps (lower efficiency, drooping head-flow curve and easily overloaded brake horsepower characteristics).

Miyamoto et al.[4] examined the influence of splitter blades on the flow and performance by measuring the velocity and pressure in un-shrouded and shrouded impeller passages. M Asuaje et al.[5] studied the splitter blades effect on the performance of a centrifugal pump through both numerical simulation and experimental results. Experimental studies were made to investigate the effects of splitter blade length on deep well pump performance for different blade numbers[6, 7]. And the authors used an ANN(Artificial Neural Network) instead of expensive experiments to predict the performance of deep well pumps with and without splitter blades[8, 9]. These ANNs predicted results show good agreement between the predicted and experimental values.

To know more about the influence of splitter on the pump performance, and to obtain a useful design method to direct practical design. We also conducted an optimization and PIV test to study the influence of splitter blades on the flow of centrifugal pump impeller. And further PIV tests, a $L_9(3^4)$ orthogonal test, and many corresponding optimizations from CFD simulation were introduced in the paper[10].

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In this paper, for a low specific speed centrifugal pump with the requirement of high efficiency of 68% and non-overload power characteristics, series experimental studies were done. By combining the former research results about the splitters and the non-overload theory in centrifugal pump, the theoretical conditions to achieve the property of non-overload in a centrifugal pump with splitters was analyzed, and formulas to estimate the maximum shaft power and corresponding flow rate are derived, which also verified by a series tests.

2. Empirical exploration

The given operation parameters of a centrifugal pump are presented: \( Q = 50 \text{ m}^3/\text{h} \), \( H = 55.0 \text{ m} \), \( n = 2900 \text{ r/min} \), \( n_s=62 \), with the requirements \( \eta=68\% \), NPSHR =3m, and non-overload at the whole flow rates, and the maximum power must be less than the 1.1 times of the rated power.

For the design work, there are two difficulties: the first is the high efficiency, for the low specific speed pump with \( n_s=62 \), the rated value at the National stander GB13007 is 62.2\%, the requirement is higher than that by about 6\%, and the second difficulty is the non-overload requirement at the meantime. For conventional design methods, it is difficult to unite the requirements of high efficiency and non-overload at the meantime. The Greater flow rate design method can increase the efficiency, while the power characteristics may be worse, for the design method of Non-overload, which is presented by Yuan[2], can improve the power characteristics based on decreasing some efficiency. So no exiting method can be referred, we must do a lot of work to explore a new method to unite the two requirements. The design method with splitter blades can improve the pump performance, so splitters are used in our study to increase the efficiency and the throat area and the sections of volute are used to aim at the non-overload requirement.

2.1. Design schemes

Based on design experience, about twenty impellers and nine volutes are designed. The main design parameters of the impellers are list in Table 1. And the change laws of the nine volutes’ section area are shown in Fig.1, which is roughly increasing with the section number, with corresponding change for the \( D_3 \) and \( b_3 \). And the photos for the pump and several volutes are shown in Fig.2.

And many tests were done for different match of the impellers and volutes, the number of the matching schemes are list in Table 2.

| No. | Main parameters of impellers |
|-----|-----------------------------|
|     | \( D_2 \) | \( b_2 \) | \( D_1 \) | \( z \) | \( \beta_2 \) | \( \Phi \) | \( D_{si} \) |
| 1   | 215    | 8     | 92    | 5+5   | 32    | 105  | 108  |
| 3   | 215    | 12/9  | 92    | 5+5   | 15    | 110  | 108  |
| 5   | 224    | 7/6   | 92    | 5+5   | 13.5  | 100  | 158  |
| 6   | 224    | 9     | 92    | 6     | 20    | 110  | --   |
| 7   | 224    | 12    | 92    | 5+5   | 13.5  | 115  | 168  |
| 8   | 224    | 12    | 92    | 6     | 13    | 150  | --   |
| 9   | 240    | 5     | 80    | 4+4   | 13.5  | 180  | 168  |
| 11  | 238    | 6     | 80    | 4+4   | 15    | 180  | 151  |
| 12  | 224    | 9     | 90    | 5+5   | 14    | 160  | 166  |
| 13  | 238    | 7     | 80    | 6     | 11    | 200  | --   |
Table 2. Empirical design schemes

| Impellers | Volutes | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|-----------|---------|---|---|---|---|---|---|---|---|---|
| 1         | 1       |  |  |  | 23|  |  |  |  | 24|
| 2         | 3       |  |  |  |  |  |  |  |  |  |
| 3         | 2       |  |  |  |  |  |  |  |  |  |
| 4         | 1       |  |  |  |  |  |  |  |  | 7 |
| 5         | 2       |  |  |  |  |  |  |  |  | 8 |
| 6         | 3       |  |  |  |  | 11|  |  |  |  |
| 7         | 2       |  |  |  | 14|  |  |  |  | 26|
| 8         | 9       |  |  |  |  |  |  |  |  | 15|
| 9         | 10      |  |  |  |  |  |  |  |  | 16|
| 10        | 11      |  |  |  |  |  |  |  |  | 17|
| 11        | 12      |  |  |  |  |  |  |  |  | 18|
| 12        | 13      |  |  |  |  |  |  |  |  | 19|
| 13        | 14      |  |  |  |  |  |  |  |  | 20|
|           | 15      |  |  |  |  |  |  |  |  | 22|

Figure 1. Change law of the volute sections area
2.2. Experimental analysis

From large amount matching experiments, the No.4 volute is found to be the best design to match several impellers, which can meet the demand primarily. And the test results are shown in Fig.3. And the No.6 and No.8 volutes also have some good matching results.

From the present study, we can get the conclusion that, the parameters of impeller, especially impeller with splitters, have more influence in the high efficiency, and the design of volute throat and sections change have more influence in the power characteristics. While there are so many parameters are changed at the same time, it is very complicated to find the main effect factors. Based on these empirical explorations, further study for the non-overload characteristics with high efficiency should be done.

![Figure 2. The pump and several volutes](image)

![Figure 3. Results of impellers matching with the No.4 volute](image)

3. Study on the non-overload characteristics

3.1. Theoretical analyses

From the Euler equations for pump and the non-overload theory presented by Yuan[2], the theory to realize non-overload characteristics for the for the centrifugal pump with splitter blades are derived, Eq.(1) is the theoretical requirement to realize non-overload, and the Eq.(2) and Eq.(3) are the pre-estimating formula for the maximum power and its corresponding flow rate.

\[
\begin{align*}
\mu_a u_i^2 + \mu_b u_i^2 - \mu_a u_i^2 &= 2\mu_a u_i v_{\text{mix}} \cot \beta_i \\
+ 2\mu_b u_i v_{\text{mix}} \cot \beta_i - 2\mu_b u_i v_{\text{mix}} \cot \beta_i \\
\end{align*}
\]

\[
P_{\text{max}} = \frac{K_s \rho Q_{\text{max}}}{2\eta_u \eta_v} \left\{ \mu_u (u_i - v_{\text{mix}} \cot \beta_i) u_i \right. \\
+ \left. \mu_b [ (u_i - v_{\text{mix}} \cot \beta_i) u_i - (u_i - v_{\text{mix}} \cot \beta_i) u_i ] \right. \\
\]

\[
Q_{\text{Pmax}} = \frac{K_s [(1 - \mu_u) \mu_a u_i^2 + \mu_b u_i^2] \pi D_b h_b}{2[\mu_u u_i D_b h_b \cot \beta_i]}
\]
where $\mu_a=$ slip factor for the $H_{aw}$, $H_{aw}$ = theoretical head created by the part without splitters,

$$\mu_a = \left[ 1 + \frac{1.1(1 + \sin \beta_1)}{z_1} \times \frac{1}{1 - (D_1 / D_2)^2} \right]^{-1}$$  \hspace{1cm} (4)

$\mu_b=$ slip factor for the $H_{bw}$, $H_{bw}$ = theoretical head created by the part with splitters,

$$\mu_b = \left[ 1 + \frac{1.1(1 + \sin \beta_2)}{z_1} \times \frac{1}{1 - (D_1 / D_2)^2} \right]^{-1},$$

$K_1, K_2 =$ correction coefficient, recommend: $K_1 = 0.85 \sim 0.95$, $K_2 = 0.8 \sim 0.9$.

From the Eq.(2) and Eq.(3), we can see that, the design parameters, such as $\beta_2$, $b_2$, $D_{si}$, $F_t$ are also have great influence in the pump performance. An Orthogonal test is set to choose an optimization design parameters to explore the influence law of the efficiency, $P_{max}$ and $Q_{max}$, and the importance order.

3.2. Orthogonal test

3.2.1. The definition of test factors. As we know, $D_2$ is one of the most important factors to pump performance, but from exiting study, the present $D_2$ can meet the requirement? So the $\beta_2$, $b_2$, $D_{si}$, $F_t$, with three levels, are chosen as the test factors, a $L_9(3^4)$ Orthogonal test schemes are designed, as shown in Table 3 and Table 4.

| Level | factors | $A/\beta_2$ | $B/ b_2$ | $C/ D_1$ | $D/ F_t$ |
|-------|---------|-------------|-----------|-----------|---------|
| 1     | 13      | 6           | 151       | 996.9     |
| 2     | 15      | 9           | 158       | 1428.7    |
| 3     | 18      | 12          | 168       | 1716.0    |

| schemes | factors | $A$ | $B$ | $C$ | $D$ |
|---------|---------|-----|-----|-----|-----|
| 1       | $A_1$   | $B_1$ | $C_1$ | $D_1$ |
| 2       | $A_1$   | $B_2$ | $C_2$ | $D_2$ |
| 3       | $A_1$   | $B_3$ | $C_3$ | $D_3$ |
| 4       | $A_2$   | $B_1$ | $C_2$ | $D_3$ |
| 5       | $A_2$   | $B_2$ | $C_3$ | $D_1$ |
| 6       | $A_2$   | $B_3$ | $C_1$ | $D_2$ |
| 7       | $A_3$   | $B_1$ | $C_3$ | $D_2$ |
| 8       | $A_3$   | $B_2$ | $C_1$ | $D_3$ |
| 9       | $A_3$   | $B_3$ | $C_2$ | $D_1$ |

3.2.2. Test. The impellers were machined by the technology of Fused Deposition Modeling (FDM), whose material is plastic named ABS, and the volutes are machined by casting, as shown in the Fig.4. As shown in Fig.5, most of the tests were carried on the open test rig, which is belonging to the Machinery Industry Drain and Irrigation Production Quality Detecting Center, with accuracy of II class. The test data is fetched and processed by a computer automatically.
3.2.3. Test results analysis. Table 5 collects all the test results, which list the rated points, the point with the maximum power, and the corresponding flow rate. From the test data analysis, nine pumps all have the maximum power value, among them, the maximum power value of the No. 1, 2, 4, 5, 7 pumps are smaller than the rated power value, while the head of the No. 1, 4, 7 pumps are so lower than the requirement, and the power of the No. 3, 8 pumps are near the rated power, and the corresponding flow rate with the maximum power of the No. 3, 6, 8, 9 are much larger than the rated flow rate.

| No. | Rated point | the maximum point | the maximum P point |
|-----|-------------|-------------------|-------------------|
|     | Q          | H     | η      | Q      | H     | η      | Q<sub>max</sub> | P<sub>max</sub> |
| 1   | 45.01      | 45.82 | 58.18  | 45.23  | 45.39 | 58.53  | 49.59  | 9.60   |
| 2   | 51.50      | 57.17 | 64.17  | 62.01  | 50.84 | 65.30  | 74.82  | 13.48  |
| 3   | 55.42      | 61.19 | 65.12  | 66.27  | 56.26 | 66.31  | 85.08  | 16.30  |
| 4   | 49.31      | 50.41 | 60.67  | 52.65  | 46.16 | 61.12  | 58.77  | 10.95  |
| 5   | 52.21      | 60.81 | 65.33  | 63.13  | 54.80 | 66.32  | 82.48  | 15.15  |
| 6   | 56.18      | 66.67 | 67.51  | 71.91  | 60.62 | 70.05  | 90.41  | 17.98  |
| 7   | 49.55      | 54.25 | 64.02  | 60.60  | 47.34 | 64.62  | 67.83  | 12.20  |
| 8   | 53.21      | 61.58 | 65.04  | 70.51  | 54.38 | 66.54  | 91.39  | 16.67  |
| 9   | 56.25      | 68.02 | 66.01  | 70.03  | 63.52 | 68.66  | 98.69  | 19.87  |

3.2.4. Further analysis of the test data. \( K_i \) is the sum of the data in the row of \( i = 1, 2, 3 \), \( k_i \) is the arithmetical mean value, \( k_i = K_i / s \), where \( s \) is the number of the factor, at this study, \( s = 3 \), \( R \) is range, \( R = \max \{ k_1, k_2, k_3 \} - \min \{ k_1, k_2, k_3 \} \). Generally, the value of \( R \) reflects the influence of factors in the test results, a larger range means much more importance of the factor.
In Table 6, the range of all the factors are list, and the importance order for the performance of the four factors can be get as $b_2$, $\beta_2$, $F_t$, $D_i$.

In the same way, the histogram of the factors on the performance at the points with the maximum efficiency and power peak are obtained, as shown in Fig.5.

**Table 6. Analysis of test results of rated point**

| Performance factors | $b_2$ | $b_2$ | $D_i$ | $F_t$ |
|---------------------|-------|-------|-------|-------|
| $Q$                 | $k_1$ | 50.64 | 47.96 | 51.47 | 51.16 |
|                     | $k_2$ | 52.57 | 52.31 | 52.35 | 52.41 |
|                     | $k_3$ | 53.00 | 55.95 | 52.39 | 52.65 |
| $R$                 | $k_1$ | 54.73 | 50.16 | 58.02 | 58.22 |
| $H$                 | $k_2$ | 59.30 | 59.85 | 58.53 | 59.36 |
|                     | $k_3$ | 61.28 | 65.29 | 58.75 | 57.73 |
|                     | $R$   | 6.56  | 15.13 | 0.73  | 1.64  |
| $\eta$              | $k_1$ | 62.49 | 60.96 | 63.58 | 63.17 |
| $R$                 | $k_2$ | 64.50 | 64.85 | 63.62 | 65.23 |
|                     | $k_3$ | 65.02 | 66.21 | 64.82 | 63.61 |
|                     | $R$   | 2.53  | 5.26  | 1.25  | 2.06  |

**Figure 5.** Influence of different factors on the pump performance
3.3. Primary selection for optimization

From above analysis, a primary selection for three points, the rated point, the points with the maximum efficiency and power peak, is done, which considering the influence of factors on the $Q_{\text{max}}$, $P_{\text{max}}$, $e_{\text{max}}$, $H_{\text{et al.}}$ comprehensively. The selected combined design parameters are $A_2$, $B_2$, $C_3$, $D_2$, which is the No.5 impeller and the No.2 Volute at the Orthogonal test.

An experiment by matching the No.5 impeller and the No.2 Volute is done, its performance curve as shown in Fig.6. There is a power peak $P = 13.51\text{kW}$ at the point of $Q = 56.84\text{m}^3/\text{h}$, with $H = 58.54\text{m}$ and the maximum efficiency 67.08%, and we can see that the design has better performance at the larger flow rate, and with a flat efficiency curve.

![Figure 6. Performance curve of the selected scheme](image)

3.4. The verification for the non-overload theory

For the exiting centrifugal pumps with splitters, who have the non-overload characteristics, the maximum power value and its corresponding flow rate are calculated by the Eq.(2) and Eq.(3), and compared with those get from test, as shown in Table 7. By calculating its mean relative error, we get that, the error for the maximum power value is 6.45%, and the error for its corresponding flow rate is 4.20%, which can be used in engineering problems to estimate the maximum power value and its corresponding flow rate.

4. Conclusions

From above series research, we accumulate a lot of experience for designing centrifugal pumps with high efficiency and non-overload characteristics, and the amount of data will also give reference for the researchers.

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