Research of dependence centrifugal pump efficiency of impeller slit seal design parameters

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Abstract. Efficiency and resource are the two main characteristics of a centrifugal pump. In present study, the dependence of these characteristics from impeller sealing wear ring configuration was considered. Range of models with various configurations has been analyzed using numerical flow simulation. Compromise curves nave been obtained as a result. This information allows select an optimal ratio of dimensions of wear ring hydrodynamic slit seal to achieve a balance between efficiency and reliability of centrifugal pump.

Introduction

Double suction centrifugal pumps (type D pumps) are one of the most common types of centrifugal pumps. Studies of these pumps characteristics using CDF simulation methods are presented in [1]–[6]. Studies results of centrifugal pumps of similar subjects are described in [7]–[11].

The employ of the LP-tau search as a method for analyzing a number of models with various parameters and determining their optimal values is given in [12]–[15].

Sealing wear ring configuration of pump’s impeller has significant impact on general efficiency and reliability of the centrifugal pump. Studies of parameters of contained fluid in slit gaps using hydrodynamic modeling are described in [16]–[20].

In type D centrifugal pumps, one of the most actual issues is the selection of optimal design parameters of impeller gap seal from the point of view of volumetric efficiency and possible hydrodynamic reaction in it. Force of hydrodynamic reaction occurs due to drive shaft deflection and as a consequence an uneven distribution of pressure over the surface of the rotor with eccentricity. Volumetric efficiency is determined predominantly by the amount of leakage.

First of all, the geometrical parameters of the gap seal are a radial clearance \( \delta \) and length \( L \) which significantly influence the value of overflows and the lifting force. The criteria under study also depend on the total radial force that arises during pump operation and leads to shaft deflection.

Method

Two methods were used to calculate the leakage and lifting force in a gap seal:

1. CFD simulation employed the STAR-CCM+ software package to calculate values of leakage (volumetric efficiency) and lift force;

2. The LP-Tau search to calculate all values of varied parameters and determine the objective function.
The LP-Tau search generates points in a quasi-random manner in the interval for two parameters. For considering pump the radial clearance $\delta$ can vary from 0,3 to 0,6 mm and the length of the gap seal $L$ can vary from 10 to 20 mm.

It is not correct to vary shaft deflection value by the LP-Tau search method, since the pump does not always work in an optimal mode (especially when using the frequency regulation method). Accordingly, the radial force also changes during operation which leads to determine only the radial clearance and the length of the gap seal by the LP-Tau search method. In this case, pump shaft deflection varies for each calculated point obtained (for fixed values of $\delta$ and $L$) as a percentage of the gap: 0%, 30%, 60%. Thus, it becomes possible to analyze the dependence of the studied criteria on pump shaft’s deflection for each specific model.

In this case, generated operating points field can be visualized as a rectangle (Fig. 1).

![Points generation area](image)

**Fig. 1. Points generation area**

Obtained using the LP-Tau search control points are summarized in table 1.

**Table 1. Control points for the LP-Tau search**

| Point № | $\delta$, mm | $L$, mm |
|---------|---------------|---------|
| 1       | 0,3           | 12      |
| 2       | 0,45          | 15      |
| 3       | 0,525         | 12,5    |
| 4       | 0,375         | 17,5    |
| 5       | 0,4875        | 11,25   |
| 6       | 0,3375        | 16,25   |
| 7       | 0,4125        | 13,75   |
| 8       | 0,5625        | 18,75   |
| 9       | 0,58125       | 10,625  |
| 10      | 0,43125       | 15,625  |
| 11      | 0,35625       | 13,125  |
| 12      | 0,50625       | 18,125  |
| 13      | 0,39375       | 11,875  |
| 14      | 0,54375       | 16,875  |
| 15      | 0,46875       | 14,375  |
| 16      | 0,31875       | 19,375  |
| 17      | 0,459375      | 10,3125 |
| Point No. | $\delta$, mm  | $L$, mm  |
|----------|--------------|----------|
| 18       | 0.309375     | 15.3125  |
| 19       | 0.384375     | 12.8125  |
| 20       | 0.534375     | 17.8125  |
| 21       | 0.346875     | 11.5625  |
| 22       | 0.496875     | 16.5625  |
| 23       | 0.571875     | 14.0625  |
| 24       | 0.421875     | 19.0625  |
| 25       | 0.440625     | 10.9375  |
| 26       | 0.590625     | 15.9375  |
| 27       | 0.515625     | 13.4375  |
| 28       | 0.365625     | 18.4375  |
| 29       | 0.553125     | 12.1875  |
| 30       | 0.403125     | 17.1875  |
| 31       | 0.328125     | 14.6875  |
| 32       | 0.478125     | 19.6875  |
| 33       | 0.5390625    | 10.15625 |

In accordance with the method of considering varying pump drive shaft deflection for each control point, three models were computed (a total number of models is 99).

To obtain the results, the pressure difference at impeller gap inlet and impeller gap outlet is of primary importance. Pressure difference value can be computed from impeller gap flow part CFD simulation method employing STAR-CCM+. Also, this method allows evaluate the quality of design.

A three-dimensional model appearance of the computational domain is shown in Fig. 2. The computational domain is includes two elements: a gap between pump casing and the impeller and cylindrical slit seal.

![Fig. 2. 3-D model of computational domain](image-url)
Results
The following describes computation results of first model (this is control point №1, pump shaft deflection is 0%). A leakage graph through the slit seal is depicted in Fig. 3.

Fig. 3. Flow rate through slit seal

The radial force is 0,0107 N according to the calculation results.

The remaining models are calculated in a similar method in accordance with the data in Table 1. The calculation results are summarized in Table 2. It should be noted that the radial force in all models with pump shaft deflection absence assumes fairly small values of the order of 0,01 N. Therefore, it can be neglected.

| Pump shaft deflection, % | $Q_{yt} \cdot 10^{-3}$, m$^3$/s | $\eta_{ob}$, % | $F_R$, N |
|-------------------------|---------------------------------|----------------|-----------|
| Model № 1, 2, 3: \(\delta = 0,3 \text{ mm}, L = 12 \text{ mm}\) | 0 2,951 90,1 0,0 | 30 3,071 90,0 33,5 | 60 3,325 89,3 70,6 |
| Model № 4, 5, 6: \(\delta = 0,45 \text{ mm}, L = 15 \text{ mm}\) | 0 4,584 85,7 0,0 | 30 4,772 85,2 37,4 | 60 5,167 84,2 78,1 |
| Model № 7, 8, 9: \(\delta = 0,525 \text{ mm}, L = 12,5 \text{ mm}\) | 0 5,697 83,0 0,0 | 30 5,927 82,4 24,5 | 60 6,419 81,2 51,2 |
| Model № 10, 11, 12: \(\delta = 0,375 \text{ mm}, L = 17,5 \text{ mm}\) | 0 3,581 88,5 0,0 | 30 3,726 88,1 53,7 | 60 4,036 87,2 112,1 |
| Pump shaft deflection, % | $Q_{yr} \cdot 10^3$, m$^3$/s | $\eta_{ob}$, % | $F_{R}$, N |
|-------------------------|-------------------------------|----------------|-----------|
| Model № 13, 14, 15: $\delta = 0,4875$ mm, $L = 11,25$ mm |
| 0                      | 5,251                         | 84,0           | 0,0       |
| 30                     | 5,464                         | 83,4           | 21,7      |
| 60                     | 5,918                         | 82,3           | 45,3      |
| Model № 16, 17, 18: $\delta = 0,3375$ mm, $L = 16,25$ mm |
| 0                      | 3,213                         | 89,6           | 0,0       |
| 30                     | 3,334                         | 89,2           | 50,7      |
| 60                     | 3,606                         | 88,4           | 105,9     |
| Model № 19, 20, 21: $\delta = 0,4125$ mm, $L = 13,75$ mm |
| 0                      | 4,201                         | 86,7           | 0,0       |
| 30                     | 4,371                         | 86,3           | 34,2      |
| 60                     | 3,733                         | 85,3           | 71,5      |
| Model № 22, 23, 24: $\delta = 0,5625$ mm, $L = 18,75$ mm |
| 0                      | 5,741                         | 82,7           | 0,0       |
| 30                     | 5,973                         | 82,2           | 46,8      |
| 60                     | 6,469                         | 81,1           | 97,7      |
| Model № 25, 26, 27: $\delta = 0,58125$ mm, $L = 10,625$ mm |
| 0                      | 6,448                         | 81,0           | 0,0       |
| 30                     | 6,709                         | 80,4           | 17,2      |
| 60                     | 7,266                         | 79,1           | 35,9      |
| Model № 28, 29, 30: $\delta = 0,43125$ mm, $L = 15,625$ mm |
| 0                      | 4,329                         | 86,4           | 0,0       |
| 30                     | 4,504                         | 85,9           | 41,1      |
| 60                     | 4,879                         | 84,9           | 85,8      |
| Model № 31, 32, 33: $\delta = 0,35625$ mm, $L = 13,125$ mm |
| 0                      | 3,562                         | 88,5           | 0,0       |
| 30                     | 3,706                         | 88,1           | 34,8      |
| 60                     | 4,013                         | 87,3           | 72,8      |
| Model № 34, 35, 36: $\delta = 0,50625$ mm, $L = 18,125$ mm |
| 0                      | 5,099                         | 84,4           | 0,0       |
| 30                     | 5,305                         | 83,8           | 47,4      |
| 60                     | 5,746                         | 82,7           | 98,8      |
| Model № 37, 38, 39: $\delta = 0,39375$ mm, $L = 11,875$ mm |
| 0                      | 4,075                         | 87,1           | 0,0       |
| 30                     | 4,240                         | 86,6           | 27,6      |
| 60                     | 4,592                         | 85,7           | 57,7      |
| Model № 40, 41, 42: $\delta = 0,54375$ mm, $L = 16,875$ mm |
| 0                      | 5,614                         | 83,0           | 0,0       |
| 30                     | 5,841                         | 82,5           | 40,1      |
| 60                     | 6,326                         | 81,3           | 83,8      |
| Model № 43, 44, 45: δ = 0.46875 mm, L = 14.375 mm | Model № 46, 47, 48: δ = 0.37875 mm, L = 19.375 mm | Model № 49, 50, 51: δ = 0.459375 mm, L = 10.3125 mm | Model № 52, 53, 54: δ = 0.309375 mm, L = 15.3125 mm | Model № 55, 56, 57: δ = 0.384375 mm, L = 12.8125 mm | Model № 58, 59, 60: δ = 0.534375 mm, L = 17.8125 mm | Model № 61, 62, 63: δ = 0.346875 mm, L = 11.5625 mm | Model № 64, 65, 66: δ = 0.496875 mm, L = 16.5625 mm | Model № 67, 68, 69: δ = 0.571875 mm, L = 14.0625 mm | Model № 70, 71, 72: δ = 0.421875 mm, L = 19.0625 mm |
|-------------------------------------------------|-------------------------------------------------|-------------------------------------------------|-------------------------------------------------|-------------------------------------------------|-------------------------------------------------|-------------------------------------------------|-------------------------------------------------|-------------------------------------------------|-------------------------------------------------|
| δ, mm | Q<sub>y</sub>, 10<sup>-3</sup>, m<sup>3</sup>/s | η<sub>об</sub>, % | F<sub>R</sub>, N | δ, мм | Q<sub>y</sub>, 10<sup>-3</sup>, m<sup>3</sup>/s | η<sub>об</sub>, % | F<sub>R</sub>, N | δ, мм | Q<sub>y</sub>, 10<sup>-3</sup>, m<sup>3</sup>/s | η<sub>об</sub>, % | F<sub>R</sub>, N | δ, мм | Q<sub>y</sub>, 10<sup>-3</sup>, m<sup>3</sup>/s | η<sub>об</sub>, % | F<sub>R</sub>, N | δ, мм | Q<sub>y</sub>, 10<sup>-3</sup>, m<sup>3</sup>/s | η<sub>об</sub>, % | F<sub>R</sub>, N | δ, мм | Q<sub>y</sub>, 10<sup>-3</sup>, m<sup>3</sup>/s | η<sub>об</sub>, % | F<sub>R</sub>, N |
| 0 | 4.844 | 85.0 | 0.0 | 0 | 3.547 | 88.6 | 0.0 | 0 | 4.984 | 84.7 | 0.0 | 0 | 3.913 | 87.5 | 0.0 | 0 | 5.452 | 83.5 | 0.0 | 0 | 6.116 | 81.8 | 0.0 |
| 30 | 5.039 | 84.5 | 33.9 | 30 | 3.692 | 88.2 | 62.6 | 30 | 5.165 | 84.2 | 19.5 | 30 | 4.071 | 87.1 | 31.9 | 30 | 5.672 | 82.9 | 44.4 | 30 | 6.143 | 81.7 | 92.8 |
| 60 | 5.458 | 83.4 | 70.8 | 60 | 3.997 | 87.3 | 130.9 | 60 | 5.594 | 83.1 | 40.6 | 60 | 4.409 | 86.2 | 66.6 | 60 | 6.143 | 81.7 | 92.8 | 60 | 6.892 | 80.1 | 92.8 |
| 0 | 4,844 | 85.0 | 0.0 | 0 | 3.547 | 88.6 | 0.0 | 0 | 4.964 | 84.7 | 0.0 | 0 | 3.913 | 87.5 | 0.0 | 0 | 5.452 | 83.5 | 0.0 | 0 | 6.116 | 81.8 | 0.0 |
| 30 | 5.039 | 84.5 | 33.9 | 30 | 3.692 | 88.2 | 62.6 | 30 | 5.165 | 84.2 | 19.5 | 30 | 4.071 | 87.1 | 31.9 | 30 | 5.672 | 82.9 | 44.4 | 30 | 6.143 | 81.7 | 92.8 |
| 60 | 5.458 | 83.4 | 70.8 | 60 | 3.997 | 87.3 | 130.9 | 60 | 5.594 | 83.1 | 40.6 | 60 | 4.409 | 86.2 | 66.6 | 60 | 6.143 | 81.7 | 92.8 | 60 | 6.892 | 80.1 | 92.8 |
| 0 | 4,844 | 85.0 | 0.0 | 0 | 3.547 | 88.6 | 0.0 | 0 | 4.964 | 84.7 | 0.0 | 0 | 3.913 | 87.5 | 0.0 | 0 | 5.452 | 83.5 | 0.0 | 0 | 6.116 | 81.8 | 0.0 |
| 30 | 5.039 | 84.5 | 33.9 | 30 | 3.692 | 88.2 | 62.6 | 30 | 5.165 | 84.2 | 19.5 | 30 | 4.071 | 87.1 | 31.9 | 30 | 5.672 | 82.9 | 44.4 | 30 | 6.143 | 81.7 | 92.8 |
| 60 | 5.458 | 83.4 | 70.8 | 60 | 3.997 | 87.3 | 130.9 | 60 | 5.594 | 83.1 | 40.6 | 60 | 4.409 | 86.2 | 66.6 | 60 | 6.143 | 81.7 | 92.8 | 60 | 6.892 | 80.1 | 92.8 |
| Pump shaft deflection, % | $Q_{yr} \times 10^{-3}$, m$^3$/s | $\eta_{ob}$, % | $F_R$, N |
|--------------------------|---------------------------------|----------------|---------|
| Model № 73, 74, 75: $\delta = 0,440625$ mm, $L = 10,9375$ mm | 0 4,697 85,4 0,0 | 30 4,887 84,9 22,9 | 60 5,292 83,9 46,4 |
| Model № 76, 77, 78: $\delta = 0,590625$ mm, $L = 15,9375$ mm | 0 6,236 81,5 0,0 | 30 6,488 80,9 34,4 | 60 7,026 79,6 71,9 |
| Model № 79, 80, 81: $\delta = 0,515625$ mm, $L = 13,4375$ mm | 0 5,465 83,4 0,0 | 30 5,686 82,9 28,3 | 60 6,158 81,7 59,1 |
| Model № 82, 83, 84: $\delta = 0,365625$ mm, $L = 18,4375$ mm | 0 3,434 88,9 0,0 | 30 3,573 88,5 59,1 | 60 3,871 87,7 123,5 |
| Model № 85, 86, 87: $\delta = 0,553125$ mm, $L = 12,1875$ mm | 0 6,102 82,1 0,0 | 30 6,243 81,5 22,7 | 60 6,761 80,3 47,4 |
| Model № 88, 89, 90: $\delta = 0,403125$ mm, $L = 17,1875$ mm | 0 3,923 87,5 0,0 | 30 4,082 87,1 50,1 | 60 4,421 86,2 104,4 |
| Model № 91, 92, 93: $\delta = 0,328125$ mm, $L = 14,6875$ mm | 0 3,159 89,7 0,0 | 30 3,286 89,3 43,9 | 60 3,559 88,5 91,8 |
| Model № 94, 95, 96: $\delta = 0,478125$ mm, $L = 19,6875$ mm | 0 4,691 85,4 0,0 | 30 4,884 84,9 56,1 | 60 5,285 83,9 117,3 |
| Model № 97, 98, 99: $\delta = 0,5390625$ mm, $L = 10,15625$ mm | 0 5,956 82,2 0,0 | 30 6,617 81,6 16,8 | 60 6,712 80,4 35,1 |

For the possibility of analyzing obtained results, values of volumetric efficiency and hydrodynamic radial force are plotted on the $\eta$-$P$ coordinate plane and compromise curves are constructed (Fig. 4, Fig. 5).
Conclusion
If there is a significant load on pump shaft, efficiency of the pump is reduced, since if a shaft deflection occurs, leakage through the gap seal increases. The compromise curves (Fig. 4, Fig. 5) show that the greatest efficiency is achieved by increasing the length of the gap seal and reducing the radial clearance. From the point of view of reliability, the radial clearance value does not make sense to choose excessively small, since in this case wear of gap seal ring occurs more intensively over time. Thus, for considering pump parameters values of the gap seal are recommended $L = 14\ldots20$ mm, $\delta = 0,4\ldots0,5$ mm, provided that there is no pump shaft load that can establish a shaft deflection more than 60%. Also, the obtained data allow select optimal range of frequency regulation application.

Reference
[1] Y. Zhang, S. Hu, Y. Zhang, and L. Chen, “Optimization and analysis of centrifugal pump considering fluid-structure interaction,” Sci. World J., vol. 2014, Aug. 2014, Art. no. 131802. DOI: 10.1155/2014/131802
[2] S. Hatano, D. Kang, S. Kagawa, M. Nohmi and K. Yokota, Study of cavitation instabilities in double-suction centrifugal pump, International Journal of Fluid Machinery and Systems, 7 (2014) 94–100. DOI: 10.5293/IJFMS.2014.7.3.094
[3] Yao, Z.; Wang, F.; Zhang, Z.; Xiao, R.; He, C. Numerical and experimental investigation on the radial force characteristic of a large double suction centrifugal pump in a real pumping station. In Proceedings of the ASME-JSME-KSME 2015 Joint Fluids Engineering Conference, Korea, Seoul, 26–31 July 2015 DOI: 10.1115/AJKFluids2015-33468

[4] A Škerlavaj et al 2017 J. Phys.: Conf. Ser. 796 012007

[5] Gonzalez, J., Oro, J. M. F., Argles-Diaz, K. M., and Santolaria, C., 2009, “Flow Analysis for a Double Suction Centrifugal Machine in Pump and Turbine Operation Modes,” Int. J. Numer. Methods Fluids, 61, pp. 220–236. DOI: 10.1002/fld.1951

[6] Tao R, Xiao R, Zhu D Multi-objective optimization of double suction centrifugal pump. ProclMechE, Part C: J Mechanical Engineering Science 2018; 232: 1108–1117. DOI: 10.1177/0954406217699020

[7] Gülich J.F. Centrifugal Pumps. Springer-Verlag Berlin Heidelberg, 2010, 964 p.

[8] Bellarya S.A.I., Husain A., Samadec A., KanaiaR.A.. Performance Optimization of Centrifugal Pump for Crude Oil Delivery. The Journal of Engineering Research (TJER), Vol. 15, No. 1 (2018) 88–101 DOI: 10.1155/2018/3987594

[9] A Petrov et al 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012036

[10] V Cheremushkin and V Lomakin 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012039

[11] T Valiev and APetrov 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012038

[12] Lomakin V O, Chuburko P S and Kuleshova M S 2017 Multi-criteria Optimization of the Flow of a Centrifugal Pump on Energy and Vibroacoustic Characteristics Procedia Engineering 176 476–482

[13] Lomakin V.O., Kuleshova M.S., Chuburko P.S., Baulin M.N. Complex wet end part optimization of hermetic pump with LP-TAU method. Nasosy. Turbiny. Sistemy [Pumps. Turbines. Systems], 2016, no. 1, pp. 55–61.

[14] V Tkachyk et al 2019 IOP Conf. Ser.: Mater. Sci. Eng. 589 01201

[15] P Chuburko and Z Kossova 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012011

[16] Nakashima, C.Y. and Caetano, E.F. “Calculation Of Pressure Drop In Narrow Rotating Annular Clearances,” ”EngenhariaTérmica (Thermal Engineering)” 2008, pp. 27–34.

[17] Lomakin V O, Kuleshova M S and Kraeva E A 2015 Fluid Flow in the Throttle Channel in the Presence of Cavitation Procedia Engineering 106 27–35

[18] Kun Wang et al 2017 J. Phys.: Conf. Ser. 916 012025 DOI :10.1088/1742-6596/916/1/012025

[19] Babayigit, O.; Kocaaslan, O.; Aksoy, M.H.; Guleren, K.M.; Kocaaslan, O. Experimental and CFD investigation of a multistage centrifugal pump including leakages and balance holes. Desalin. Water Treat. 2017, 67, 28–40. DOI: 10.5004/dwt.2017.20153

[20] Prausova, H., Bublík, O., Vimmr, J., Luxa, M., Hála, J., Clearance gap flow: Simulations by discontinuous Galerkin method and experiments, EPJ Web of Conferences 92 (2015) 02073. DOI:0.1051/epjconf/20159202073.