The analysis of the performance of a sewage pump in terms of the wear of hydraulic components

O Moloshnyi1, P Szulc2, G Moliński3, S Sapozhnikov1 and S Antonenko1

1 Applied Hydro– and Aeromechanics department, Sumy State University, Sumy 40007, Ukraine
2 Department of Mechanics, Machines, Devices and Energy Processes, Wroclaw University of Science and Technology, Wroclaw, 50–370, Poland
3 Pompax Sp. z o.o., Rydzyna, 64–130, Poland

E-mail: o.moloshnij@pgm.sumdu.edu.ua

Abstract. The paper presents the comparison of the performance of the three-blade centrifugal pump operating in sewage water in the function of time. Two conditions of the analysed pump were investigated: a new one and one after a long period of utilization. The main objects of research were the impeller, due to its enormous wear, and the L–shape wear ring. Three dimensional optical scanning was applied to reflect the real geometrical parameters of the analysed pump. The investigations were conducted by means of numerical simulations and a real test. The achieved results indicate the impact of the wear on the performance curves of middle specific speed sewage pump.

1. Introduction

Sewage pumps pump fluid with a large amount of solid particles. It causes loss of material of wetted elements of the pump, namely hydro–abrasive wear. As a result, the wetted surfaces have a wavy wear pattern. Gulich [1] noted that the value of the wear rate corresponds to velocity to the power of three and additionally depends on the concentration of solids, flow patterns, vortices, turbulence, impingement angle, grain hardness, grain shape, corrosion, cavitation and material properties.

Hydro–abrasive wear changes the geometrical parameters of the hydraulic pump elements and, as a result, the pump performance. Leading and trailing edges as well as the wear ring are the most vulnerable elements of the pump [2]. In recent years, the analysis of the wearing processes, prediction erosion models and methods of its calculation have been discussed in many articles [3]. For example, Huang et. al. [4] detected the place in the analyzed pump hydraulic component where the wearing was the strongest by means of CFD method. Cai et al. [5] analyzed wearing processes of the back blades in impeller shrouds of the centrifugal slurry pump and they concluded that application of vorticity analysis is acceptable for determining the place of wear. The author identified that the increase of the flow rate causes the rise in the relative wear amount located in the impeller. Noon et al. [6] found that the tongue and belly portions of the volute casing of the lime slurry pumps are the most affected regions of erosion damage.

Another element of the centrifugal pump exposed to erosion is the wear ring. The influence of volumetric losses observed in the wear ring of the impeller on the performance of pump are well known. Stepanoff [7] presented the diagram of dependence of the leakage losses related to the specific
speed of the pumps. Aronen [8] investigated that the reduction of the clearance in the wear ring to 50\% causes increase of the efficiency at 2–3\% and the authors noted that the rise of a rotor speed revolution causes the increase of efficiency. Nelik [9] noted the decrease of the efficiency from 62.5\% to 53.6\% when the pump was equipped in a wear ring and without it. Zhu [10] presented the results of the comparison of the different clearances between the impeller and the housing of the pump and found out that the head and efficiency were decreased while the increase of the clearance height, and it has more influence on the value of efficiency and less on the head. Ha and Choe [11] analyzed annular seals and noted that the leakage prediction obtained from the 3D CFD is a good alternative for the experiment, moreover, the increase of the eccentricity from 0 to 0.5 causes the rise of the leakage to 4.5\%. Li et al. [12] compared the results of transient CFD analysis of the annular seals and the outcomes of experimental test and received 14\% error.

The above-mentioned and many other researches [13] considered the influence of reduction of the clearance on the working process (efficiency) of the pump. Another interesting aspect, which is not investigated enough, could be defined as the influence of clearance increase caused by the wear. It is very difficult to calculate leakage from the wear ring after deterioration of its structure by using a theoretical formula due to irregular form of its walls. The main idea of the article is the determination of the wear influence affected on the pump wetted elements on the working processes of the unit. Especially the geometrical changes of the impeller, increase of the volumetric losses and its impact on the head and efficiency performance were analyzed. The structure of flow and the distribution of velocity at the impeller and seal were also taken into consideration.

2. Object of research
The object of the current research is KSB KRP K 500–540/2 pump. Two conditions of the analyzed pump were investigated: the new one and after a long period of utilization. The main object of research was the impeller due to the enormous wear (figure 1). Three-dimensional optical scanning was applied to reflect the real geometrical parameters of the analyzed elements. After a long period of operation many places of hydro–abrasive wear of wetted parts of the pump were observed. Most of them were on the impeller (figure 1). The damage of the leading edges near the shroud was identified. The main reason of its occurrence could be defined as the highest linear velocity of this element and the influence of centrifugal forces acting on the transported solid particles. The generated wear was the cause of the local vortexes were created. Also, the L–shape wear ring was the object of research (figure 1 (f)), where significant and uneven wear of the sealing surface was observed. The materials of the impeller and wear ring are made of GJL250.

The nominal working pump parameters are as follows: head \( H = 8,9 \, m \), flow rate \( Q_{\text{nom}} = 2772 \, m^3/\text{h}(770 \, l/s) \), rotational speed \( n = 725 \, \text{rpm} \), specific speed \( n_q = 125 \, \text{rpm} \). Impeller outer diameter was defined as \( D_z = 550 \, (500) \, mm \) and the number of impeller blades was \( z = 3 \).

The processes of wear of the seal (wall of the wear ring and the impeller) were divided into 6 states: 0\% it is a new seal; 12.5\%, 25\%, 50\%, 75\% are intermediate percentage of wear; 100\% it is the state at the end of the long period of utilization (figure 2).

3. Methods of research
As the main method of research numerical simulations were assumed. A solid model of a fluid computational domain of the sewage centrifugal pump includes: the inlet pipe, the outlet pipe, the impeller, the volute, rear side chamber (figure 3) and front side chamber with a suction wear ring (figure 4). The first step of the investigations was focused on the pump performance identification thus the computational model did not include the front side chamber with the suction seal of the pump. This reduces complexity of the analyzed structure and increases the speed of the calculation. To understand the flow character in the suction wear ring, which was the second step of the simulation, the front side chamber with the seal was simulated separately from the other hydraulic parts and the numerical model was constrained by the angle range of 6 degrees.
Figure 1. Elements of the pump after the enormous wear: (a) impeller, (b) scanned 3D model of impeller, (c) trailing edge, (d) scanned 3D model of trailing edge, (e) leading edge, (f) suction wear ring.

Prepared 3D models were discretised by means of tetra mesh for: the impeller, volute, rear side chamber and front side chamber with seal (for the new pump) and hexa mesh was used for: inlet, outlet pipe and front side chamber with suction wear ring (for the worn pump). The meshes were generated by means of software ICEM–CFD. The grids were presented in figure 3 and figure 4. The size of the elements was selected due to mesh independence test. Layers of prismatic elements were created near solid walls in all boundary layers (figure 4). The total number of elements of the pump flowing part without the front side chamber was equalled 2 million nodes. The front side chamber with seal contains 1.18 million nodes for hexa mesh, and from 0.41 million to 0.45 million of nodes for tetra mesh.
Figure 2. Schema of the seal with different value of wall wear.

The numerical simulation of the operational process of the sewage pump was conducted by means of ANSYS CFX software. Reynolds–Averaged Navier–Stokes method was used, with the standard $k$–$\varepsilon$ turbulence model [17]. Boundary conditions were set as the total pressure at the pump inlet, the mass flow rate at the pump outlet, the total pressure at the inlet to the front side chamber with suction wear ring, opening pressure and direction at its outlet. The numerical calculation was carried out at the flow rate $(0.3–1.2) \: Q_{\text{nom}}$ for the pump and $(0.8; 1.0; 1.2) \: Q_{\text{nom}}$ for front side chamber with wear ring. Working fluid was clear water at a temperature of $25^\circ C$.

Figure 3. The object of analysis: (a) computational domains: 1 – inlet pipe, 2 – impeller, 3 – volute, 4 – outlet pipe; (b) calculation mesh of the new impeller.

Figure 4. Calculation mesh of the suction wear ring: (a) 0% of wear; (b) 100% of wear.
4. Results and discussion
The working parameters of the pump were determined as the imputed date for the validation of the results of the numerical simulation realized for new and worn wetted elements. The outcomes of this process were presented in figure 5. The difference between the results of the simulation and experiment is less than 4%. The discrepancies between the performance of the new and worn impeller, volute and wear ring can be clearly visible. The pump unit after a long period of utilization has the lower head, power and efficiency. The value of the difference is: 18% for the head, 5% for power and 14% for efficiency at $Q_{\text{nom}}$. The main reason of this deterioration is the damage of trailing edges, which causes the reduction of the outlet diameter (figure (1)). The other reason is not uniform wear of the wetted surfaces, which cause an additional vortex and hydraulic losses, especially the losses of disc friction. Also the value of volumetric losses will affect the performance achieved from a new and worn hydraulic component of the pump.

Figure 5. Pump performance curves.

In the next step the quality analysis of the flow structure was made. The absolute pressure distributions in the $Q_{\text{nom}}$ in the analysed objects were presented in figure 6. The comparison of the absolute pressure distribution in the impellers and volutes cross sections indicates that inner pressure distributions are similar for both cases. There are no sudden changes in the pressure gradient. Instead of the small number of blades the flow is quite uniform and the hydraulic energy increases along the main flow line. Same differences could be observed on the pressure side of the blades. For the worn impeller, the pressure is a little bit bigger, but it is a result of assumed position of cross-section. In the inlet surface of the spiral casing the pressure is globally smaller compared to the new unit, which causes the drop of the head. It means that reduction of the outer diameter has significant impact on the performance during the process of wear.

The flow in the spiral casing could also be called uniform. The absolute pressure is equalled almost in all cross–sections. Small deteriorations could be observed in the elbow part of outlet diffuser and on the interface between impeller and spiral.

The comparison of the velocity distribution in the impellers and volutes cross–sections was presented in figure 7. The conducted analysis indicates that there is not a great difference in achieved outcomes. Similar to the pressure distribution, some, small differences on the pressure side of the blades could be observed. The reasons of its occurrence are the wearied trailing edges of the blades.
Figure 6. Absolute pressure distribution at the $Q_{nom}$: (a) impeller, new pump; (b) impeller, worn pump; (c) impeller and volute, new pump; (d) impeller and volute, worn pump.

The analyses of the value of the volumetric losses in the suction wear ring indicate that the rise of the losses is not directly proportional to the wear. Base geometry (new rings), which was named 0%, has a leakage losses equal 1% of the flow rate. The main increase of the losses is at first 25% of the wear of the ring structure. For this range of seal geometry deterioration (0%–25%) observed losses are half, noted for fully wearied (named 100%). After 25% of wear the increase of losses is slower. It can be caused by the not great change in resistance of the seal. After 25% of wear the suction ring does not meet the assumed requirements. The maximum volumetric losses at the level 8% according to Stepanoff [7] should be assumed, and in analysed situation they are higher. Additional volumetric losses cause the rise of the Life Cycle Cost (LCC). According to the calculation, the pump with a new seal (0% of wear) lost 0.05 Euro per hour for the leakage losses, and it is not a significant at value. At the end of the period of utilization this value is 1.5Euro – the rise more than 30 times. The changing of the wear ring at the moment of the 8% losses gives an opportunity to save approximately €3,800 /a.
Figure 7. Velocity distribution at $Q_{nom}$: (a) impeller, new pump; (b) impeller, worn pump; (c) impeller and volute, new pump; (d) impeller and volute, worn pump.

Figure 8. Comparison of suction seal leakage for different value of the wear: (a) absolute values; (b) rise of the values.
Figure 9. Absolute pressure distribution in the seal at $Q_{nom}$: (a) new pump (0%); (b) 12.5%; (c) 25%; (d) 50%; (e) 75%; (f) 100%.

Figure 10. Velocity distribution in the seal at $Q_{nom}$: (a) new pump (0%); (b) 12.5%; (c) 25%; (d) 50%; (e) 75%; (f) 100%.
For better understanding of the flow behaviour the analyses of pressure distribution and velocity in the axial cross section of the front side chamber with the suction wear ring were presented in figure 9 and 10. The views of the absolute pressure distribution in the front side chamber show the zones of great reduction of the pressure in the base geometry (figure 9). The big difference in the pressure distribution along the clearance was observed. For 50%, 75% and 100% of wear very small pressure drop was noticed. The main reason is the height of the gap between two rings. The lowest pressure was recorded in the outlet of the wear rings, when the direction of the flow is changing rapidly (for example figure 9 (c)). This place corresponds to the highest velocity (figure 10). The low value of the velocity indicates the zones of the low wearing of the analysed object. The significant difference in the velocity distribution in achieved outcomes could be observed. Mainly the flow direction in the throttling clearance is parallel to the gap and the abrasion of the material could be observed. After 50% of wear of the rings walls, the value of the velocity is not so great, which causes fewer losses and also wear.

Additional analysis of the vortex creation was conducted. Gulich [1] claims that vortices are very abrasive. Place and value of vorticity indicates the place of the wear intensification of the material. The archived outcomes were presented in figure 11, and observed places correspond to the real locations.

![Vortices distribution in the seal at Qnom.](image)

**Figure 11.** Vortices distribution in the seal at $Q_{\text{nom}}$: (a) new pump (0%); (b) 12.5%; (c) 25%; (d) 50%; (e) 75%; (f) 100%.

5. Conclusions
The paper presents the results of the comparison of the performance, losses and flow structure in the three bladed centrifugal sewage pump in the function of time. In accordance with conducted analysis the following conclusions could be drawn:

1. Accurate numerical simulation results were obtained in comparison with real pump performance.
2. The long period of utilization causes erosion of the wetted elements of the pump and consequently the reduction of the working parameters was observed – the head 18%, efficiency 14% and power 5%.
3. Comparison of the distribution of the pressure and velocity in the pump indicates a small difference in the new and worn unit. The biggest difference is located at the zone near the outlet of the impeller and inlet to the volute.

4. According to the results first 25% of the seal wear cause increase of 50% of the volumetric losses.

5. Additional volumetric losses in the suction seal cause the rise of the LCC. The pump with worn seal lost 30 times more money than the new one.

6. The pressure, velocity and vorticity distribution are very different for different value of the suction wear ring damage. Vorticity distribution indicates the places of the significant erosion, which correspond to the real location.

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