Analysis of the reasons of double suction centrifugal pump's failure

Analiza przyczyn awarii dwustrumieniowej pompy odśrodkowej

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The issue of a pump operation beyond its recommended range **and required flow conditions to the suction flange and consequently failure of double suction axially split centrifugal pump was explained. The increased vibration level, improper configuration of the suction pipeline and used hydraulics were presented. It was also illustrated how to diagnose and solve a problem which after implementation at the target workplace has confirmed the correctness of implemented design solutions.**
KEYWORDS: centrifugal pump, double suction axially split centrifugal pump, pump's vibration, caviation, CFD simulations

Wyjaśniono problem pracy pompy poza zalecanym zakresem, wymaganymi warunkami napływu cieczy do króćca ssawnego i w konsekwencji ulegania awarii przez 2-strumieniową pompę wirową. Przedstawiono zagadnienie zwiększonego poziomu drgań, niewłaściwej konfiguracji zabudowy rurociągu ssawnego oraz zastosowanej armatury. Zilustrowano sposób diagnozowania i rozwiązywania problemu, który po wdrożeniu w docelowym miejscu pracy potwierdził słuszność wprowadzonych rozwiązań konstrukcyjnych.
SŁOWA KLUCZOWE: pompa odśrodkowa, pompa 2-strumieniowa, drgania pomp, kawitacja, obliczenia CFD

Proper operation of the pump installation depends on the correctness of the selection and operation of the individual elements in relation to arising needs. The key issue, apart from installing correct fittings and a pumping unit is according to recommendations designing the piping system, particularly on the suction side of the pump. Its geometry determines pump supply conditions with transported liquid and affects the lifetime of entire pumping aggregate.

Centrifugal pumps are selected for specific parameters, however they actually work in the scope of operation; the larger scope, the larger changing demand for discharged factor. Scope of operation, reliability and timeliness of correct exploitation of centrifugal as presented in fig. 1 [5, 6]. The best scope of pump operation in the system is range (0.9÷1.05) Q_{BEP}, less recommended (0.8÷0.1) Q_{BEP} and the least recommended but still good is a range (0.7÷1.15) Q_{BEP}.

The reduction of the pump lifetime, with its efficiency different from the optimum, is due to the presence of reverse currents and local separation of the stream resulting from the mismatch of speed and direction of flow to the geometry of the impeller blades, and stator if it exists. In addition, increasing efficiency over the recommended range or local acceleration of the liquid causes the pressure drop to evaporation pressure value and as a consequence, to precipitate vapour bubbles from the liquid called cavitation. Cavitation can also occur as a result of very strong

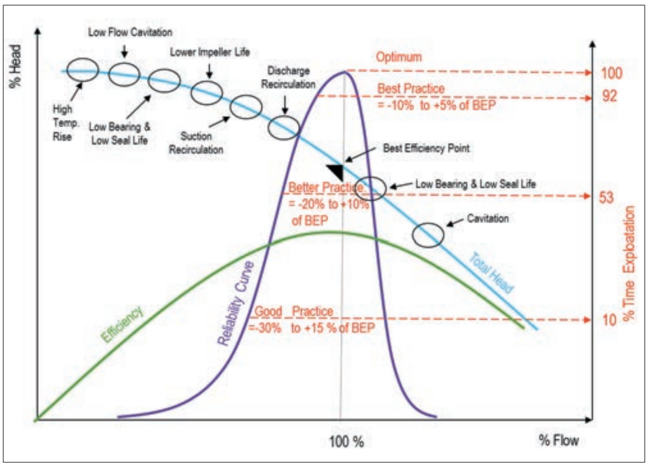

Fig. 1. Pump curve sensitivity for pump reliability [5, 6]

turbulence due to high velocity and pressure drop inside the whirl. The appearance of vapour bubbles does not jeopardize the pump operation, however it reduces delivery head. Pumping of liquid with precipitated vapour to the zone of higher pressure causes disappearance of separated gas. The bubble implosion process is very violent and occurs at a very high frequency (~25 kHz [12]) causing noise and vibrations increase. Additionally, a collapsing bubble produces a micro stream which can strike against the wall of a flow system causing its accelerated erosion. Cavitation can be divided into 3 phases. In the 1st stage, although a slight amount of bubbles is present in the flow, it does not disturb the pump parameters, but speeds up its wearing. In the 2nd, generated delivery head is reduced, noise and vibrations are increased which is associated with a considerable reduction of pump failure-free operation. In the 3rd phase, there is a strong decrease of power parameters which practically unable the normal pump operation.

The consequence of the above things is the occurrence of increased vibration of the pump mechanical system, which causes shortening of pump operating time. Depending on the local working conditions and the design pump resistance, accelerated wearing can concern sliding rings of mechanical seal and rolling bearings and can cause the erosion of the walls of the flow system elements. It is also possible to overlap some causes and results of vibration, which as a result greatly reduces the pump life time.

Description of the problem

The aim of the research was to clarify the problem of vibrations and accelerated wear of a double suction pump's bearings working in the city water supply system. The calculated bearings lifetime in this case was 20,000 hours, while the local working conditions caused failures following after ~1,000 hours, it means 20 times faster than expected.

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The subject of research was horizontal double suction pump which was selected for below duty point parameters: capacity Q_{pp} = 700 m^3/h, delivery head H_{pp} = 39 m, required anti cavitation surplus $NPSHR_{pp}$ = 5 m, power on the pump's shaft Pw_{pp} = 91 kW, rotation speed n_{pp} = 1,485 min^{-1}. A distinctive feature of double suction pumps is the impeller with 2 inlets which are located oppositely – mirror (fig. 6). This design solution eliminates the basic reason of axial forces and simplifies the way bearing of the rotating assembly. [19, 9]. The schematic diagram of discussed system is presented in fig. 2.

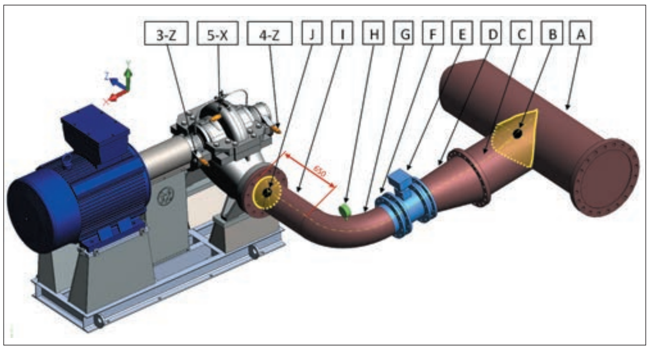


Fig. 2. Considered suction pipeline with double suction pumping unit: *A* – the main supply line DN500; *B* – contractual starting point of suction pipeline; *C* – pipeline DN400; *D* – asymmetric orifice DN400/DN250; *E* – butterfly valve DN250; *F* – compensator DN250; *G* – side bend 3D"; DN250; *H* – compound vacuum gauge; *I* – calming section DN250; *J* – contractual ending point of suction pipeline; *3-Z* – vibrations measurement, bearing 3; *4-Z* – vibrations measurement, bearing 4; *5-X* – vibrations measurement, spiral

The main power supply DN500 (*A*) delivered cold drinking water to the right angle suction pipeline of a pump. Pumped medium flowed through a straight section DN400 (*C*), asymmetric orifice DN400/DN250 (*D*), butterfly valve DN250 (*E*), rubber compensator DN250 (*F*), side bend „3D", DN250 (*G*) and straight pipe with length L = 650 mm and flows to the pump inlet (*J* – cross section marked in yellow).

Research and analysis of results

Following one after another bearings failure caused that the pump was taken to the manufacturer and tested at the factory pump testing station. Its aim was to eliminate the defect of the pump. The results confirmed the correctness of pump design and production, so the reason of failure was sought in the pumping system. During analysing the liquid supply system to the pump in terms of design correctness it was assessed that it was properly done because before the pump calming section was made (*I*). However, the fluid in this section and in fittings installed before the pump flowed at a speed of almost 4 m/s.

With reference to literature, the liquid flow velocity in the well-designed suction pipeline should be between 1 and 1.5 m/s, and the calming section should be as long as possible, between 5 and 7, and in justified cases, even 10 diameters of used characteristic dimension of the pipeline [20, 16].

The least invasive test method in the pumping system was executing of non-invasive vibration diagnosis [15, 17] and analysis of the meters installed in the pumping station. Depending on the type and the way of the pump operation, the permissible vibration level is determined by standards: PN-EN ISO 5199:2014, ISO 10816-3:2009, ISO 10816-7:2009 and ANSI/API 610 [10, 11, 2]. For this type of pumping unit, refer to ISO 10816-3:2009, effective values of vibration velocity should not exceed v_{RMS} < 4.5 mm/s.

In order to explain the reasons for the accelerated wearing of bearings and noisy pump operation at the place of its installation vibration measurements were carried out for different values of water level in the suction tank, for *hp1* = 1.9 m and *hp2* = 4.5 m. Measured effective values of the speed (v_{RMS}) for bearings (*3-Z*, *4-Z*, fig. 2) did not exceed v_{RMS} = 4.5 mm/s but were higher than during the measurements at the manufacturer's pump test station. Then the effective values of acceleration were measured (a_{RMS}) for pump spiral (*5-X*). Obtained results, sorted according to the formula below, defining coefficient K [14], gave result K = 1.8. The increase of vibrations with lowering of the pressure before the pump indicated the presence of cavitation, and obtained value of K coefficient determined unequivocally that it was its 1st phase.

$$K = \frac{a_{RMS,1.9}}{a_{RMS,4.5}} = \frac{13.5\,\frac{m}{s^2}}{7.6\,\frac{m}{s^2}} = 1.8$$

Due to the above, there was a suspicion of high pressure drop on the control section, i.e. from the outlet of the main power supply (*B*, fig. 1) do the pump inlet (*J*). However, observations of the compound vacuum gauge mounted on the suction pipe line (H), for both values of water level, indicated the gauge pressure correspondingly $p_{m1} \cong +0.2 \div 0.4$ bar and $p_{m2} \cong +0.6$ bar. The installed pump was characterized by required anti cavitation surplus $NPSHR_{pp}$ = 5 m which means that it should work correctly at gauge pressure up to $p_m \cong -0.4$ bar. In order to find the cause of the sudden pressure drop the compound vacuum gauge and to illustrate the flow numerical analysis CFD were carried out. Digitized according to the mathematical modelling 3D geometry of the system was implemented in the commercial computational code ANSYS Fluent and simulations of flow in steady-state simulations were carried out [3].

As a result of numerical tests, static pressure maps were obtained in the longitudinal section of the suction pipe line (fig. 3) and pressure drop in control section was determined (*B*÷*J*). The loss of pressure height was h_{los} = 1.2 m (fig. 3), so after taking into account and considering the pressure measured on the compound vacuum gauge (*H*), before the pump inlet (*J*), gauge pressure was accordingly $p_{I1} \cong +0.1 \div 0.3$ bar for h_{p1} = 1.9 m and $p_{I2} \cong +0.5$ bar for h_{p2} = 4.5 m. Obtained results, apparently excluded the possibility of operation in the first stage of cavitation as confirmed at fig. 3.

The next stage of research was to check the flow quality in the considered section (*B*÷*J*). The liquid particle trajectories are shown at fig. 4*a*. The liquid from the main supply

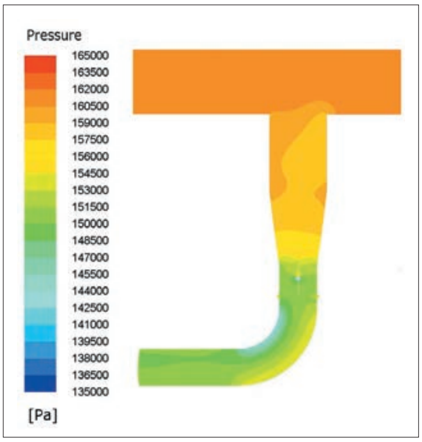


Fig. 3. Static pressure profile in the control section of the suction pipe line

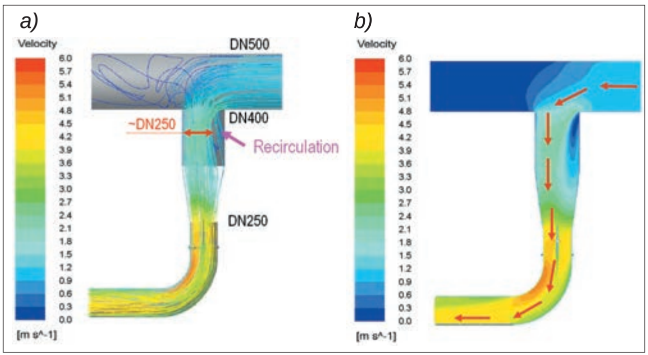


Fig. 4. Suction pipe line, tracks of liquid particles (*a*), speed map in control cross section (*b*)

line DN500 flowed to DN400 in this way that the active part of the cross section in which liquid flowed was reduced to dimension corresponding to pipeline type-size ~DN250. This was due to recirculation, which narrowed the active cross section. As a consequence, the liquid in the pipeline flowed at a higher velocity than it appears from the pipeline cross section, and after flowing through side bend (*G*), caused considerable unevenness of the velocity field in control cross section (*B*÷*J*, fig. 4*b*) and in pump's suction flange (*J*, yellow colour – fig. 1). The numerically obtained speed profile for flow in steady state was presented at fig. 4*b*. Due to the fact that the flow accompanying physical phenomenon was variable over time, another numerical simulation of flow in unspecified state was carried out. It was simulated t = 4.1 s of flow and the result is shown in the form of 4 speed maps, read in cross section (*J*). The results have shown that the speed change profile depends on time. Analysing the obtained velocity profile of the liquid flowing into the suction flange of the pump, significant increase in speed was noticed ($v_{IR} \geq$ ~4,4 m/s) in the right part of the pump inlet (*R*, fig. 5*a*) and decreasing speed in the left part (*L*) of inlet ($v_{IL} \cong$ 3.5 m/s). In addition, the structure of the profile varied over time, causing its instability illustrated at fig. 5*b*.

In double suction pump the liquid after flowing through the inlet section splits into 2 streams which are brought to mirror set inlets of double stream impeller. For capacity Q_{pp} = 700 m^3/h, each impeller inlet should be fed with $Q_{thIL,R} \cong$ 350 m^3/h. Such solution would ensure the balance of axial forces (fig. 6).

Whereas, in the case of the analysed problem, on the basis of obtained speed profile it was assumed that the first part of pump inlet (*R*) is supplied with flow $Q_{IR} \cong$ 410 m^3/h ($Q_{IR}/Q_{thIR} \cong$ 1.17), while second part

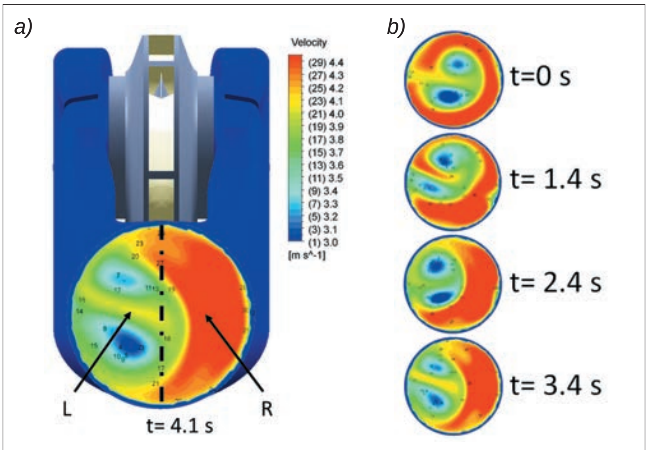
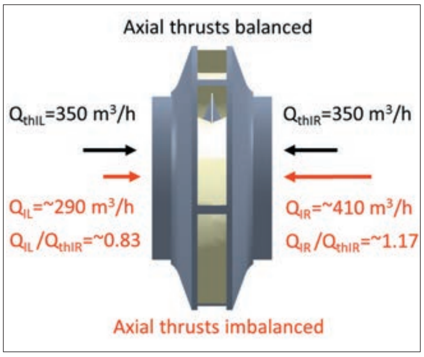


Fig. 5. Pump inlet with speed profile after t = 4.1 s (*a*) and intermediate steps during the analysis (*b*)

Fig. 6. Flow to the double stream impeller



– $Q_{IL} \cong$ 290 m^3/h ($Q_{IL}/Q_{thIL} \cong$ 0.83). Such strong, variable speed field at the inlet to the pump caused the creation of excessive and variable (over time as to the value and operation direction) of the axial forces which were damaging bearings. Referring to estimated capacities of individual impeller's inlets and comparing them with pump reliability chart, it is noticed that half of the impeller fed with capacity ($Q_{IL}/Q_{thIL} \cong$ 0,83), operates in the field of increased vibration caused by recirculation of the liquid at the pump outlet, shortening optimal operating time to ~58% in comparison to its computational life time (fig. 7). The second part of impeller worked with excessive capacity ($Q_{IR}/Q_{thIR} \cong$ 1.17) in the area of increased wearing of bearings and seals and close to the first phase of cavitation, as indicated by previous vibration measurements. Cavitation, associated with increased flow velocity, affected on increased vibrations, which further reduced the life time of bearings up to ~2% (fig. 7).

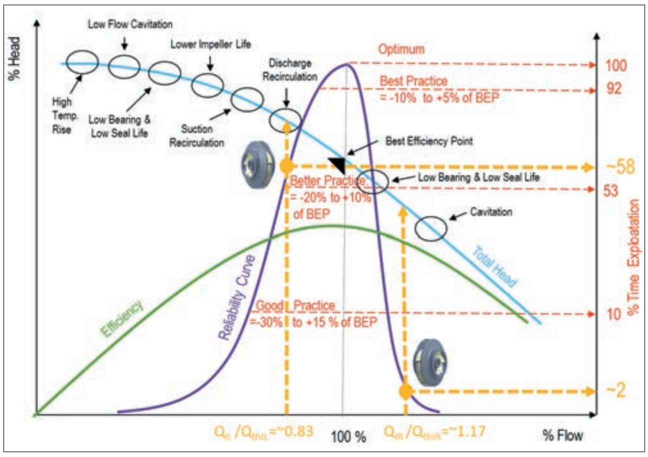


Fig. 7. A summary of the operation of both pump's impeller inlets with a pump reliability chart

On the basis of carried out simulations and analyses, taking into account local installation conditions, the pipeline on the suction side has been rebuilt. Considered test section together with elbow was made in size DN400, and only before the pump inlet an asymmetric reducer was applied. The visualization of speed maps in this section obtained by numerical calculations is illustrated at fig. 8*a* and 8*b*. The speed profile in the inlet section of the pump has been aligned, and difference between minimal and maximal speed was only $\Delta v_l \cong$ 0.3 m/s. Due to the fact that there was no possibility of rebuilding the main supply line connection (*A*) with considered section of the suction pipeline, the recirculation zone in area (*B*) was not eliminated. However, overriding and possible for implementation aim in the presented layout was to align the speed profile at the pump inlet (fig. 8*a* and 8*b*) this was achieved by lowering the speed in the elbow and fittings and the

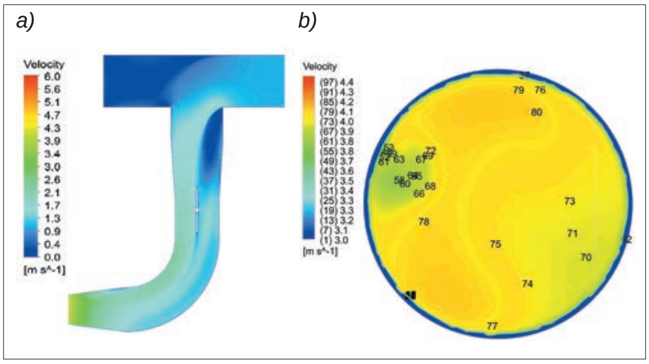


Fig. 8. Speed maps in cross-section (*a*) and in pump inlet cross-section (*b*)

Fig. 9. Implementation of proposed suction line design

liquid was accelerated only before the pump inlet. Confirmation of the rightness of conducted research was its verification on the target working station of the pump. The implementation of the proposed design solution is shown at fig. 9 and it completely eliminated the problem.

Conclusions

The presented problem and the correct solution to the problem concerning incorrect conduction of the suction line to double suction pump illustrates how important it is to ensure proper conditions and hydraulic parameters before the pump inlet. Failure to do this often results in improper pump operation, short life time and increased operating costs. In the event of a breakdown, it is extremely important to properly define the reasons of the breakdown. Removal of these reasons will ensure pumping aggregate operation continuity.

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