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Numerical Analysis Of Cavitation Due To The Recirculation

Heavy Duty Process Pumps

Andrej Lipej
Faculty of technologies and systems,
Novo mesto, Slovenia
andrej.lipej@fts-nm.si

Duško Mitruševski
SM Pumps,
Ljubljana, Slovenia
dusko.mitrusevski@gmail.com

***Abstract* — Design of heavy duty process pumps usually based on various design criteria depends on pumps application. Cavitation due to the recirculation is not often mentioned as design criteria although many problems in pump operation appear because of the cavitation due to the inlet vortices [1]. Cavitation due to the recirculation is important criteria to design the heavy duty process pumps. Operation range with partial flow between 0.5 - 0.8 Q_{opt} is very often in pumping systems.**

Computational Fluid Dynamics - CFD analysis is very important approach to design optimal casing/impeller inlet geometry to avoid recirculation in the operating range of pumps in the systems.

In this article cavitation due to the recirculation is numerically analysed as design criteria. The paper gives an overview of operating conditions in various pumping systems of the old split casing pump where damage due to the recirculation cavitation at part load occur. New hydraulic design for pump is developed to improve the operating range and to avoid the harmful influence of recirculation on partial flow.

Keywords — pump; cavitation; recirculation; CFD

Flow recirculation at the inlet of centrifugal pump may cause a noise, vibration, erosion damage and large forces on the impeller [2]. It can also cause the cavitation due to recirculation. The chance of damage is heavily dependent on the suction energy level [3], specific speed of the pump, the NPSH margin in the pump and the nature of the flow provided to the suction piping. According to experience that low suction energy pumps are not susceptible to damage from suction recirculation. Solid particles and corrosives can increase damage during suction recirculation, similar as with classic cavitation, even the NPSH characteristic is good.

A lot of papers dealing with CFD analysis of the classical cavitation problems due to NPSH characteristics can be found in the literature. But there is not a lot of papers dealing with the numerical analysis of the cavitation due to recirculation.

The article presents the cavitation in a centrifugal pump, resulting in partial discharges due to swirl at the inlet of the impeller. In normal situations cavitation arise at the suction side of the impeller blade because certain conditions appears to the pressure reduction, which can be equal to the evaporation pressure. In most cases, the development of centrifugal pumps are only able to analyse conditions concerning cavitation along the walls of the blades. Cavitation due to vortex is analysed in a very small number of cases. Even this type of cavitation may cause material damage, particularly on the pressure side of the blades, which are usually no problems with cavitation.

Designing the inlet geometry of heavy duty process pumps is important to define the operating range free of recirculation at partial flow. Cavitation due to the recirculation reduce the level of pump reliability and danger of impeller damage is substantial although NPSH available is much higher than NPSH required of the pump. Operating conditions of the pumps in the system dictate technical solution of inlet geometry of the casing and impeller. Under part load operation Q << Q_{opt} recirculation vortex occurs at impeller inlet.

Recirculation free operating range of the pump depends on:

- Specific speed n_q
- Suction specific speed SS
- Inlet geometry of the casing / impeller

Designing the impeller hydraulic with wide recirculation free operating range is not an easy task. Even more, testing the recirculation range and intensity on model pumps on the test rig is demanded procedure. Computation fluid dynamic combined with experience and testing result statistics could be a very effective method for design the recirculation free range for pumps with different specific speed n_q.

I. THEORETICAL OVERWIEW

Cavitation and recirculation in the centrifugal pumps have a significant effect on pump performance and pump life time. Recirculation at part flow at impeller inlet directly influence reliability of the pump and limits the operating range.

Damages appear on the pressure side of impeller inlet blade because of cavitation due to the recirculation. Hydraulic design of pump should prevent many negative effects of cavitation due to the recirculation in pumps. Basic criteria for recirculation free range is suction specific speed SS of the pump.

$$SS = nQ^{0.5} / NPSH^{0.75} \quad (1)$$

SS is defined for BEP and 3% $NPSH_{req}$.
Normal values of SS:

- SS = 160-220 for axial inlet impellers,
- SS = 220-280 for suction impeller with axial inlet.

Design criteria for SS is usually more important than criteria for high level of efficiency, but for recirculation free range suction specific speed should be optimized with the value of maximal allowed $NPSH_{req}$ of the pump.

Other important parameters (Fig. 1) for recirculation free range are as follows:

- Impeller inlet diameter D_1
- Impeller hub diameter D_o
- Impeller blade inlet angles.

In general parameters, improving $NPSH_{req}$ and higher SS increase the recirculation range and limit pump safety operating range.

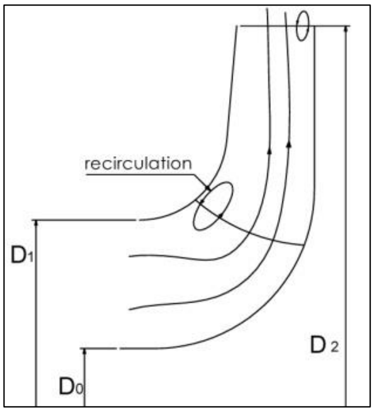


Fig. 1 *Position of inlet recirculation*

Pump recirculation can cause surging and cavitation even when the available $NPSH_a$ exceeds the supplier's published $NPSH_{req}$ by considerable margin. Suction recirculation typically produces a loud crackling noise in the pump. Recirculation noise is of greater intensity than the noise from low NPSH cavitation and is a random knocking sound. Discharge recirculation will produce the same characteristic sound as suction recirculation with the exception that the highest intensity is noticed at the discharge volute or diffuser.

II. EXISTING HYDRAULIC GEOMETRY

Cavitation damage due to the recirculation happens very often in the pump operation. One example is water cooling pump with wide operation range and very often operating point reach recirculation range [4].

Basic pump data:

- Water cooling split casing pump n_q = 40
- Medium: Water 62 °C
- SS = 206 (for half flow rate)

Pump operates in the range 0.7-1.0 of Q_{opt}. Although available $NPSH_a$ in the system is higher (Fig. 2) than $NPSH_{req}$ of the pump, cavitation due recirculation can damage impeller blades (Fig. 3).

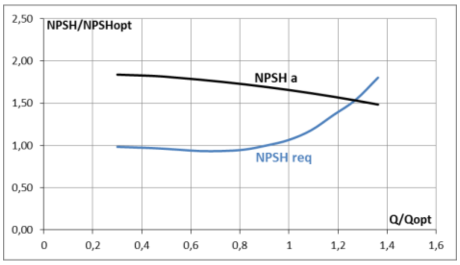


Fig. 2 *Dimensionless Q-NPSH characteristic of the existing pump*

Fig. 3 *Damage of impeller of split casing pump because of cavitaiton of recirculation*

Damages due to recirculation cavitation on the existing pump are presented on the Fig. 3. Usualy damages are only near the suction side of the impeller blade. In this case demages are seen at the pressure side of the impeller blades [4]. From the existing cavitation characteristics there was no cavitation damage expected for this pump.

III. Design of new geometry using CFD analysis

The main task was making a design of the new pump impeller with less recirculation at the inlet and to avoid the cavitation due to recirculation. With this purpose some new impellers were designed and the final geometry is presented on the Fig. 4.

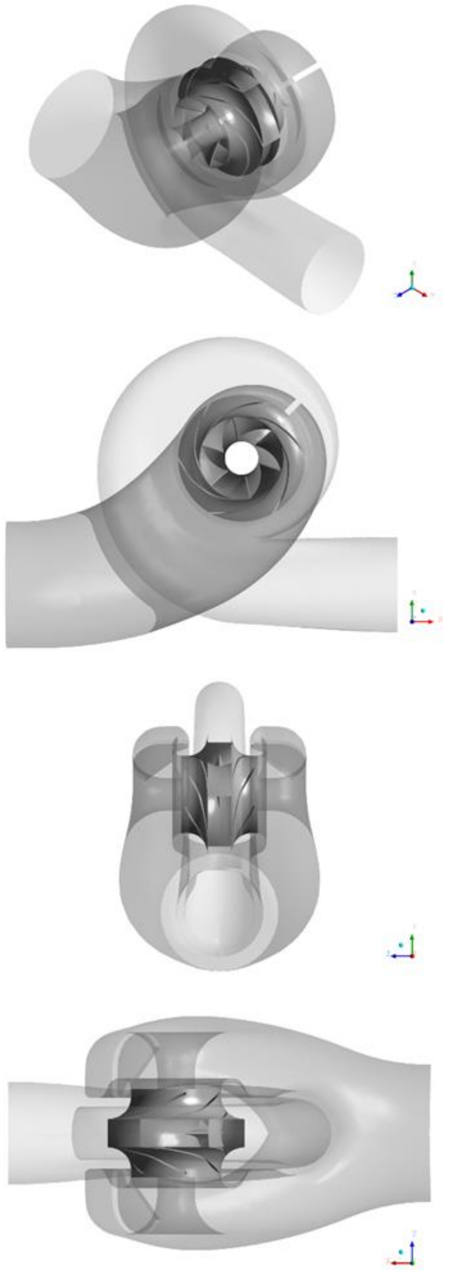


Fig. 4 *Four views of the computational domain*

Some changes on the existing impeller geometry have been done to obtain improved cavitation characteristics.

For the new designed geometry the computational grid (Fig. 5) was generated in order to find out the energetic and cavitation characteristics and to predict the location, intensity and operating regime for recirculation appearance.

The quality of computational grid is an important condition for accurate numerical flow analysis results, particularly in case of dominant decelerating flow in the pump [5].

Grid refinement is very important near the impeller walls. Besides the grid refinement, the special attention has been done on the grid quality parameter - y^+, on mesh orthogonality and on expansion and aspect ratio. In case of analysed pump, the y^+ in the almost whole computational domain is between 20 and 50. Just in the very small area, which is not relevant on the accuracy of the results, the y^+ exceed 50 (Fig. 6).

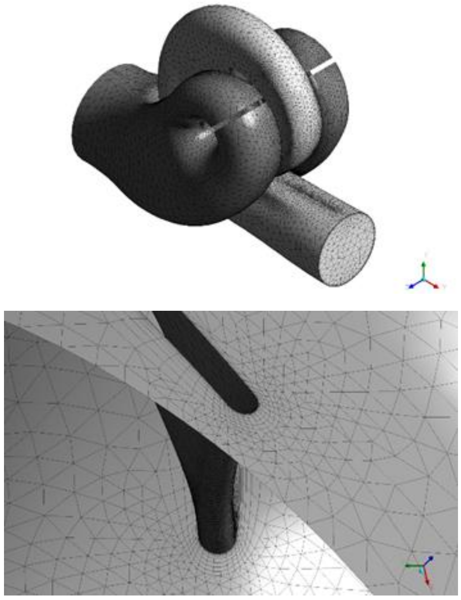


Fig. 5 *Computational grid – complete pump (above); impeller inlet (below)*

The computational grid was done separately for inlet chamber, impeller and spiral casing. Special attention on the mesh quality was paid to the impeller grid generation and this is the main reason that the number of elements in the impeller is more than a half of the complete computational grid.

Computational grid data:

- Total number of elements – 3.500.000
- Impeller number of elements – 2.000.000.

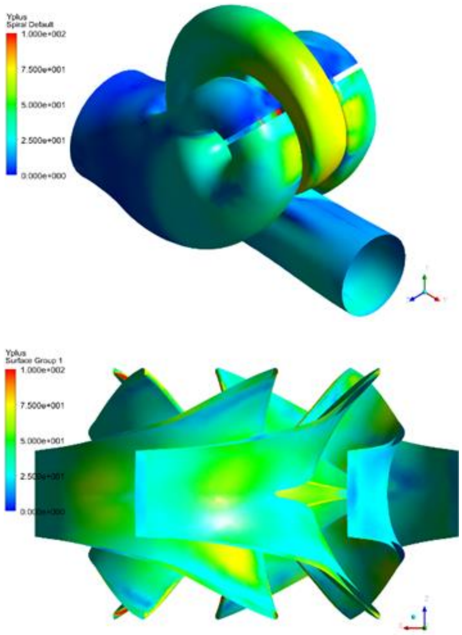


Fig. 6 *Distribution of y+ - complete pump (above); impeller (below)*

The numerical analysis in pump (Fig. 7) has been performed for four different flow rates presented on the table 1.

TABLE I.

Operating point	Flow rate [m^3/h]
A	650
B	830
C	1000
D	2000

Operating regimes

The important issue in CFD analysis is the turbulent model. In our case the k-omega SST turbulent model was used.

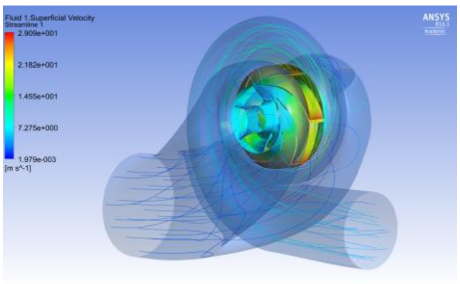


Fig. 7 *Flow distribution in inlet chamber and spiral casing*

Usually for the accurate prediction of pump characteristics unsteady analysis is required, but in this case decision of steady state calculations were appropriate, because we just wanted to improve cavitation characteristics due to inlet recirculation and did not want to obtain accurate absolute values of efficiency or distribution of losses in each part of the pump. The relative comparison between old and new designed pump shows satisfactory results.

New hydraulic design was developed to provide wider operating range without recirculation and to reach operating range around BEP.

CFD analyses is done for more partial flow rates when Q << Q_{opt} as follows:

- Point A - 0.28 Q opt
- Point B - 0.38 Q opt
- Point C - 0.45 Q opt
- Point D - 0.91 Q opt.

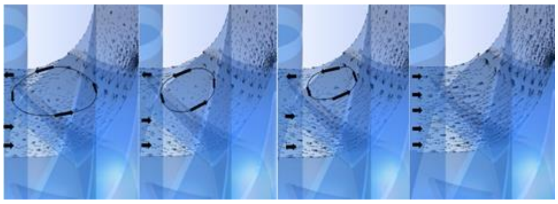


Fig. 8 *Recirculation vortex for operating points A, B, C and D (from left to right)*

If the damage occur at the hidden (pressure side) side of the impeller and cannot be seen without the use of special experimental device, the most probable cause is suction recirculation. Classic cavitation damage usually occurs on the visible (suction side) side of the impeller, close to the leading edge. Using CFD analysis it is possible to locate the recirculation zone quite accurately and to find out the intensity and size of the vortex region (Fig 8 and Fig. 10).

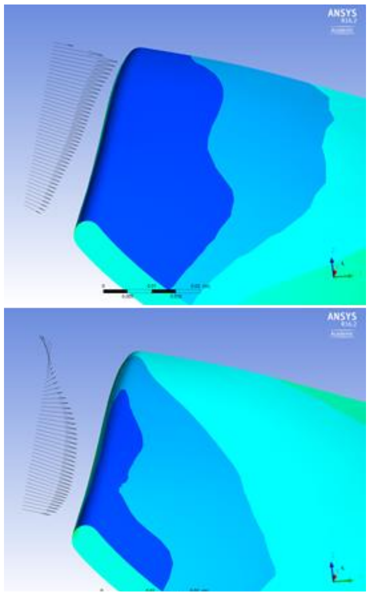
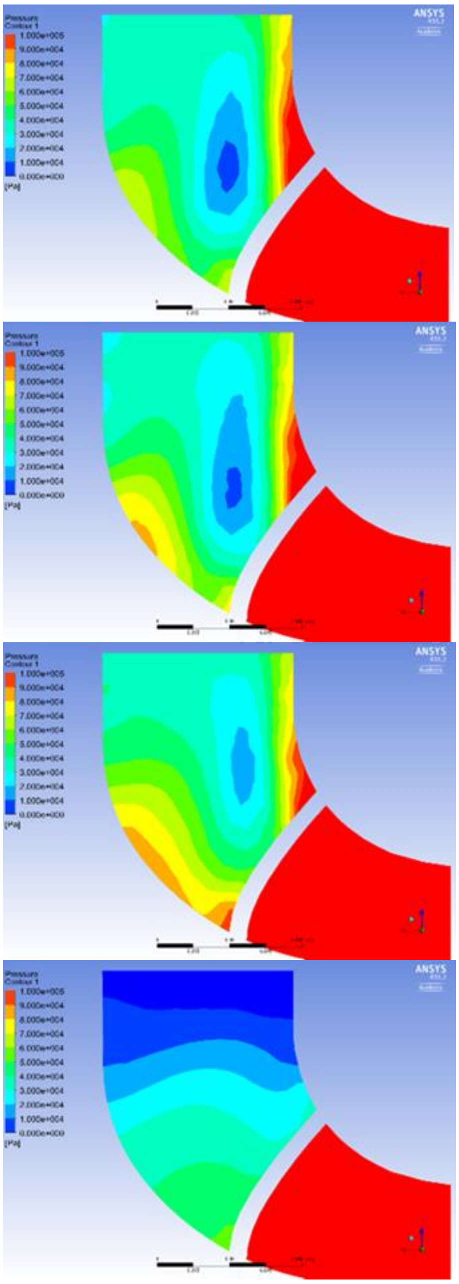


Fig. 9 *Velocity vectors near leading edge of the impeller – OP A (bellow), OP D (above)*

Inlet recirculation at part load is one of possible reasons for the cavitation damage. But at this operating regime the angles of attack of the velocity



vectors near the leading edge are not optimal (Fig. 9).

Fig. 10 *Pressure distribution in the recirculation zone for operating points A, B, C and D (from top to bottom)*

When the flow rate is smaller than BEP, the inlet velocity distribution is more irregular. Near the hub the velocity distribution remain only the same as near optimal flow rate. For the radius bigger than the half of inlet channel the velocity drops significant and this is the reason for the back flow and vortex occurrences. The axial velocity component distribution at the impeller inlet is presented for two operating points on the Fig. 11.

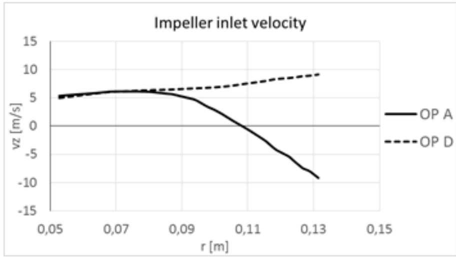


Fig. 11 *Impeller inlet axial velocity component distribution for two operating regimes*

New designed pump has better Q-H characteristics. The head in part load is higher (Fig. 12) and the influence of the inlet recirculation is much smaller than in case of the old pump geometry (Fig. 13).

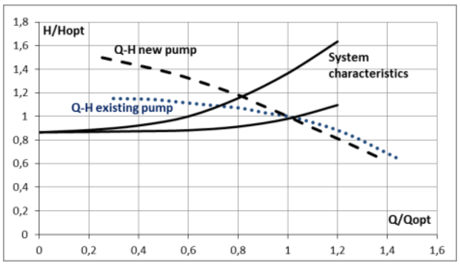


Fig. 12 *Dimensionless Q-H characteristic of the new pump compared with existing one*

Small recirculation zone can be visible at the operating point C, but the real problems with the cavitation can start only at operating point A which is not very important operating regime. The new designed pump can operate at much wider operating regime without cavitation due to recirculation, when harmful recirculation appears much earlier around Q/Q_{opt}=0.7, in comparison with the old pump.

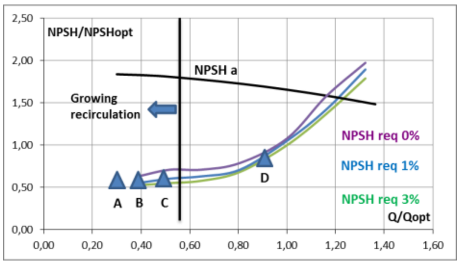


Fig. 13 *Dimensionless Q-NPSH characteristic of the new pump*

V. Conclusions

Each impeller design has its specific recirculation characteristics which can be improved only with the new hydraulic design. The paper shows the basic characteristics of the existing pump and consequences of the cavitation damage due to recirculation. After the optimization procedure the improved pump was obtained and the results of new geometry show better cavitation characteristics especially minimized recirculation range in the part load operation. The cavitation characteristics of the pump were improved using the CFD analysis.

There are also some other possible improvement interventions, which are capable to minimize the recirculation consequences:

- Increasing output capacity,
- bypass between the discharge and the suction side,
- harder and better impeller material.

All above mentioned corrections can reduce the intensity of the cavitation damage.

Before starting the development procedure it is necessary to diagnose the location and intensity of recirculation zones for different operating regimes. Geometry improvement can be done the most efficiently with the accurate CFD analysis presented in the paper. Each step of CFD procedure is explained in order to obtain reliable and acceptable results for industrial projects.

NOMENCLATURE

D_o	[m]	Impeller hub diameter
D_1	[m]	Impeller eye diameter
H	[m]	head
H_{opt}	[m]	optimum head
n	[rpm]	revolution per minute
$NPSH$	[m]	net positive suction head
$NPSH_a$	[m]	available net positive suction head
$NPSH_{req}$	[m]	required net positive suction head
$NPSH_{0\%}$	[m]	net positive suction head at 0% head drop
$NPSH_{1\%}$	[m]	net positive suction head at 1% head drop
$NPSH_{3\%}$	[m]	net positive suction head at 3% head drop
Q	[m^3/s]	flow rate
Q_{opt}	[m^3/s]	optimum flow rate
SS	[-]	suction specify speed

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