The optimization of a low specific speed pipeline pump

A Zhao, P Wu, D Z Wu and L Q Wang
Institute of Process Equipment, Department of Chemical and Biochemical Engineering, Zhejiang University, Hangzhou, 310027, P. R. China
E-mail: wudazhuan@zju.edu.cn

Abstract. A low specific speed pipeline pump is researched to improve work performance through several certain modifications. The target is to raise the pump’s head up to 80m and increase its total efficiency to 26.5% when the volume flow rate is 6.3m³/h with a rotatory speed of 2960rpm. CFD Numerical simulation is employed to predict the hydraulic performance of the pump. The volute is redesigned and splitters are imported to change the structure of the impeller. A lot of factors are taken into account, for instance number of vanes, forms of vanes, width of the volute, shape of volute’s cross section, etc. These transformations ameliorate the distribution of pressure and velocity in the impeller and volute that finally increase the pump’s hydraulic efficiency. The impeller trim has also been made according to the affinity law to fatherly decrease the disc friction which causes pump’s mechanical loss and then achieves the optimization goal finally. It is a practical case in low specific speed pump’s optimization and conclusions given at the end may be experience in the design of this sort of device.

1. Introduction
Low specific speed pumps are widely used in agricultural irrigation, petrochemical industry and many other fields to deliver small flow rate and high head liquids. However, low specific speed pumps designed by the conventional method [1] frequently present too narrow impeller passage height at the outlet to manufacture [2] and undesirable performance, such as head curve’s hump character and low efficiency. So a lot of researches, e.g. Kelder and Visser [3, 4], include experimental study, theoretical analysis and numerical simulations have been done on overcoming low specific pumps’ shortcomings.

This paper aims at performance improvement of a pipeline pump as an application instance of several already built improvement approaches based on some semi empirical conclusions of predecessors’ work.

2. Optimization objective and the archetype
The low specific speed pipeline pump under discussion is a single suction, single stage centrifugal pump. Parameters of the optimization objective at Best Efficiency Point (BEP) are listed in Table 1.

| Q, m³/h | H, m | n, r/min | Efficiency |
|--------|------|----------|------------|
| 6.3    | 80   | 2950     | 26.5%      |
The archetype of the pipeline was designed by enlarged flow method. Therefore, the best efficiency point is larger than 6.3m³/h. Though take this path may get the advantage of the large pump’s stabilization and high efficiency among a big extent, it will cause many negative factors. Because the nature of the enlarged flow design method is having a higher specific pump operating in a non-optimal condition to meet the requirement of external pipes, certainly it will demand more shaft power at the same flow rate and more materials in the casting that is not economical friendly. Even worse the motor may overload when it comes to the large flow rate [5]. In order to avoid these drawbacks, further research is needed to be done.

3. Numerical simulation process

3.1. Geometrical model

The fluid zone is comprised of a closed impeller, a vaneless volute and the clearance between casing and shroud or hub [6], which can be seen in Figure 1. Some main variables of the pipeline pump are shown in Table 2, where symbols are consistent with the manual [7].

![Figure 1 Numerical simulation’s fluid zone](image)

| Table 2 Archetype pump’s main geometrical parameters |
|-----------------------------------------------|
| $D_2$, mm | $b_2$, mm | $\phi$ (°) | $\beta_2$ (°) | $Z$ | $F_t$, mm² |
|---------|-----------|-------------|---------------|----|-----------|
| 250     | 4         | 210         | 40            | 4  | 261.3     |

3.2. Numerical simulation method and conditions

The commercial Computational Fluid Dynamics (CFD) software ANSYS-Fluent has been used to forecast the pump's hydraulic performance and the internal flow condition in detail. Water is assumed to be viscous, incompressible and isothermal. For turbomachinery's rotor-stator interaction is relatively weak, the Multiple Reference Frame (MRF) model is used so as to make the solution a steady-state problem and interfaces are used to transfer flow field parameters between the moving and stationary zones. Thus the absolute velocity form is chosen to solve the governing Navier-Stokes (N-S) equation that will result in most of the flow domain having the smallest velocities to reduce the numerical diffusion in the solution and lead to a more accurate solution. The standard wall function is utilized near the wall and the realizable k-epsilon model is used to describe the turbulence approximately. Velocity-inlet, pressure-outlet and no-slip wall with 100μm roughness boundary condition, second order upwind discretization scheme, Semi-Implicit Method for Pressure Linked Equations Consistent (SIMPLEC) algorithm and segregated Finite Volume Method (FVM) solver [8] are appropriate experience selection to emulate the actual flow field of the low specific pump as far as possible.

3.3. Check of grid independence

As shown in Table 3, mesh refinement gives rise to few changes in pump's hydraulic performance at design point that can be neglected in industry production. Therefore, the check of grid independence passes and numerical simulation latter shall elide this procedure.
Table 3 Grid independence

|                | H, m | N, W   | Efficiency |
|----------------|------|--------|------------|
| 30k cells      | 93.6 | 3660.6 | 43.8%      |
| 60k cells      | 92.6 | 3560.2 | 44.6%      |
| Rate of change | 1.1% | 2.7%   | 1.8%       |

4. Improvement methods and results

4.1. Redesign of the volute
Because low specific pump’s volute determines much of the environment in which the impeller works and sometimes it has a profound effect on impeller performance. If the volute doesn't match the impeller, the departure from a free vortex velocity distribution in the volute will produce a circumferential pressure gradients at the impeller periphery so that the flow in each impeller bade passage pulsates as the impeller rotates in the casing. This causes a variation in the total head generated, and the subsequent mixing of streams of different total energy will be accompanied by energy losses: it can also cause reverse flow from the volute back into the impeller at low deliveries [9]. Just like the static pressure distribution contours shown in Figure 3, a pressure drop can be observed, which means a bad interaction between the stator and the rotor. So as to eliminate this negative factor to decrease the energy loss, a new rectangular section volute, some main parameters can be seen in Figure 2, has been designed according to the manual [7]. Figure 3 and Table 4 shows it has dispelled the pressure drop and attained a higher hydraulic efficiency.

![Figure 2 Redesigned volute's 10 sections](image)

![Figure 3 Static pressure distribution contours of original and volute-redesigned pumps](image)
Table 4 Hydraulc performance of original and volute-redesigned pumps

|                      | H, m | N, W | Efficiency |
|----------------------|------|------|------------|
| Original pump        | 83.0 | 2695.8 | 52.7%      |
| Volute-redesigned pump | 90.5 | 2608.4 | 59.4%      |

4.2. Adding splitters

Low specific speed pump always has the jet-wake flow structure in the impeller passage, whose outline is sketched by a black solid curve in Figure 5, caused by the Coriolis force produced by the impeller's rotating and the curvature of the vane. It consists of the wake flow zone that has relatively small velocity and jet flow zone that can be treated as the potential inviscid flow [5]. In the wake flow zone, the boundary layer inclines to separate from the suction side wall of the vane because of the adverse pressure gradient. Then backflow and separate flow arise and causes severe energy loss, even worse in the lower flow rate [10].

A lot of researches, e.g. [11] have pointed out adding splitters may improve this situation and countless experiments, numerical simulations have been made to find the optimal solution, so this paper just adopts some existing empirical principles to modify the impeller. 2 degrees of the outlet offset angle and 0.6 long vanes' length of the splitters' lengths are chosen as the reference value. We can see from Figure 5 that adding splitters suppresses the formation and development of the wake flow zone and the axial swirl. Figure 4 indicates that turbulence has been weakened and homogenized through the entire flow field in the impeller, either. Nonetheless, splitters increase slip coefficient and decrease excretion coefficient simultaneously which raises the head but may block the impeller passage resulting in the bigger impact loss. For this sake a pump model modified by reducing the blade number is conducted as well, which can be seen in Table 5, and it has a slightly lower head and a better hydraulic efficiency.

Figure 4 Turbulence kinetic energy distribution contours of original and splitters-added pumps

Figure 5 Relative velocity distribution vectors of original and splitters-added pumps
Table 5 Hydraulic performance of original and splitters-added pumps

|                        | H, m | N, w | Efficiency |
|------------------------|------|------|------------|
| Original pump          | 90.5 | 2608.4 | 59.4%      |
| Splitters-add pump (5+5)| 92.8 | 2610.1 | 60.9%      |
| Splitters-add pump (4+4)| 91.9 | 2549.9 | 61.7%      |

4.3. Impeller trim

According to the affinity law of impeller trim of the low specific speed pump [7],

\[
\frac{H'}{H} = \left(\frac{D_2'}{D_2}\right)^2
\]

(1)

the impeller outer diameter is cut to 230mm to obtain the final model of the pump whose head has a suited allowance without large change of shaft power and hydraulic efficiency. It's worth noting that the diameter of base circle is subtracted the same difference to maintain the similar flow field in the volute.

Figure 6 Wooden pattern of the trimmed impeller

Figure 7 Comparison of hydraulic performance between the archetype pump and the optimized pump
Compared with the archetype pump in Figure 7, the hydraulic efficiency curve shifts to the left, the power curve moves down and the head curve becomes “soft”. To sum up, use equation (2) and equation (3) to estimate the disc friction loss is 55.3% and volume loss is 90.6% and suppose the bearing and packing seal loss is 2%.

\[
P_{\text{m}} = 0.07 \left( \frac{100}{n_i} \right)^{3/6} \quad (2)
\]

\[
\eta_i = \frac{1}{1 + 0.68n_i^{-2/3}} \quad (3)
\]

Thus the optimized low specific speed pipeline pump has an 84m head a 30.6% total efficiency at the design point of a 6.3 m³/h volume flow rate with the 2950rpm motor speed, showing the optimization objective has been achieved.

5. Conclusion and prospect
This paper presents an application instance of several low specific pump’s improvement approaches based on several semi empirical conclusions that may be experience in the design of this sort of device:

1) Redesigning the volute. It dispels the pressure drop and make the flow field in the volute more ideal;
2) Adding splitters. It suppresses the formation and development of the wake flow zone, the axial swirl, weakens the turbulence and homogenizes the entire flow field in the impeller;
3) Trimming the impeller. It attains a suitable head and a higher hydraulic efficiency.

To acquire a more profound quantitative understanding of low specific pipeline pump’s internal flow field and hydraulic performance in the future, areas ratio principle, boundary layer control theory and intelligent optimization methods shall be taken into account in the optimization procedure.

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