

Effect of hydraulic parts construction arrangement on the overall performance of API 610 VS6 type pumps

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Abstract. The article presents the results of computing investigations of various construction arrangements of the VS6 type pumps. Variants of hydraulic parts differing in geometric parameters have been compared. Experimental performance curves, measured during the full-scale pump tests, corroborate the investigation results for two constructions of hydraulic parts.

1. Introduction

At present, a demand for higher level of economical operation of centrifugal pumps becomes more and more vital. In case of pumps with a high power input (over 1000 kW) even a 1 to 2 % increase in the efficiency has an essential impact on their power consumption. Savings of electric energy may reach the figure of 80 thousand kW per year.

Certainly, one of the most common ways of raising the economical operation of centrifugal pumps is to reduce friction losses in the hydraulic passages. Mostly, it implies an enhancement of hydraulic design of impellers and a reduction of friction losses in the outlet hydraulic passage.

This article deals with methods of increasing pump efficiency, which relate mainly not to hydraulic parts themselves, but their mutual influence. Investigations were conducted on the basis of the existing design of a VS6 type pump according to API 610 [1].

2. Theoretical basis

VS6 type pump is a vertically suspended diffuser radially split double casing centrifugal pump. The suction and discharge nozzles of the pump are arranged horizontally and oriented in the opposite directions. The pump consists of a can and a withdrawable element of (multistage) ring section cartridge type, featuring a diffuser downstream of each impeller. In view of high cavitation performance (low NPSH value) of the pump, an inducer is typically arranged immediately upstream of the first stage.

As VS6 type pumps have such design features (vertically suspended, low NPSH pumps), they find their applications as condensate extraction pumps in the power generation industry as well as booster pumps for main pipeline pumps.

While designing new machines, with VS6 type pump design features in mind, it is not economically sound to modify outer casings (cans), as frequently only cartridge (rotating assembly) is supplied for the cans already existing in site.



3. Methods of investigation

In order to improve the economical operation of centrifugal pumps, the design of their hydraulic parts is tried out and developed on a multi-purpose experimental test stand that enables to investigate one, two or three pressure stages of the pump. Such test stands do not replicate in all details design features of the pump, for which hydraulic parts are tried out and developed, and do not allow to take account of mutual influence of design features of a full-scale pump. Making experiments on a real pump is a too costly process. Therefore, in this case a virtual experiment is the most efficient method of investigation. The Nasosenergomash Company applies various software products for pump investigations, the most widely used of them being PumpLinx and ANSYS CFX.

PumpLinx software makes it possible to obtain rather quickly and easily integral characteristics of any particular pump on the whole and even elementwise, but not analyzes in depth the flow in the hydraulic passages with the purpose of revealing potential areas to be optimized. ANSYS CFX software offers in-depth studies of the flow pattern in the elements of hydraulic parts with allowance for their interaction.

Thus within the scope of this research the PumpLinx was used to perform a prompt relative assessment of the effect of one or another modification on the performance characteristics of the pump on the whole, and the flow pattern in the individual elements of hydraulic parts was investigated in ANSYS CFX. The use of ANSYS CFX enabled to obtain a more comprehensive flow pattern in the same and to determine locations where corrections (adjustments) are to be made.

4. Analysis of investigation results

The analysis was carried out for two construction arrangements of hydraulic parts. The following elements, namely: a suction casing, a first stage impeller combined with an upstream inducer, an intermediate stage impeller, a last stage impeller and a flow transfer passage between the first and subsequent pressure stages were optimized in terms of reduction of hydraulic losses.

Suction casing. When analyzing the configuration of its suction chamber, two variants were studied: P1 – the variant, which had been earlier applied in the serially produced pumps of the similar type; P2 – an updated variant, in which the suction chamber is axially more extended and thus offers a smooth approach flow to the inducer. In addition, the P2 suction casing features integral flow straightener ribs having outlet tips tapered off to minimize the length of their downstream vortex wake (see Figure 1).

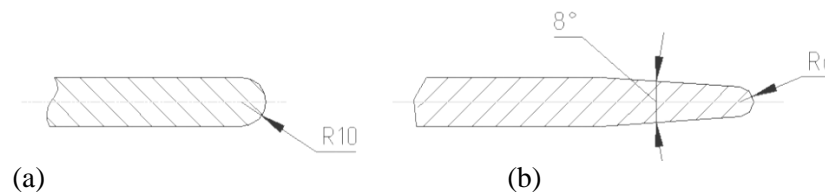


Figure 1. Rib Cross Section in the P1 (a) and P2 (b) Suction Casings.

Figure 2 shows a flow pattern in the P1 suction casing as against that in the P2 suction casing. For comparison, we may mention that the P2 suction casing has a more uniform flow pattern.

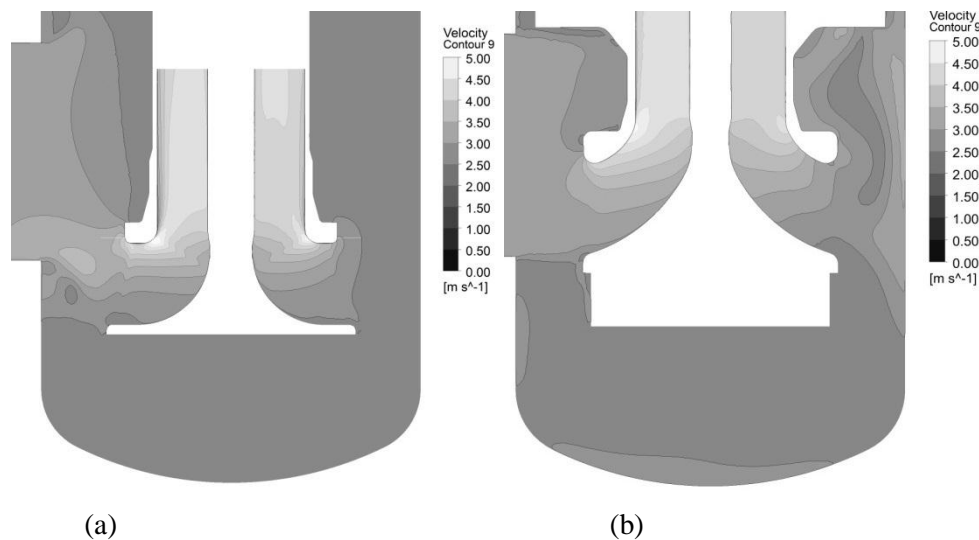


Figure 2. Flow Pattern and Velocity Distribution in the P1 (a) and P2 (b) Suction Casings.

First stage impeller combined with an upstream inducer. In order to ensure a cavitation-free operation (with a low NPSH required), the first pressure stage of pumps of such type is equipped with a suction impeller in combination with an inducer installed upstream of the same. Besides that, the first stage is frequently lowered, i. e. is arranged at the bottom of the can (caisson) at a considerable distance from the rest of pressure stages. Within the scope of this research two construction arrangements of the first stage were compared: arrangement No. 1 with an inducer located at the bottom of the can and the first stage impeller elevated up to the impellers of subsequent stages, and arrangement No. 2 with all the first stage, comprising an inducer and a suction impeller as well as its diffuser, lowered down to the can bottom. A spacer with flow transfer passage is provided between the first stage diffuser and the impeller of the subsequent stage for maintaining the axial length of the cartridge (a withdrawable element).

A relative position of the inducer and suction impeller close to each other in the arrangement No. 2 allows to apply an inducer generating a lower head. Besides that, losses in the flow transfer passage between the inducer and suction impeller, which occur in case of the arrangement No. 1, are eliminated.

In addition, an inducer without a cuneate ledge was applied in the arrangement No 2 for loss reduction.

Figure 3 shows 3D images of fluid models of the first stage incorporating a suction impeller and an upstream inducer.

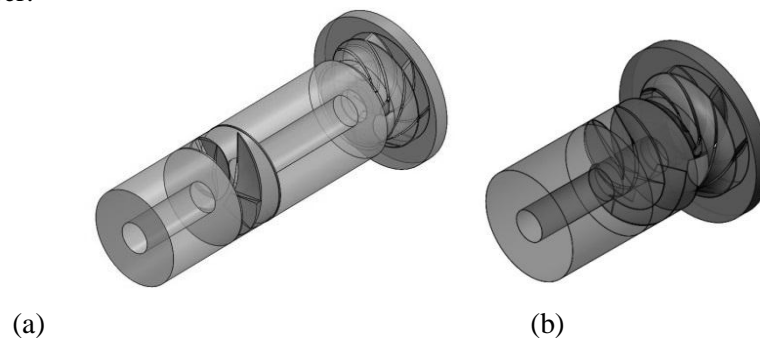


Figure 3. Fluid Models of the First Stage Incorporating Suction Impeller and Upstream Inducer in the Arrangement No. 1 (a) and in the Arrangement No. 2 (b).

Flow fields of distribution of relative velocities in the blading cascades of the first stage arrangements No. 1 and No. 2 are represented in Figure 4 and Figure 5. It should be mentioned that the arrangement No. 2 has a more uniform distribution of velocities both in the inducer and in the suction impeller.

By the numerical calculation it was established that the application of the combination of the suction impeller and upstream inducer according to the arrangement No. 2 resulted in a 11.5 % gain in efficiency as compared with that produced in case of the arrangement No. 1 due to the reduction of power consumption.

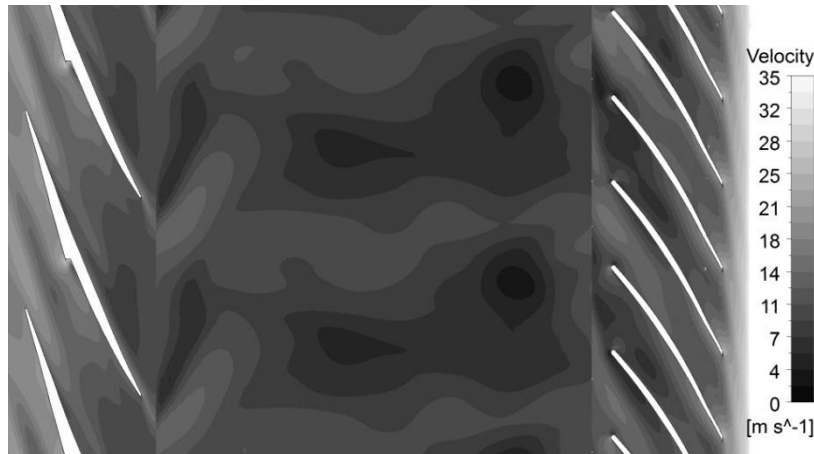


Figure 4. Relative Velocity Distribution in the Blading Cascade in Case of the Arrangement No. 1 First Stage.

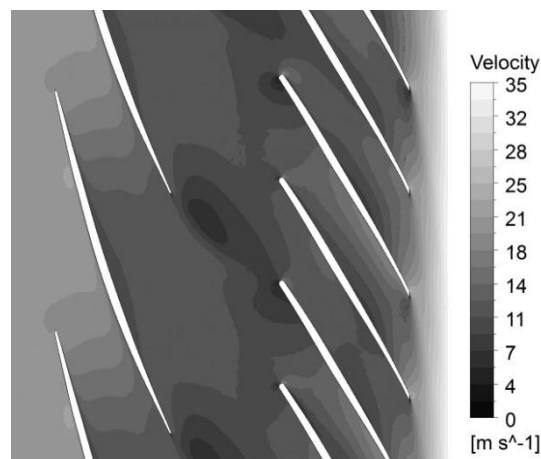


Figure 5. Relative Velocity Distribution in the Blading Cascade in Case of the Arrangement No. 2 First Stage.

Intermediate stage and last stage impellers. Within the scope of this research we studied the flow pattern in the intermediate stage impeller and last stage impeller and optimized their geometry.

We compared two variants of impellers: RK 1 (a standard one) and RK 2 (an upgraded one) with each other. The geometry of the upgraded impeller RK 2 differs in the suction eye diameter as well as in the entrance angle and outlet angle of impeller blades. A comparison table containing major geometric dimensions of the impellers is given below.

Table 1. Major Geometric Dimensions of the Impellers.

	D_h/D_2	D_1/D_2	W_1/W_2	β_{2b}/β_{2f} , degrees	β_{1b}/β_{1f} , degrees	$\Delta\beta_{1b}/\Delta\beta_{1f}$, degrees
RK 1	0.298	0.716	1.396	28.2 / 24.3	35.1 / 17.7	9.1 / 1.1
RK 2	0.298	0.690	1.397	28.5 / 23.5	35.5 / 22.6	6.0 / 3.7

The reduction of the suction eye diameter in the RK 2 impeller permitted to reduce its outer hub diameter at the leakage joints too and thereby to improve the volumetric stage efficiency.

Flow patterns in the channels between blades of the RK 1 and RK 2 impellers are shown in Figure 6 and Figure 7 for a circulation-free inlet flow condition.

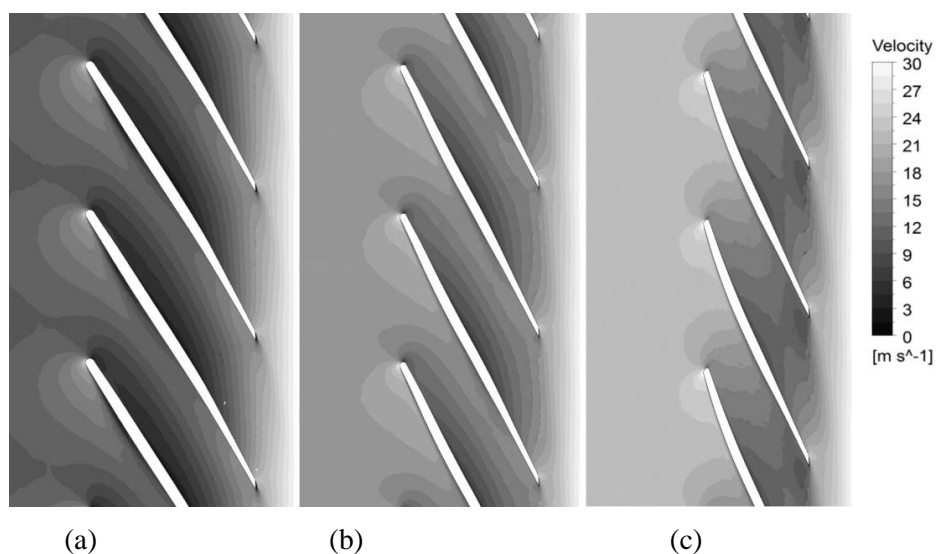


Figure 6. Flow Patterns in the Channels between Blades in Case of the RK 1 Impeller: (a) – near the hub; (b) – in the middle of the channel; (c) – near the front plate.

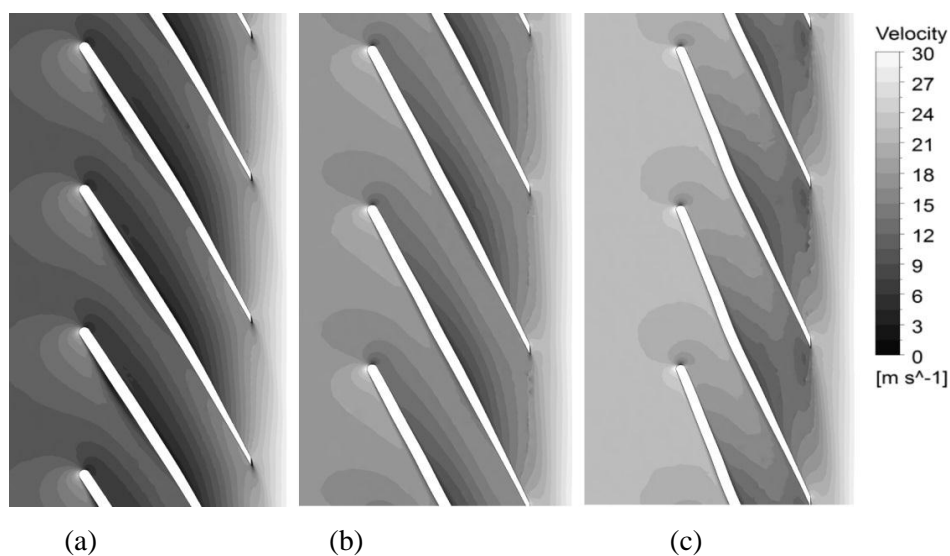


Figure 7. Flow Patterns in the Channels between Blades in Case of the RK 2 Impeller: (a) – near the hub; (b) – in the middle of the channel; (c) – near the front plate.

It should be noted that the reduction of angles of incidence in case of the RK 2 impeller resulted in a more uniform velocity distribution in the channels between blades.

Flow transfer passage between the first and subsequent pressure stages. As the first stage in the arrangement No. 2 is lowered down to the bottom of the can (caisson), between the first stage diffuser and impeller of the next stage there is a circular passage where, although the flow downstream of the diffuser being practically straightened, a hydraulic loss occurs. When selecting the best variant, we considered the following two constructions: the 1st one without edge at the diffuser outlet machined to a radius, for flow velocity of 5.1 m/s through the transfer passage and the 2nd one with edge at the diffuser outlet machined to a radius, for flow velocity of 4.0 m/s through the transfer passage.

Flow fields of velocity distribution in the transfer passage for both construction variants are represented for comparison in Figure 8. It shows that with the introduction of the diffuser outlet edge rounding-off (a kind of fairing) the extent and rate of flow separation at the transfer passage inlet, and, in addition, the enlargement of the free cross sectional area of the passage resulted in the reduction of losses therein due to the flow velocity decrease. Besides that, the flow pattern at the inlet of the intermediate stage impeller was improved by providing a convergent zone in the transfer passage upstream of the same.

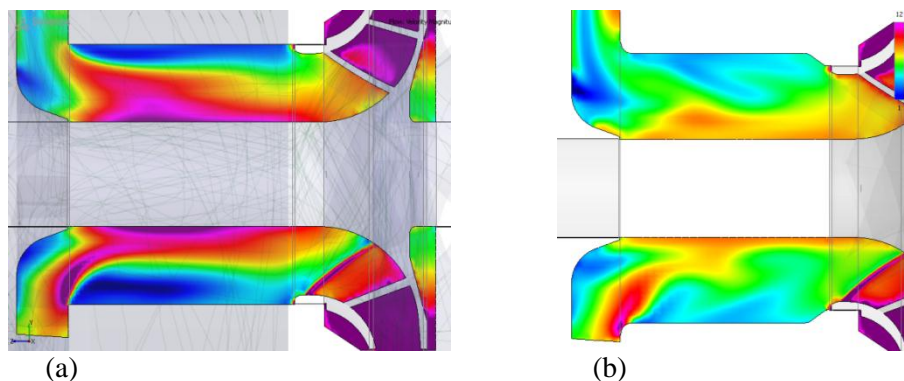


Figure 8. Flow Fields of Velocity Distribution in the Transfer Passage between the First Stage Diffuser and Impeller of the Next Stage: (a) – 1st variant of construction and b) – 2nd variant of construction.

According to the results of comparison of integral quantities losses calculated for the 2nd variant of construction are 0.8 m less than those in case of the 1st variant.

Investigation of hydraulic parts on the whole. In order to enable a relative assessment of the economical operation of the two suggested construction arrangements of hydraulic parts of the VS6 type pump, numerical computations were performed in PumpLinx for both systems. The numerical computations were conducted with the same grid parameters and boundary conditions. Figure 9 shows computation models of both construction arrangements as well as check cross-sectional areas, which were used for determining values of power input, head, hydraulic loss and volumetric leakage loss. The arrangement No. 1 hydraulic parts comprised the following: the P1 suction casing, the arrangement No. 1 first stage incorporating suction impeller and upstream inducer as well as the RK 1 variant of intermediate stage and last stage impellers. The arrangement No. 2 hydraulic parts comprised the following: the P2 suction casing, the arrangement No. 2 first stage incorporating suction impeller and upstream inducer along with the flow transfer passage between the first and subsequent pressure stages made according to the 2nd variant of construction as well as the RK 2 variant of intermediate stage and last stage impellers.

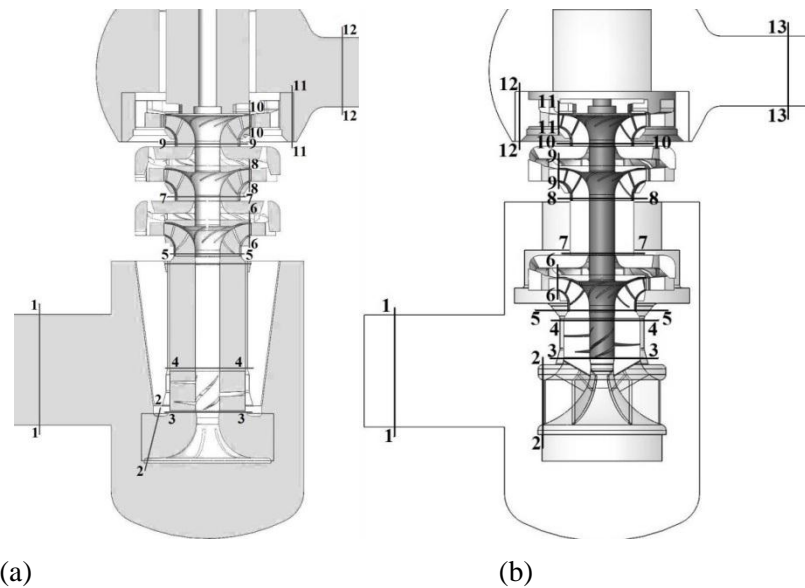


Figure 9. Locations of Check Cross Sectional Areas in the Pumps under Study: (a) – arrangement No. 1 and b) – arrangement No. 2.

It has been established that on the whole the hydraulic parts in the arrangement No. 2 is more economical than those made according to the arrangement No. 1.

Verification of the computing experiment results. In order to verify the results of computing experiment, the pumps (No. 1, an earlier serially produced pump with hydraulic parts in the arrangement No. 1, and No. 2, a new pump with hydraulic parts according to the arrangement No. 2) were subjected to comparison tests. As it is clear from the comparison of performance curves (refer to Figure 10), the maximum efficiency obtained with the pump No. 2 is 3.5 % higher than that in case of the pump No. 1.

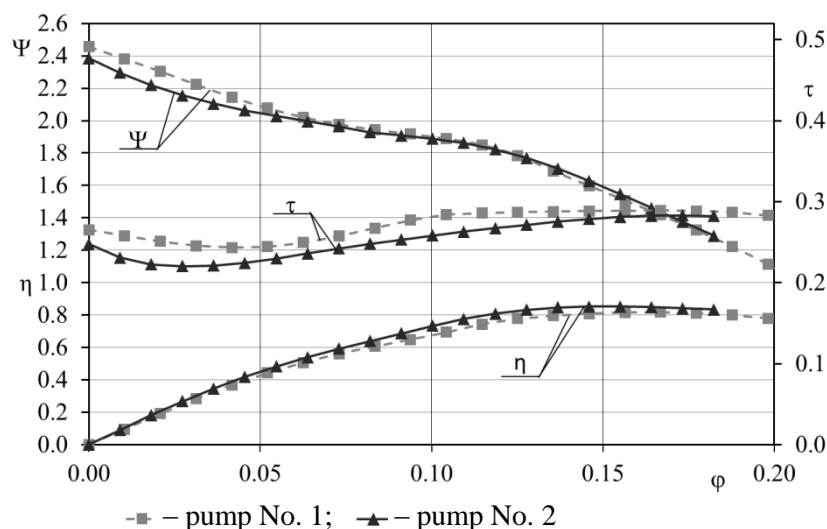


Figure 10. Comparison of Performance Curves of the Pumps with Different Construction Arrangements of Hydraulic Parts.

5. Conclusions

Two construction arrangements of hydraulic parts of the VS6 type pump have been compared. It was demonstrated that the arrangement No. 2 is preferable in terms of loss minimization. The decrease in losses has been obtained as a result of:

1. Application of the combination of the suction impeller and upstream inducer spaced apart on the minimum distance;
2. Application of an inducer without a cuneate ledge;
3. Application of intermediate stage and last stage impellers with an optimized hydraulic design;
4. Optimization of the geometry of stationary hydraulic parts both in terms of the decrease of flow velocity and in terms of obtaining a more uniform velocity distribution.

References

- [1] ANSI/API Std 610/ISO 13709, Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries.