Selection of design and determination of quality targets for VS6 pump stages

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Abstract. A comparative analysis of VS6 pump type in the range of specific speed \( n_q = 25...55 \) was carried out basing on information about analogous pumps produced by the world’s leading manufacturers. The focus was made on mass-dimensional characteristics of the pump stages. It was found that rational design of a diffuser with radial vanes provides lower material consumption at similar hydraulic characteristics compared to using a semi-axial vanes diffuser in the stage.

Centrifugal vertical semi-submersible pumps of a double casing design are widely used in the mainline oil transportation, processing of hydrocarbons as well as condensate pumps at power stations. The pumps of VS6 type with in-line suction and discharge nozzles are the most popular in the world practice [1,2].

Perfection of the pump hydraulics largely determines the pump competitiveness in general and first of all by the criteria of efficiency and minimum weight and size. Two types of stages are used in VS6 pump type: with radial diffuser (hereinafter – type 1) and semi-axial diffuser (hereinafter – type 2). The type 1 is traditionally preferred by European manufacturers, the type 2 – by the North American ones. These types of hydraulic have identical hydrodynamic parameters in the range of specific speed \( n_q = 25...55 \) [2]. Hence the choice between type 1 and type 2 for this specified speed range should be based on the possibility to get the minimum weight and size. Obviously, the stage type 1 has relatively smaller axial and larger radial dimensions in comparison with the stage type 2. Therefore, the comparison of the stages of both types should be carried out according to the mass of the stage, and also by the mass of the corresponding section of the outer pump casing.

In view of the foregoing, the development of methodological recommendations for the selection of optimal design for VS6 pump stages in the range of the specific speed \( n_q = 25...55 \) has considerable scientific and practical interest.

To solve this problem we collected, summarized, processed and analyzed the available statistical information on analogous pumps manufactured by the world’s leading manufacturers. As information sources there were catalogs [3,4], programs for selection of pumps with authorized access on the manufacturers websites [5,6], as well as information bases with open access [7,8].

At the first stage of the work the upper limit of the stage specific speed value \( n_{q,BEP} \) corresponding to the maximum energy efficiency was determined. Analysis of the tender specifications for the purchase of VS6 type pumps showed that presently almost all customers limit the pump head rise from
the design point to zero flow by value not above 20 ... 25%. In view of this the upper limit of the VS6 pump stage specific speed was defined as $n_{q,BEP} = 55$ (Figure 1).

![Figure 1. Dependence of head rise of stage type 2 on $n_{q,BEP}$.](image1)

When estimating the mass-dimension characteristics, the volume of the outer contour of a stage of type 1 was defined as a volume of a conditional cylinder with a diameter equal to the outer diameter of the diffuser $D_{5R}$ and a length equal to the length of the stage $L_{st,R}$ (Figure 2).

![Figure 2. To an estimate of the mass-dimensional characteristics of a stage of type 1.](image2)

Dependence (proportionality) of the mass of a stage of type 1 on the geometric characteristics of this cylinder was represented in a dimensionless form relative to the maximum diameter of the impeller $D_{2max}$:

$$m_R \sim \frac{\pi}{4} \times D_{5R}^2 \times L_{st,R} = \frac{\pi}{4} \times D_{4R}^2 \times L_{st,R}$$

(1)

In turn, dimensionless criteria $\overline{D_{5R}}$ and $\overline{L_{st,R}}$ were determined empirically from the analysis as the functions of $n_{q,BEP}$:

$$\overline{D_{5R}} = D_{5R}/D_{2max} = f_1\left(n_{q,BEP}\right), \quad \overline{L_{st,R}} = L_{st,R}/D_{2max} = f_2\left(n_{q,BEP}\right)$$

(2)

To find the relative diameter of the diffuser of type 1 averaging of the known dependence from the monograph [2] was used:

$$\overline{D_{5R}} = (1.05 \div 1.15) + n_{q,BEP}/100 \approx 1.1 + n_{q,BEP}/100$$

(3)
The statistical dependence of the relative length $L_{st,R}$ of the stage of type 1 on $n_q,BEP$, obtained with the help of the one of the pump selection programs, is shown in Figure 3.

![Figure 3](image1)

**Figure 3.** Dependence of the relative length $L_{st,R}$ of a stage of type 1 on $n_q,BEP$.

An analysis of the design of stages with semi-axial diffuser with respect to pumps of type VS6 showed that in considered range of values of $n_q,BEP$, the stages of most manufactures have an outer contour in the conditional form of a cylinder and its associated truncated cone (Figure 4).

![Figure 4](image2)

**Figure 4.** To an estimate of the mass-dimensional characteristics of a stage of type 2.

When estimating the mass-dimensional characteristics, the volume of the outer counter of the stage of type 2 was determined as the sum of the volume of the cylinder with a diameter equal to the outer diameter of the diffuser $D_{SS}$ and the length equal to the length of the cylindrical stage section $L_{st,S, cyl}$ and the volume of the truncated cone with a diameter of a large base equal to the outer diameter of the diffuser $D_{SS}$, diameter of a smaller base equal to the diameter of the impeller eye $D_{0S}$ and length equal to the length of the conical stage section $L_{st,S, con}$.

Dependence (proportionality) of the mass of a stage of type 2 on the geometric characteristics of its outer contour was represented in a dimensionless form relative to the maximum diameter of the impeller $D_{2max}$. 
The value of \( L_{\text{st,con}} \) is close to the relative axial length of the blade system of the semi-axial diffuser and was estimated from the known dependence from the monograph [2]:

\[
L_{\text{st,con}} \approx 0.72 \times \left( \frac{n_q \cdot \text{BEP}}{200} \right)^{0.19}
\]  

(5)

Taking this into account, the value of \( L_{\text{st,cyl}} \) was calculated as:

\[
L_{\text{st,cyl}} = L_{\text{st,S}} - 0.72 \times \left( \frac{n_q \cdot \text{BEP}}{200} \right)^{0.19}
\]  

(6)

Dimensionless criteria \( D_{5S} \), \( D_{0S} \) and \( L_{\text{st,S}} \) were determined empirically from the analysis as the functions of \( n_q \cdot \text{BEP} \):

\[
D_{5S} = \frac{D_{5S}}{D_{2 \text{max}}} = f_3 \left( n_q \cdot \text{BEP} \right), \quad D_{0S} = \frac{D_{0S}}{D_{2 \text{max}}} = f_4 \left( n_q \cdot \text{BEP} \right), \quad L_{\text{st,S}} = \frac{L_{\text{st,S}}}{D_{2 \text{max}}} = f_5 \left( n_q \cdot \text{BEP} \right)
\]  

(7)

The empirical dependences of the relative diffuser outer diameter \( D_{5S} \) on \( n_q \cdot \text{BEP} \) (Figure 5), the relative stage length \( L_{\text{st,S}} \) on \( n_q \cdot \text{BEP} \) (Figure 6) and the relative diameter of the impeller eye \( D_{0S} \) on \( n_q \cdot \text{BEP} \) (Figure 7) were obtained as a result of analysis and generalization of information about the stages of type 2 of analogous pumps produced by the world’s leading manufacturers.

**Figure 5.** Dependence of the relative diffuser outer diameter of a stage of type 2 on \( n_q \cdot \text{BEP} \).
Figure 6. Dependence of the relative length of the stage of type 2 on $n_q.BEP$.

$$L_{LS} = 0.204 \times n_q^{0.444}$$

Figure 7. Dependence of relative impeller eye diameter of the stage of type 2 on $n_q.BEP$.

$$D_{0S} = 0.081 \times n_q^{0.508}$$

On the basis of formulas 1 and 4, using the obtained empirical relationships, were estimated the ratio of the mass of a stage of type 1 to the mass of a stage of type 2 as a function of $n_q.BEP$. In this case, the value of criteria $L_{st,R}$, $D_{LS}$, $L_{st,S}$, and $D_{0S}$ were determined according to Figure 3, Figures 5 – 7. The values of criteria $D_{SR}$, $L_{st,S_con}$ and $L_{st,S_cyl}$ were calculated using formulas (3), (5) and (6).

As a result, it was established that in the whole range of the values of $n_q.BEP$ considered, the mass of a stage with the semi-axial diffuser will be large than the mass of a stage of the type 1 (Figure 8).
Figure 8. Dependences of the ratio of the mass of a stage of type 1 to the mass of a stage of type 2 on $n_{q, BEP}$.

It should be noted that this statement is valid if the stator elements of the stage of type 1 are designed rationally from the point of view filling its outer contour with metal (Figure 9, a). Otherwise, the mass of a stage of type 1 may exceed the mass of a stage of type 2 (Figure 9, b).

Figure 9. Examples of rational (a) and irrational (b) design of stator elements of a stage of type 1 from the point of view of metal consumption.

Recommendation of the Hydraulic Institute of the USA [9] on the velocity of liquid between the inner and outer casing of VS6 pump (no more than 1.5 m/s) allows to move from the maximum outer diameter of the diffuser to the diameter of the outer casing of the pump during the design process (Figure 10).

Figure 10. To an estimate of the mass of outer casing section of VS type pump with a stage of types 1 (a) and 2 (b) per one stage.
For a stage of type 1 the distance between the flange for fastening the pack of stages and the outer casing of the pump \( w \) determined as:

\[
\Delta_R = \left[ Q_{\text{BEP}} / (\pi \times v) + \left( D_{5R} / 2 + \delta_{1R} + \delta_R \right)^2 \right]^{0.5} - D_{5R} / 2 - \delta_{1R} - \delta_R
\]  

(8)

where \( \delta_{1R} \) is the wall thickness of the section of the stage of type 1;
\( \delta_R \) is width of flange for fastening the pack of stages of type 1.

Similarly for a stage of type 2:

\[
\Delta_S = \left[ Q_{\text{BEP}} / (\pi \times v) + \left( D_{5S} / 2 + \delta_{1S} \right)^2 \right]^{0.5} - D_{5S} / 2 - \delta_{1S}
\]  

(9)

Dependences of the volume of the section of the outer pump casing per one stage of types 1 and 2 were represented as:

\[
V_R = \pi \times L_{st.R} \times \left[ \left( D_{5R} / 2 + \delta_{1R} + \delta_R + \Delta_R + \delta_{2R} \right)^2 - \left( D_{5R} / 2 + \delta_{1R} + \delta_R \right)^2 \right]
\]  

(10)

\[
V_S = \pi \times L_{st.S} \times \left[ \left( D_{5S} / 2 + \delta_{1S} + \Delta_S + \delta_{2S} \right)^2 - \left( D_{5S} / 2 + \delta_{1S} \right)^2 \right]
\]  

(11)

where \( \delta_{1R}, \delta_{1S} \) are the wall thickness of the outer pump casing with stages of types 1 and 2, correspondingly.

Using formula (3), as well as Figures 3, 5 and 6, a series of preliminary evaluation construction drawings of pumps of type VS6 with rotational speed \( n = 1500 \text{ rpm} \) were made (Table 1).

| \( \text{BEP} \) \( Q_{\text{BEP}} \) \( m^3/h \) | \( D_{2\text{max}} \) \( mm \) | \( \delta_{1R} \) \( mm \) | \( \delta_{2R} \) \( mm \) | \( \Delta_R \) \( mm \) | \( \delta_{1S} \) \( mm \) | \( \Delta_S \) \( mm \) | \( V_R \times 10^3 \) \( mm^3 \) | \( V_S \times 10^3 \) \( mm^3 \) | \( V_R/V_S \) |
|-----------------|----------|-----|-----|-----|-----|-----|-------|-------|-----|
| 200             | 31.5     | 290 | 12  | 12  | 50  | 21  | 29    | 2324  | 4603 | 0.50 |
| 315             | 34.2     | 320 | 16  | 16  | 60  | 29  | 40    | 4157  | 8170 | 0.51 |
| 500             | 38.4     | 350 | 20  | 20  | 60  | 41  | 56    | 6825  | 13635| 0.50 |
| 800             | 43.8     | 385 | 22  | 22  | 70  | 57  | 79    | 10243 | 20271| 0.51 |
| 1250            | 43.8     | 450 | 24  | 24  | 100 | 73  | 105   | 15863 | 30822| 0.51 |
| 1800            | 52.1     | 465 | 26  | 26  | 100 | 97  | 140   | 21253 | 41524| 0.51 |

As can be seen from Table 1, the volume and, correspondingly, the mass of the section of the outer pump casing per one stage for stages of type 2 is much larger than in case of stages of type 1.

The obtained results were used by the HMS Group during the development of its own new product line of the NMV pumps type VS6 with in-line suction and discharge nozzles during the formation of the range chart for the entire pump series and also for the creation and assessment of the actual competitiveness of the hydraulics of this pump type. The use of data of the world's leading manufacturers as targets for the quality parameters of the pump hydraulics predetermined the high competitiveness of the NMV pumps for both energy efficiency and mass-dimension characteristics. The last is clearly demonstrated in Figure 11.
Figure 11. Pumps of type VS6 with in-line suction and discharge nozzles produced by the HMS Group: (a) the first generation (pump KsV 1000-220); (b) the new generation (pump type NMV 500-350).

Conclusions
1 To use of hydraulics with a specific speed \( n_{q,BEP} < 55 \) ensures the demand for pumps of type VS6 in practice by the criterion of the limit head rise.

2 With rational design the use of stages with radial diffusers in pumps of VS6 type ensures a lower pump material consumption with similar hydraulic parameters compared to the hydraulics based on semi-axial diffusers. In each case the final choice of the stage type should be performed taking into account the comparative evaluation of the prime cost of machining its stator elements.

3 Effective use of the results of the available information analysis on analogous pumps produced by the world’s leading manufacturers at the design stage of the pump hydraulics ensures a high level of competitiveness of the new pumping equipment.

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