RESEARCH OUTCOMES ON REPLACEABLE FLOW PARTS IN DOUBLE SUCTION PUMPS

Elena Knyazeva*, Andrew Rudenko*, Igor Tverdokhleb*

One of vital and challenging tasks is to improve double suction pump performance in conditions when essential changes of their parameters are required without playing with their dimensions in case of adapting their capacity to seasonality or flow-rate change for a long time interval or for pipelines at various stage of their development.

In this paper a few design options of flow parts executed for the same casing have been examined:

– Nominal mode: Semi-volute suction chamber, double suction impeller and double volute discharge chamber;

– Suboptimal mode: replaceable impellers and diffusers are used (same suction and discharge chambers)

Numerical analysis results of the flow parts are presented in this article. Double suction pumps with projected diffuser for replaceable rotor with 0.6 Qn flow-rate and with perspective diffusers engineered within ANSYS CFX-12.0.

The latter (perspective diffusers) have been analyzed to find ways to reduce hydraulic losses.

Q-H, power consumption and performance curves are also given. Numerical experiment is an effective method for predicting patterns of characteristic curves even for such sophisticated flow parts as double suction pumps with a combined discharge chamber.

Keywords: *oil-trunk pump, impeller, diffuser, volute, numerical analysis*

1. Introduction

At present the use of replaceable flow parts is quite common to extend the operating flow-rate range while maintaining a high level of efficiency at required head.

This paper covers how replaceable flow parts are engineered on the example of oil-trunk pumps designed to operate in booster stations at the initial pipeline development stage, when flow-rates would typically only be a fraction of the projected capacity rate.

Single-stage double suction BB1 pump has been selected, with semi- volute suction and twin-volute discharge chambers.

Diffuser has been installed to ensure flow consistency between replaceable impeller and same twin-volute discharge and semi-volute suction chambers.

One of the main criteria was that diffuser should be designed for hydraulic loss minimization. Therefore initially diffuser no. 1 (Fig. 1) has been developed with dimensions and parameters selected according to existing guidelines [3] assuming that impellers would give the required head.

* E. Knyazeva, A. Rudenko, I. Tverdokhleb, HMS Group, Russia

However, diffuser no. 1 is characterized with increased losses due to sudden flow expansion at the diffuser output, because optimal expansion angle and length ratio does not allow smooth flowing from diffuser into twin-volute discharge chamber.

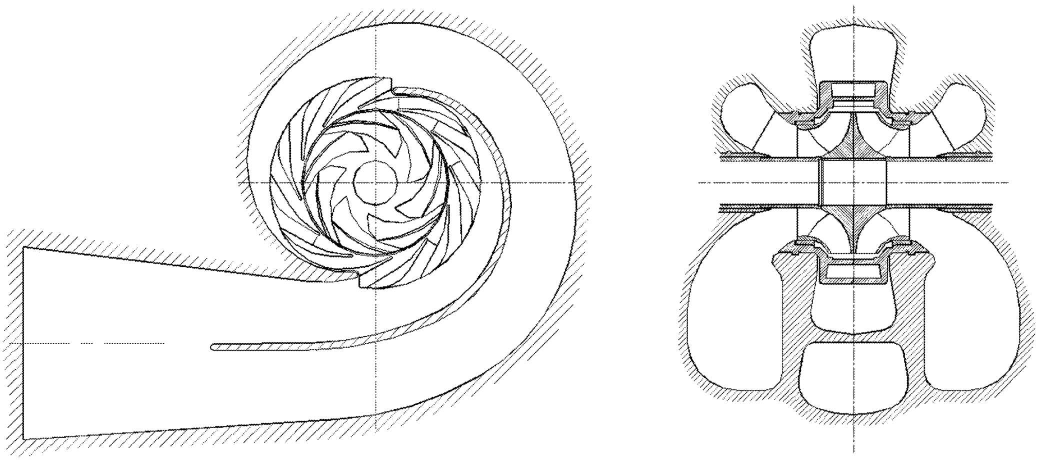

Fig.1: Cross-section of diffuser no. 1

To lower losses at the diffuser output diffuser no. 2 has been developed with intermediate disk. Channels have not been changed (the same as in diffuser no. 1), but in the axial direction the output width has been increased to provide the optimal equivalent opening angle of diffuser channel and for each half of diffuser (Fig. 2)

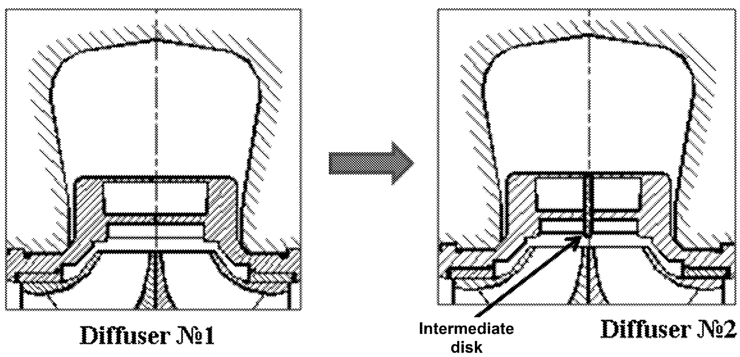

Fig.2: Cross-section of diffuser no. 2 with intermediate disk

Diffuser no. 3 has also been developed (Fig. 3). Its design was supposed to provide flow consistency from impeller with twin-volute chamber dimensions. It has been predicted that losses in that kind of flow parts would be minimal and its manufacturing would be much simpler than traditional diffusers. Its design has been developed on the base of analysing cross-section of twin-volute chamber, cutwater location and distance from trailing edges to cutwaters.

Diffuser no. 3 is made in the form of two spiral channels with input dimensions calculated for rated flow-rate and head of replaceable rotor.

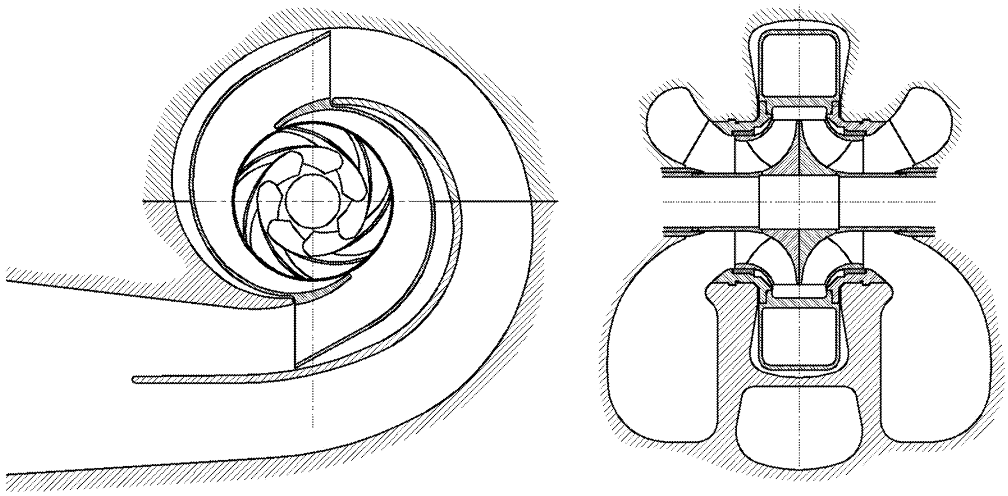

Fig.3: Cross-section of 2-channel diffuser no. 3

For calculating predicted energy efficiency with flow part designs described above ANSYS CFX has been used. Calculated model for carrying out numerical simulation is shown on Fig.4 including the following flow parts.

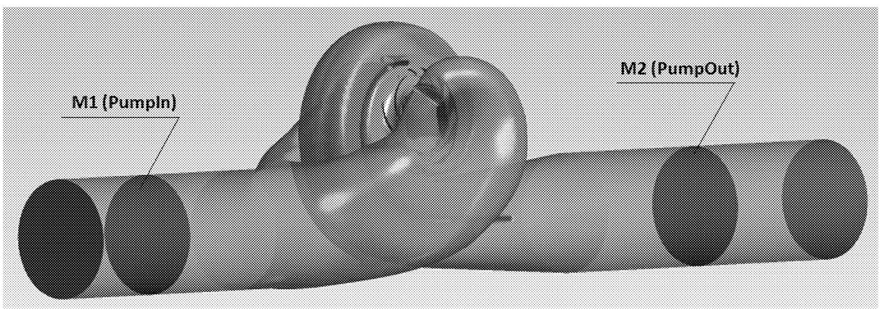

Fig.4: Computational model of flow parts

Numerical grids of flow part elements have been received in ICEM CFD. Total number of nodes for a model amounted 10.5 million in average. Numerical simulation has been carried out in the transient type for the three modes.

The k-ε turbulence model has been applied.

As results of numerical simulation we have obtained functions of head $H = H(Q)$, hydraulic power $N = N(Q)$, hydraulic efficiency $\eta = \eta(Q)$. Mechanical and leakage losses have been calculated on the basis of semi-empirical functions.

Predicted parameters of all three flow part versions and parameters derived from physical tests are shown on figs. 5, 6, 7 in dimensionless form. Next dimensionless factors were used:

$$K_{\rm H} = \frac{g\,H}{n^2\,D^2}\;, \qquad K_{\rm Q} = \frac{Q}{n\,D^3}\;, \qquad K_{\rm N} = \frac{N}{\varrho\,n^3\,D^5}\;,$$

where H – head (m), Q – flow rate (m^3/s), D – outlet diameter of impeller (m), n – rotational speed (revolutions per second), n – density of water (kg/m^3), g – acceleration due to gravity ($g = 9.81\,$m/s^2).

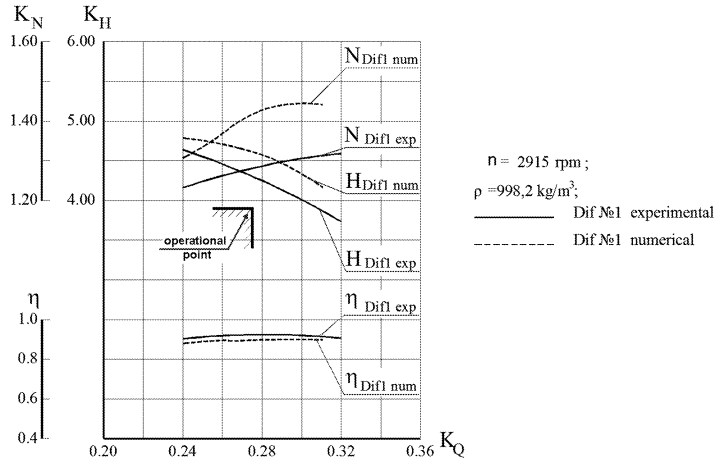

Fig.5: Predicted parameters and physical test with diffuser no. 1

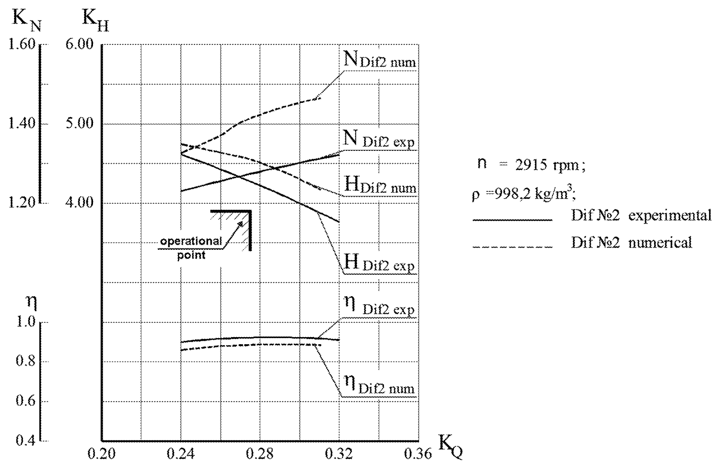

Fig.6: Predicted parameters and physical test with diffuser no. 2

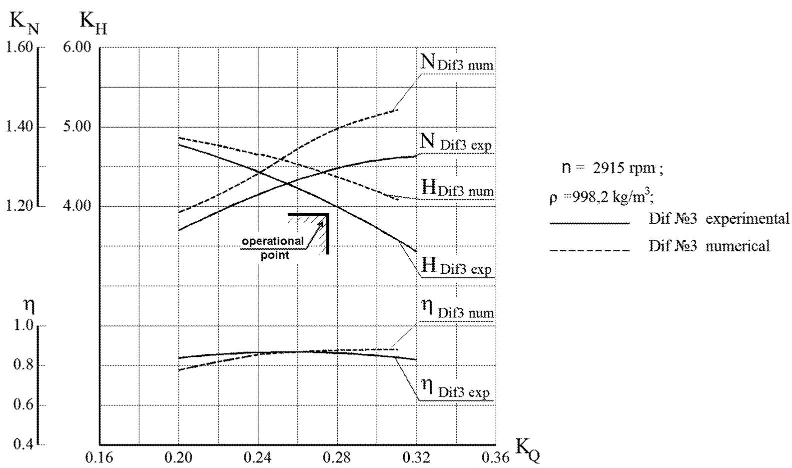

Fig.7: Predicted parameters and physical test with diffuser no. 3

As shown in Fig. 5, 6, 7 flow part with diffuser no. 3 has the lowest efficiency in terms of head and performance.

Comparison of flow patterns has shown that the largest irregularity of velocities and vortex formation is observed in the case of diffuser no. 3 (Fig. 8).

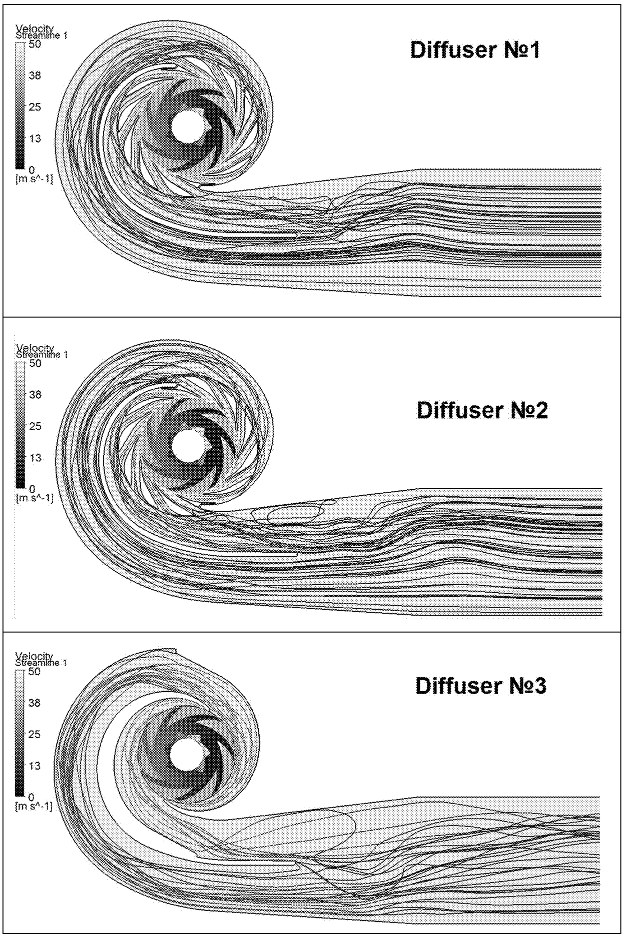

Fig.8: Flow stream visualization based on the results of numerical simulation in the rated mode

Flow analysis in diffuser no. 1 and no. 2 has shown that implementation of intermediate disc enabled to receive more even flow at the channel input (as clearly seen in the figures, that demonstrate velocity areas in the twin-volute cross-section).

Decrease of high velocity areas to lower losses from sudden flow expansion has been observed at the channel output.

However, the integral values of head and pump power with diffuser no. 2 remained at the same level as with diffuser no. 1.

Considering that manufacturing of diffuser no. 1 is less time-consuming it has been concluded that this version is the best option to choose for operation.

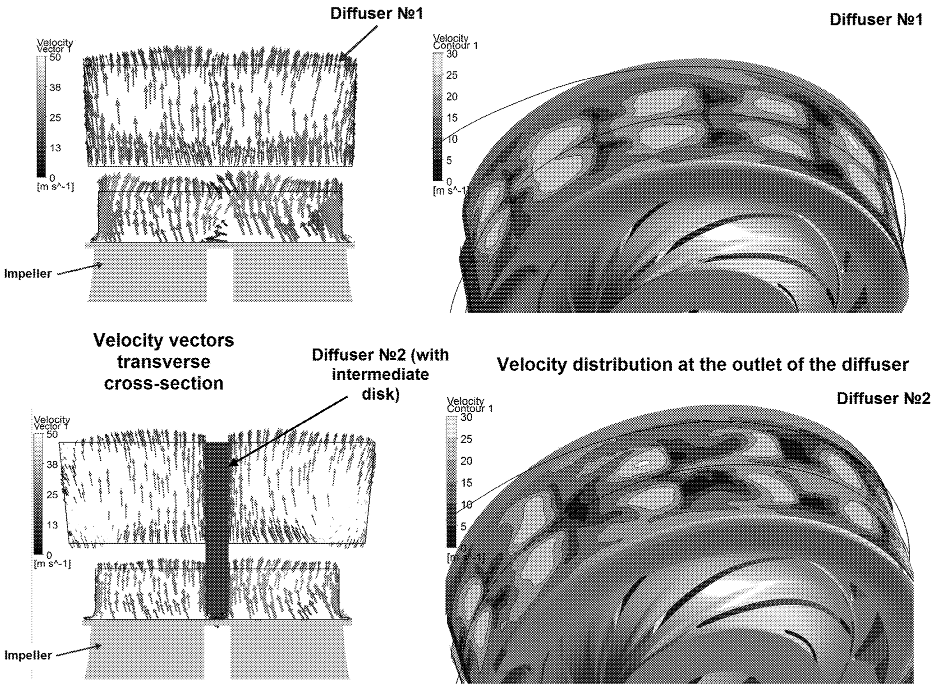

Fig.9: Flow analysis in diffusers no. 1 and no. 2 based on numerical simulation results

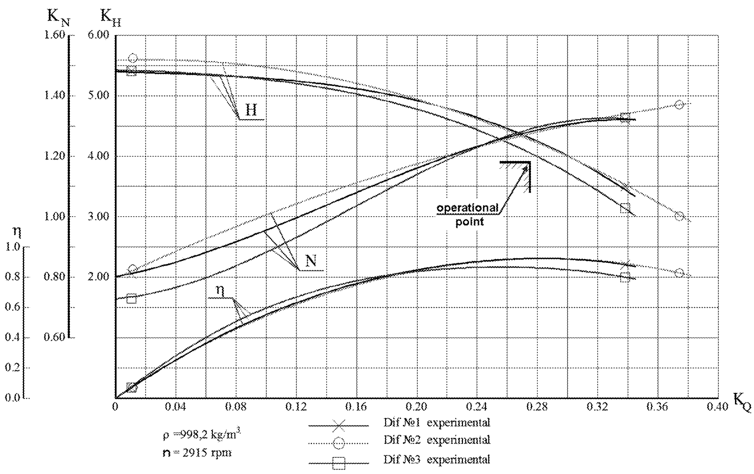

Fig.10: Characteristic curves of pump

All developed diffusers have also been tested (as part of a pump) on the stand. Pump parameters are shown in Fig. 10.

Thus, comparison of numerical simulation with physical tests showed qualitatively good match of head and performance at the optimal flow-rate.

At the same time use of diffuser no. 1 is the most effective and no. 3 provides the maximum losses, while deviation of rated and experimental heads being grown with capacity increase.

Numerical simulation of other flow parts have been planned to carry on as well as data gathering and search for corrections to improve parameter prediction.

2. Conclusion

1. As the outcome of this research introduction of the intermediate disk does not necessarily reduce the irregularity flow at the outlet of diffuser no. 2 and special spiral channels (diffuser no. 3) would make it worse by increasing hydraulic losses. Therefore it has been decided to use diffuser no. 1 for manufacturing as the most efficient option.
2. Transient simulation (for simulating the mutual influence of impellers & diffusers) should be applied for precise prediction of main parameters (Q-H, performance) for sequentially arranged impellers and diffusers
3. Numerical experiment is quite effective for predicting patterns of characteristic curves even for reasonably sophisticated flow parts as those of double suction pumps with a combined discharge chamber.

References

[1] Tverdohleb I., Vizenkov G.: The right choice, New oil-trunk , Pumps & Equipment, No. 4(45) – 5(46), pp. 28–31, 2007
[2] Tverdohleb I., Vizenkov G., Biryukov A.: Oil pipeline from Siberia to the sea, WORLD PUMPS, May 2012
[3] Ivanushin A., Kolesnik E.: Centrifugal pump with a twin-volute discharge chamber, Patent No. 103149, registered in the Russian State Patent Register 27.03
[4] Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries, ANSI/API Standard 610

Received in editor's office: November 6, 2012
Approved for publishing: March 10, 2013