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Experimental and Numerical Analysis of a Non-Newtonian Fluids Processing Pump

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Abstract

Centrifugal pumps are used in many applications in which non-Newtonian fluids are involved: food processing industry, pharmaceutical and oil/gas applications. In addition to pressure and temperature, the viscosity of a non-Newtonian fluid depends on the shear rate and usually is several orders of magnitude higher than water. High values of viscosity cause a derating of pump performance with respect to water. Nowadays, pumping and mixing non-Newtonian fluids is a matter of increasing interest, but there is still lack of a detailed analysis of the fluid-dynamic phenomena occurring within these machines. A specific design process should take into account these effects in order to define the proper pump geometry, able to operate with non-Newtonian fluids with specific characteristics. Only few approaches are available for correcting the pump performance based on the Hydraulic Institute method.

In this work, an experimental and numerical campaign is presented for a semi–open impeller centrifugal pump elaborating non-Newtonian fluids. An on-purpose test bench was built and used to investigate the influence on pump performance of three different non-Newtonian fluids. Each pump performance test was accompanied by the rheological characterization of the fluid, in order to detect modifications of the rheological phenomena and allow a proper Computation Fluid Dynamics (CFD) modeling. The performance of the machine handling both Newtonian and non-Newtonian fluids are highlighted in relation with the internal flow field.

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Keywords: centrifugal pump, semi-open impeller, non-Newtonian fluids, pseudoplastic, CFD

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Introduction

Non-Newtonian fluids are characterized by a non-linear relation between shear stress and shear rate and, for time independent non-Newtonian fluids the apparent viscosity, defined as the ratio between the shear stress and shear rate values, depends on the local shear rate value [1]. For this reason, Non-Newtonian fluids are able to exhibit values of apparent viscosity higher than those of water viscosity. Therefore, when centrifugal pumps are used with non-Newtonian fluids their efficiency, head and shaft power are altered. The knowledge of non-Newtonian fluid effects on the pump performance is fundamental in the design process as well as in the pump choice (manufacturers provide pump performance curve obtained with water as operating fluid). In literature, there is a lack of experimental data about centrifugal pumps handling non-Newtonian fluids, and even less about numerical simulations of centrifugal pumps which process non-Newtonian fluids. Experimental tests of centrifugal pumps handling non-Newtonian fluids are reported for example in [2 – 9].

Some authors have proposed methods for estimating the performance of centrifugal pumps with non-Newtonian fluids [2, 3, 7]. These methods asses a representative value of viscosity of non-Newtonian fluids to be used in the Hydraulic Institute Method, which is specific for predicting the performance of pumps handling Newtonian fluids having a viscosity higher than water. Recently, CFD has been used to evaluate the performance of centrifugal pumps with non-Newtonian fluids but the number of applications is limited. In [10] an analysis of the performance and flow structures of two open impeller centrifugal pumps, operating with tomato paste (non-Newtonian), water and many Newtonian fluids is carried out. Other two relevant works are reported in [11], in which the internal flow of an open impeller pump is analyzed by using different volutes, and in [12] which presents a study on the laminar/turbulent transition inside an open impeller pump. In [13] the flow characteristics within a shrouded centrifugal impeller were investigated highlighting the different phenomena occurring with water and with non-Newtonian fluids. In [14] a comparison between the performance of two pumps operating with non-Newtonian fluids was obtained by means of CFD simulations and by applying the estimation methods reported in literature. In this case, the performance of the pumps with non-Newtonian fluids, calculated by numerical simulations, are not compared with experimental data.

In this paper, the results of experimental tests, conducted with non-Newtonian fluids on a small centrifugal pump [14, 15], will be presented to highlight the differences in performance with respect to the data obtained with water. The rheological behavior of non-Newtonian fluids was experimentally analysed by means of a rotational rheometer. Subsequently, CFD simulations were performed with the same fluids used throughout the experiments, with the aim of evaluating the differences between experimental and numerical results.

Pump description

The experimental tests have been carried out on an electric motor driven centrifugal pump of small size characterized by specific speed n_{sQ} equal to 18 $\text{rpm}\cdot\text{m}^{3/4}/\text{s}^{1/2}$, and designed to work at 2900 rpm. The electric motor has a nominal power of 0.37 kW. The pump is characterized by a semi-open type impeller with a diameter of 95.5 mm with seven blades. The pump was designed for operating with dirty water (industry or agriculture applications). The maximum diameter of the solid bodies is equal to 4 mm. Figure 1 shows the pump volute and impeller.

Experimental test rig

The experimental test rig allows an accurate measurement of the mass flow processed by the pump through to

Nomenclature

H	Head [m]	k	Consistency index [$\text{Pa}\cdot\text{s}^n$]
η	Efficiency [%]	n	Viscosity index [ ]
Q	Volumetric flow rate [l/min]	$\dot{\gamma}$	Shear rate [s^{-1}]
n_{sQ}	Kinematic specific speed [$\text{rpm}\cdot\text{m}^{3/4}/\text{s}^{1/2}$]	ϑ	Tangential coordinate [°]
ρ	Density [kg/m^3]	Vu	Tangential component of absolute velocity [m/s]
T	Temperature [°C]		

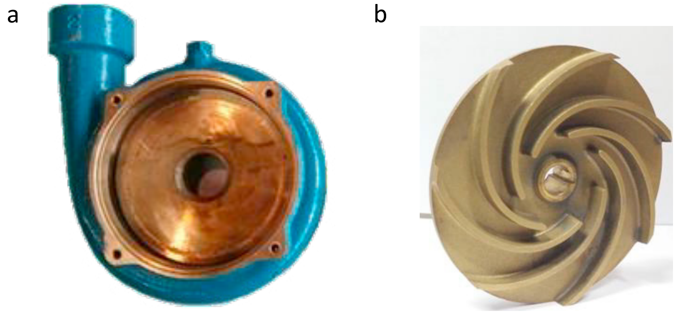

Fig. 1. Semi-open impeller pump: (a) volute; (b) impeller

the weighing method. The piping system is equipped with a switch valve which allows the use of two circuits: a closed-loop circuit and a separated circuit that provides the pumped fluid to a tank positioned above the a weighing system. The flow rate is regulated by means of a throttle device. The weighing method, described in detail within the standard UNI EN 24185, is the most accurate method for measuring the mass flow rate.

The experimental system is equipped with absolute pressure gauges, having a 0.75 % total uncertainty (total error band), positioned at the pump suction and discharge sections to measure the pump head. The rotational speed of the pump is set and controlled by means of an inverter which is connected to the electric motor. In order to accurately determine the mechanical power supplied to the pump shaft, the test bench was fitted with a torquemeter which is placed on the shaft connecting the electric motor and the pump. The accuracy of the torquemeter is 0.02 % FS (full scale percentage); whereas the maximum measured torque is 20 Nm. Figure 2 depicts the test rig equipped with electric motor, torquemeter, pump, valves and pressure transducers. The pressure and torque data are acquired using a NI DAQ modular data acquisition device driven by a custom application developed in NI Labview® environment.

Rheological data of non-Newtonian fluids were obtained with the rotational rheometer Advanced Rheometric Expansion System® (ARES, Rheometric Scientific) equipped with 25 mm titanium parallel plates. The tests were accomplished under continuous flow conditions and flow curves were obtained by changing the shear rate from 0 s^{-1} up to 400 s^{-1}. The fluid temperature was monitored throughout the test.

Test method

The procedure for determining the characteristic curves of non-Newtonian fluids consists in:

- the adjustment of the kaolin powder in the water tank in order to reach the desired concentration. The pump works as a mixing device forcing the fluid to circulate within the piping system until the desired level of homogenization is reached;
- the measurement of temperature, density and viscosity on a fluid sample. The rheological tests were performed

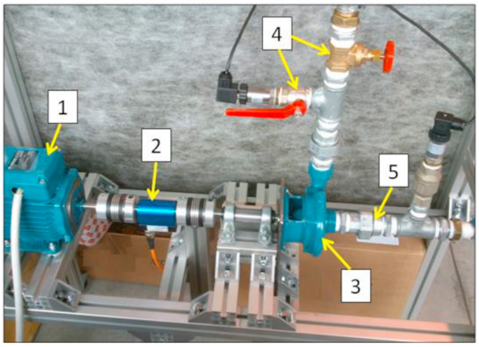

Fig. 2. Centrifugal pump test rig: 1-electric motor, 2-torquemeter, 3-pump, 4-5 pressure taps and discharge throttle device

up to the maximum shear rate applicable before sample instability;

- the determination of the characteristic curve of the pump at constant rotational speed. The pressure (suction and discharge), torque, fluid temperature and collected fluid mass measurements are simultaneous;
- The additional rheological characterization of the fluid at the end of the performance tests with the same procedure described at the second point. Thus a rheological characterization was obtained before and after the determination of each constant speed curve.

Through this procedure it was possible to check in detail the effect of temperature and non-Newtonian fluid rheological history on the performance of the centrifugal pump.

Fluids tested

Non-Newtonian fluids were obtained by mixing kaolin powder with water in different weight fractions: 30 %, 35 % and 40 %. As mentioned, the mixing procedure was performed by the pump itself. Table 1 shows the measured density value for the three different concentrations of kaolin after the test at 2000 rpm. The rheological results show the classic pseudoplastic fluid behaviour. Moreover, as reported in Fig. 3, increasing the kaolin powder concentration, the apparent viscosity value increases as well. Bands of variability of the apparent viscosity of 15 %, 8 % and 7 % for the Kaolin concentrations of 30 %, 35 % and 40 % respectively, have been observed. These variations are probably due to the variability inherent in the measurement of the fluid rheology.

Experimental pump performance

Figures 4 shows the pump performance curves experimentally obtained with water and non-Newtonian fluids at three different rotational speeds 2000 rpm, 2500 rpm and 2900 rpm. At high flow rates, with kaolin 30 % at all rotation speed, and with kaolin 35 % at 2900 rpm and 2500 rpm, the pump head increases with respect to water. This effect is also recognize looking at the efficiency trends. This phenomenon, called *sudden rising head*, has also been observed by [16] with Newtonian fluid with slightly higher viscosity than water. For the 35 % test at 2000 rpm and for the 40 % test at all pump rotation speeds, there is a decrease of pump performance with respect to water. Furthermore, with kaolin 35 % and specially with 40 % at low flow rates a significant decrease of head is present with compared to water; this decrement changes the concavity of the curve. This phenomenon with non-Newtonian fluids was observed also by [2, 3].

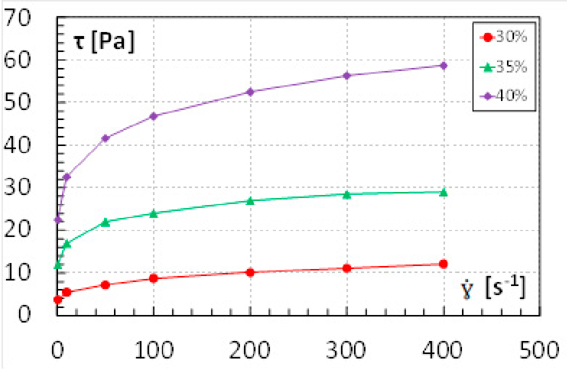

Fig. 3. Flow curves for the non-Newtonian fluids after the test at 2000 rpm

Table 1. Density and temperature of fluids after the test at 2000 rpm

Parameter	Kaolin 30%	Kaolin 35%	Kaolin 40%
ρ [kg/m^3]	1190.4	1241.7	1326.0
T [°C]	25.0	22.8	23.2

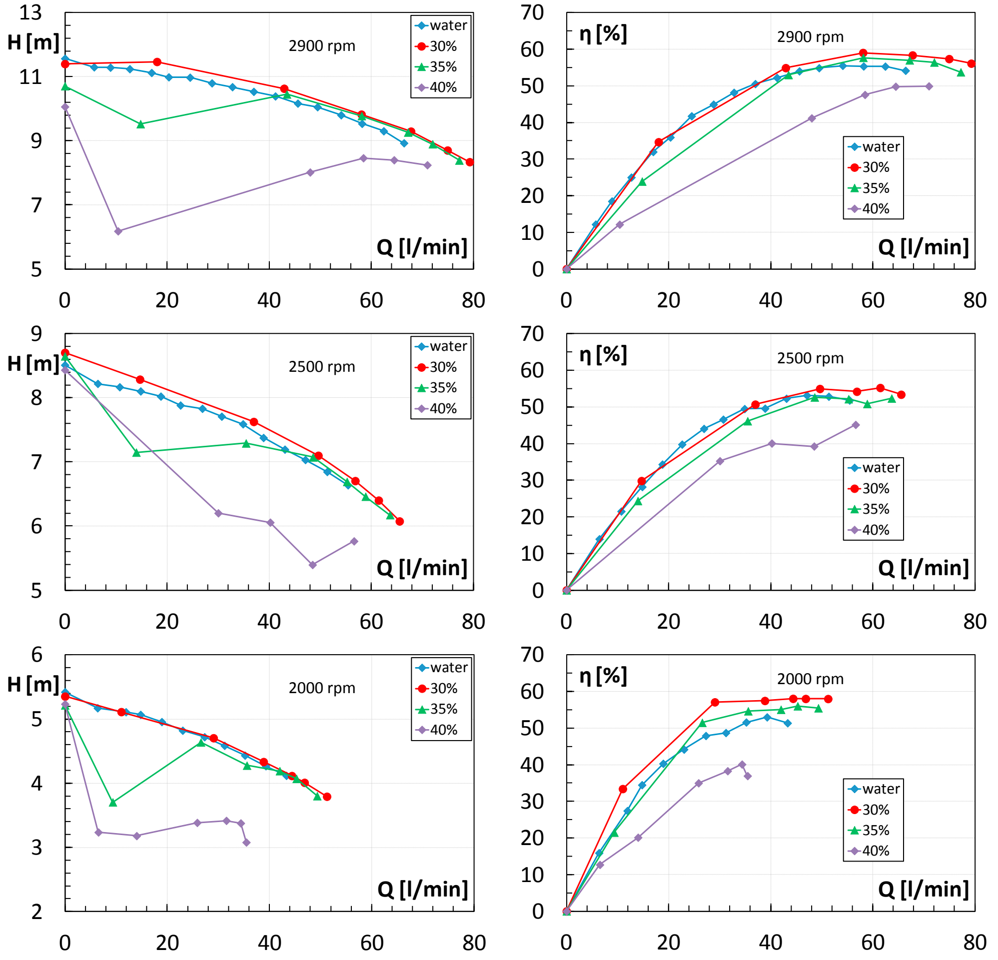

Fig. 4. Experimental head and efficiency with water and the three concentrations of kaolin at different rotation speed

Numerical model

For the numerical simulation of the pump, the pump geometry was reproduced in details. The size of the impeller has been measured directly considering its small size and the simple shape of the blades. The internal surfaces of the volute were obtained by measuring with a three-dimensional modeler the mold of the volute made with silicone rubber. The solid model was generated according to the procedure reported in [10]. The numerical simulations were carried out by means of the commercial CFD code ANSYS CFX 15.0. The grid used in the calculations is a hybrid grid of 11 million elements composed of tetrahedral elements on the core and prismatic elements on walls and it is depicted in Fig. 5. A second-order high-resolution advection scheme was adopted to calculate the advection terms in the discrete finite volume equations. The simulations were performed in a steady multiple frame of reference, taking into account the contemporary presence of moving and stationary domains. In particular, a mixing plane approach was imposed at the rotor/stator interface between the impeller and the volute and between the impeller and the inlet duct. A rotating frame of reference approach was used for the impeller domain with a rotation speed of 2900 rpm, 2500 rpm and 2000 rpm. As the inlet boundary condition, a constant velocity value with normal direction and a turbulent intensity equal to 5 % was imposed. The no-slip wall boundary condition was used for all the solid surfaces. At the outlet, a static pressure condition was used. As turbulence model, standard k-ω is chosen. This model has shown successful applications in the simulation of centrifugal pump as reported, for instance in [17, 18]. In order to model the non-Newtonian fluids behavior, the rheology values

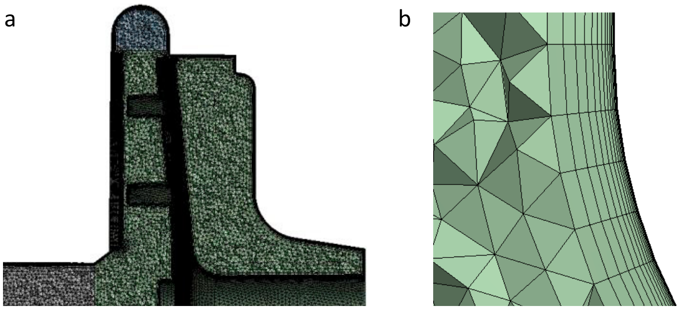

Fig. 5. Grid used: (a) section view; (b) detail of prismatic layers placed on the wall

measured at the beginning of each performance test are imposed. Because of pseudoplastic fluids, rheological behavior has been described mathematically through power law (Ostwald de Waele model) reported as follow

$$\tau = k\dot{\gamma}^{n} \qquad (1)$$

The parameters k and n of power law were obtained by interpolating the experimental data in the shear rate range tested with the rheometer (1 s^{-1} – 400 s^{-1}). Within the CFD model the power law was considered valid in the range of shear rates between 10^{-3} s^{-1} and 10^{6} s^{-1}, while outside this range the apparent viscosity was assumed constant. All fluids were considered isothermal with a constant density value equal to that measured for each concentration.

Numerical pump performance

Figure 6 reports the comparison between experimental and CFD head with water and the three concentrations of kaolin tested at 2900 rpm, 2500 rpm and 2000 rpm. With water, kaolin 30 % and kaolin 35 % at all rotational speeds, the deviation between experimental and numerical values is very low (maximum error bar of 6 %). Conversely, with kaolin 40 % this difference is higher (maximum error bar of 33 % at 2000 rpm) and the numerical model overestimates experimental head. This deviation seem to be due to the underestimation of the pressure losses by the CFD model (position of the pressure taps/probe, effects of duct and fitting roughness, etc.).

Figure 7 shows the shear rate and apparent viscosity fields according to a blade-to-blade plane with kaolin 35 %

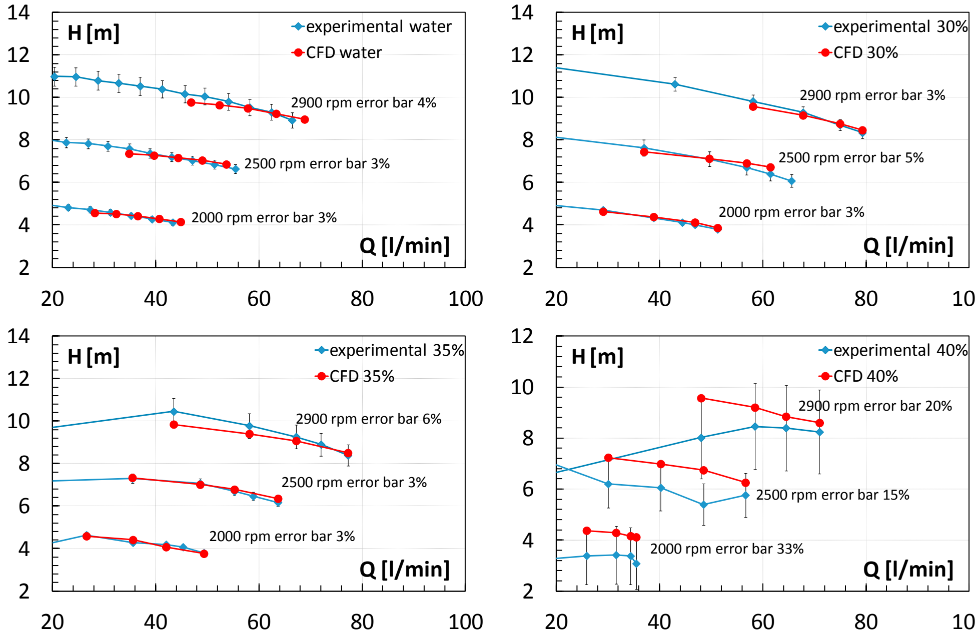

Fig. 6. Comparison between experimental and CFD pump head with water and the three concentrations of kaolin

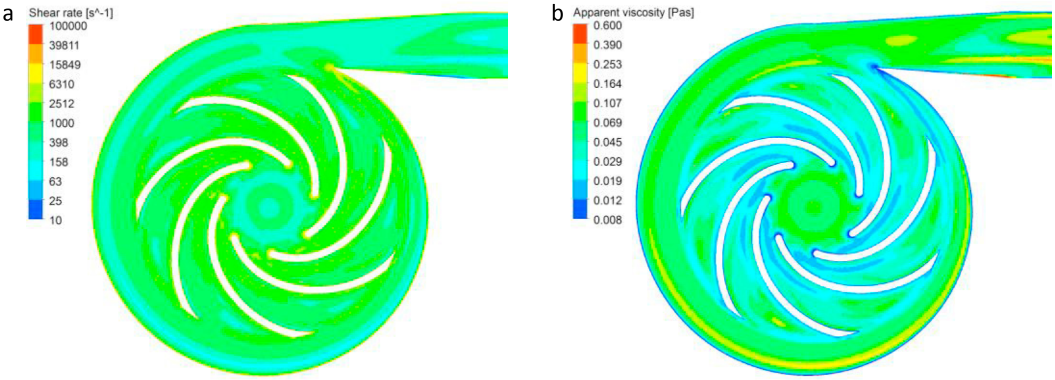

Fig. 7. Kaolin 35%: (a) shear rate distribution; (b) apparent viscosity distribution

at 2900 rpm and a volume flow rate of 58 l/min. Within the impeller the shear rates reaches higher values than in the volute, and as a consequence of the pseudoplastic fluid behaviour, the apparent viscosity is lower, in particular the lowest values are shown close to the pump walls.

Figure 8 reports the variation of the tangential component of absolute speed according the tangential coordinate at the exit of the impeller with water and kaolin 30 % at 2900 rpm at 58 l/min. The report shows a higher tangential component, both close to the duct axis and to the trailing edge, for the kaolin 30 % respect to the water which is responsible for the phenomenon of sudden rising head.

In Fig. 9 a comparison between the tip leakage flow at 2900 rpm at 58 l/min between the water and all the non-Newtonian fluids can be observed. From this comparison, it can be seen how, by changing the fluid, the end streams change, resulting in different velocity fields within the internal blade passages.

Conclusions

In this paper a detailed analysis of the performance of a centrifugal pump with water and non-Newtonian fluids was performed. Experiments revealed that at high flow rates, with the two non-Newtonian fluids with lowest kaolin powder concentrations (and then lower apparent viscosity values), there was a slight increase in head with respect to water, while with kaolin 40 % (that has the higher apparent viscosity), there is a high derating of pump performance compared to water curves. With kaolin 35 % and 40 %, at lower flow rate the pump head decreases, with a significant modification of the pump performance trend.

Subsequently, the pump performance was calculated by means of numerical simulations and compared with the experimental data. The numerical results well agree with experimental data for water, kaolin 30 % and 35 %, but they show an overestimated head with kaolin 40 %. The deviations between experimental and numerical results are acceptable if considering the complexity of the non-Newtonian fluid behaviour and the particular conditions of the experimental tests. Numerical simulations are also a useful tool to investigate the flow characteristics within the

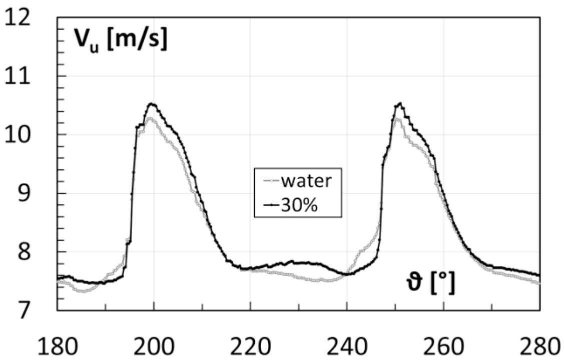

Fig. 8. Comparison of tangential component of absolute speed at the exit of the impeller with water and kaolin 30 %

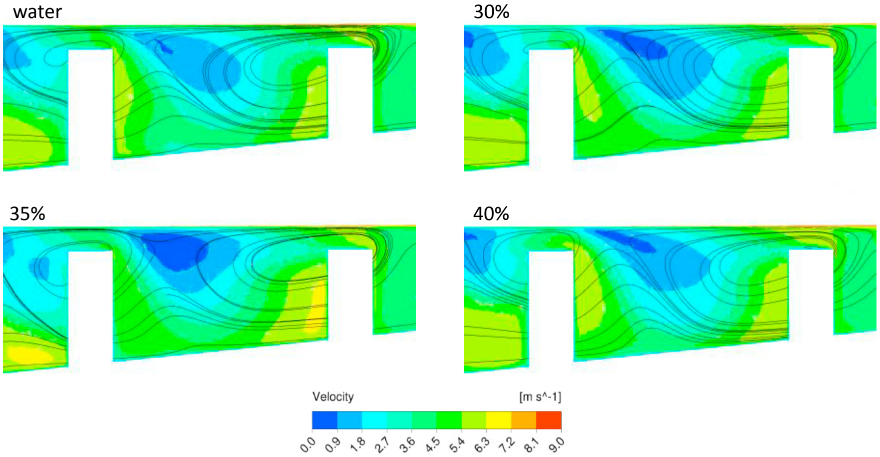

Fig. 9. Tip leakage flow for operation with water and kaolin 30 %, 35 % and 40 %

pump which can not be easily measured , such as shear rate, viscosity and velocity responsible of the performance modification and pump derating.

References

[1] Chhabra RP, Richardson JF. Non-Newtonian Flow In The Process Industries. First Edition 1999, Butterworth-Heinemann, Great Britain.
[2] Walker CI, Goulas A. Performance Characteristics of Centrifugal Pumps When Handling Non-Newtonian Homogeneous Slurries. Proc. Inst. Mech Eng 1984;198(A):41–9.
[3] Pullum L, Graham LJW, Rudman M. Centrifugal Pump Performance Calculation for Homogeneous and Complex Heterogeneous Suspensions. J South Afr Inst Min Metall 2007;107:373–79.
[4] Graham LJW, Pullum L, Slatter P, Sery G, Rudman M. Centrifugal Pump Performance Calculation for Homogeneous Suspensions. Can J Chem Eng 2009;87:526–33.
[5] Pullum L, Graham L, Wu J. Centrifugal pump performance with non-newtonian slurries." T&S 15th International Conference on transport and sedimentation of solid particles 6-9 September 2011, Wroclaw, Poland.
[6] Sery GA, Slatter PT. Centrifugal pump derating for non-Newtonian slurries." Proc. Hydrotransport 15, BHR Group Limited 2002;II:679–92.
[7] Sery G, Kabamba B, Slatter P. Paste Pumping with Centrifugal Pumps: Evaluation of the Hydraulic Institute Chart De-Rating Procedures. Proceedings of the Ninth International Seminar on Past and Thickened Tailings 3-7 April 2006, Perth, Australia, pp. 403–12.
[8] Kalombo JJN, Haldenwang R, Chhabra RP, Fester VG. Centrifugal Pump Derating for Non-Newtonian Slurries. J Fluids Eng 2014;136:031302.
[9] Furlan J, Visintainer R, Sellgren A. Centrifugal pump performance when handling highly non-newtonian clays and tailings slurries. BHR Group's 19th International Conference on Hydrotransport, Golden 24-26 September 2014, Colorado, USA.
[10] Buratto C, Pinelli M, Spina PR, Vaccari A, Verga C. CFD Study on Special Duty Centrifugal Pumps Operating with Viscous and Non-Newtonian Fluids. Proceedings of 11th European Conference on Turbomachinery Fluid dynamics & Thermodynamics ETC11, 23-27 March 2015, Madrid, Spain
[11] Allali A, Belbachir S, Lousdad A, Merahi L. Numerical approach based design of centrifugal pump volute. *Mechanika*, 2015;21:301–6.
[12] Ye DX, Li H, Zou CH, Jiang B. Characterization of the fluidization of medium consistency pulp suspensions in a centrifugal pump. WIT Trans. Eng. Sci 2015;89:67–76.
[13] Xuan-Linh N, Huan-Xin L. The Simulation of Non-Newtonian Power-Law Fluid Flow in a Centrifugal Pump Impeller. J Appl Mech Eng 2016;5:1000230.
[14] Buratto C, Occari M, Aldi N, Casari N, Pinelli M, Spina PR, Suman A. Centrifugal Pumps Performance Estimation with Non-Newtonian Fluids: Review and Critical Analysis. Proceedings of 12th European Conference on Turbomachinery Fluid dynamics & Thermodynamics, ETC12, April 3-7 2017; Stockholm, Sweden.
[15] Aldi N, Buratto C, Pinelli M, Spina PR, Suman A, Casari N. CFD Analysis of a non-Newtonian fluids processing pump. Proceddings of 71st Conference of the Italian Thermal Machines Engineering Association, ATI2016, 14-16 September 2016, Turin, Italy.
[16] Li WG. Effect of exit blade angle, viscosity and roughness in centrifugal pumps investigated by CFD computation. Task Quarterly 2011;15:21–41.
[17] Zubanov VM, Shabliy LS, Krivcov AV. Rotational Technique for Multistage Centrifugal Pump CFD-Modeling. ASME Paper 2016:GT2016-57689.
[18] Song X, Wood HG, Day SW, Olsen DB. Studies of Turbulence Models in a Computational Fluid Dynamics Model of a Blood Pump. Artificial Organs 2003;27:935–37.