Hydrodynamic modelling of the impact of viscosity on the characteristics of a centrifugal pump

V Tkachuk¹, H Navas¹, A Petrov¹ and A Protopopov¹,²

¹Bauman Moscow State Technical University, 5 Second Baumanskaya Street, Moscow, 105005, Russian Federation
²E-mail: proforg6@yandex.ru

Annotation
During operation of industrial centrifugal pumps, the main factor affecting its performance is viscosity, which can change its value during operation of the pump unit. In most cases, the main cause of such changes is a change in the environmental conditions or an increase in the temperature of the unit during operation. This paper explores the possibility of using computer simulation to calculate a centrifugal pump at various viscosities of the working fluid. The results obtained are compared with data from the American Hydraulic Institute.

Introduction
During the process of operation of a pump under new conditions of use or when the performance of an existing pump is being assessed, it is often necessary to consider the influence of the viscosity of the pumped liquid. Issues related to the operation of centrifugal pumps at high viscosities are widely reported in the literature [1–16]. It is known that the pressure curves presented in the catalogs of pump suppliers are constructed using water as the pumped liquid. These curves are suitable for use when the actual fluid used for pumping has a viscosity that is less than or equal to that of water. However, in some cases, this method is not suitable, for example, for some petroleum products and other liquids at low temperatures. The performance of a viscous working fluid pump is usually estimated using corrections applied to the characteristics obtained on water, since the test facilities of the pump manufacturers provide only tests on water, and most of the accumulated data and experience relate to characteristics obtained on water.

If a highly viscous fluid circulates through the pump and the piping system during the operation, more energy is needed to pump such fluid, because the friction between the fluid layers is much greater. As a result, the following phenomena occur:

• Reduced suction lift.
• Reduced pump flow.
• Reduced efficiency.
• Increased power consumption.

The purpose of this article is to simulate the operating conditions of a type D pump with the following characteristics (for water) with different viscosity values:

Table 1. Pump Characteristics

| Head pressure, m | Flow Rate, m³/h | Rotor speed, rpm | Cavitation stock, m | Efficiency, % | Specific speed, n_s |
|-----------------|----------------|-----------------|---------------------|--------------|-------------------|
| 80              | 80             | 2900            | 4,8                 | 67           | 42                |
Using CFD STAR CCM ++ simulation software, it is necessary to compare the results obtained with the hydrodynamic simulation software and the method proposed by the American Institute of Hydraulics. [9].

**Method**

The process of experimental determination of the effect of fluid viscosity on a working pump has now been studied quite well. In the book Centrifugal and Axial Flow Pumps, A.J. Stepanoff [7] lists losses that affect pump performance:

- Mechanical losses;
- Losses on the impeller due to reduced rotational speed;
- Leaks;
- Losses of disk friction.

To find out what the behavior of a viscous fluid pump is, the American Hydraulic Institute (HI) has developed a graphical method based on laboratory tests. The HI method suggests taking known pump data for water and applying some correction factors for viscosity, which are obtained from Figure 1. The nomenclature is defined below:

Ha - head with water; fH is the head correction factor for a specific viscosity;
Qa - Flow with water; fQ is the flow rate correction factor for a specific viscosity;
HPa - power with water; fη is the efficiency correction factor for a specific viscosity.

![Figure 1. The graph obtained by the American Institute of hydraulics](image-url)
Corrective formulas:

\[ H_v = H_a \times f_H. \]  
\[ Q_v = Q_a \times f_Q. \]
\[ HP_v = H_{Pa} \times f_Q \frac{f_H}{f_H}. \]

The following parameters are obtained through:
- Original water pump curve (Ha, Qa, HPa)
- Graph of viscosity correction from Hydraulic Institute (fH, fQ, fη)
- Application of the formula (Hv, Qv, HPv)

Limitations to consider

Before making estimates and explaining how to obtain correction factors, it is necessary to take into account the conditions of the method in order to avoid errors.

- **Flow Rate**: If the pump has a double suction impeller, consider half the flow rate of the pump, that is, the flow rate for each inlet section.
- **Head Pressure**: If the pump has more than one stage, it is necessary to consider the pressure of one stage.
- **Fluid viscosity**: Corrections are made for Newtonian liquids, i.e. liquids whose viscosity can be considered constant in time and which does not change with stirring.
- **Curve Range**: It is not recommended to extrapolate outside the range of the curves, because it can lead to a big error.
- **Viscosity limit**: There are various criteria regarding the maximum allowable viscosity in a centrifugal pump. Some authors recommend a maximum kinematic viscosity of 1000 cSt, but there are pumps that have an operating mode with liquids up to 3000 cSt.

**Mathematical model.** In order to obtain the hydrodynamic modeling, the following equations were used:

- Equation of continuity of the liquid medium:
  \[ \frac{\partial u_x}{\partial x} + \frac{\partial u_y}{\partial y} + \frac{\partial u_z}{\partial z} = 0, \]

  Where \( u \) – time-averaged projections of fluid velocities on the corresponding axes;

- Equation of the change in the amount of motion averaged over time:
  \[ \rho \left[ \frac{\partial u_i}{\partial t} + u_j \frac{\partial u_i}{\partial x_j} \right] = - \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_i} \left[ T^{(v)}_{ij} - \rho u_i u_j \right], \]

  Where \( u_i, p \) – averaged speed and pressure;
  \( T^{(v)}_{ij} = 2\mu \dot{s}_{ij} \) – viscous stress tensor for incompressible fluid;
  \( \dot{s}_{ij} = \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \) – strain rate tensor;
  \( \rho \{ u_i u_j \} \) – Reynolds stresses.

The Reynolds system of equations is open due to the presence of unknown Reynolds stresses. The system is closed using the k-ω SST turbulence model.

**Terms of computer simulation.** In the process of the computer simulation, a three-dimensional model of the flow part of the pump was created, and physical models and parameters of the
computational grid were set for it. The density value remained unchanged throughout the experiment. During the process of the hydrodynamic modeling with a change in viscosity, the flow rate, which is considered by the program, does not change, which is contrary to reality. Therefore, to obtain a valid approximation in the computer model, the flow rate was set according to the theory described above. Also, on the basis of the limitations for considering this technique, half the flow rate from the optimal value was taken, due to this type of pump. As a result of the experiment, the parameters (head and power) found using hydrodynamic modeling were compared with the parameters found by the graphical method.

Results

As an example, it was defined the correction factors for a liquid with a kinematic viscosity of 50 cSt, according to the graphs in Figure-1. The initial data from which the recalculation will be carried out are indicated below: \( Q_a = 40 \text{ m}^3 / \text{h}; \ Ha = 80 \text{ m}; \ HP_a = 26.4 \text{ kW} \)

There were obtained three correction factors: \( f_\eta = 0.76; \ f_Q = 0.98; \ f_H = 0.95 \)

Further, analogically, were found the correction factors and recalculate the indicators for a liquid with different kinematic viscosity values, increasing the subsequent value by 50 cSt and to 600 cSt. The results of recalculated indicators are shown in Table 2.

| viscosity (cSt) | 50 | 100 | 150 | 200 | 250 | 300 | 350 | 400 | 450 | 500 | 550 | 600 |
|----------------|----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|
| \( f_\eta \)  | 0.76 | 0.66 | 0.63 | 0.54 | 0.50 | 0.45 | 0.43 | 0.37 | 0.36 | 0.33 | 0.30 | 0.28 |
| \( f_Q \)   | 0.98 | 0.96 | 0.94 | 0.90 | 0.87 | 0.85 | 0.84 | 0.80 | 0.76 | 0.75 | 0.73 | 0.71 |
| \( f_H \)   | 0.95 | 0.93 | 0.90 | 0.87 | 0.86 | 0.84 | 0.84 | 0.81 | 0.80 | 0.79 | 0.78 | 0.76 |
| \( Hv \)    | 76.00 | 74.40 | 72.00 | 69.60 | 68.80 | 67.20 | 67.20 | 64.80 | 64.00 | 63.20 | 62.40 | 60.80 |
| \( Qv \)    | 39.20 | 38.40 | 37.60 | 36.00 | 34.80 | 34.00 | 33.60 | 32.00 | 30.40 | 30.00 | 29.20 | 28.40 |
| \( HPv \)  | 32.34 | 35.71 | 35.45 | 35.28 | 39.50 | 41.89 | 43.32 | 46.24 | 44.59 | 47.40 | 50.11 | 50.88 |

After the recalculation of indicators has been carried out, we compare them with the values calculated in STAR CCM + for the same kinematic viscosity values. The results of the indicators calculated in STAR CCM + are shown in Table 3.

Table 3. Parameters obtained in STAR CCM +

| viscosity (cSt) | 50 | 100 | 150 | 200 | 250 | 300 |
|----------------|----|-----|-----|-----|-----|-----|
| \( Hv \)    | 74.92 | 67.91 | 65.62 | 62.92 | 60.94 | 58.68 |
| \( Qv \)    | 78.19 | 76.80 | 75.20 | 71.74 | 69.00 | 68.00 |
| Efficiency    | 0.5789 | 0.5278 | 0.4964 | 0.4690 | 0.4402 | 0.4224 |
Окончание табл. 3

| Head (m) | Flow Rate (m³/h) | Efficiency |
|---------|------------------|------------|
| Hv      | Qv               |
| 350     | 56.02            | 66.97      | 0.4014 |
| 400     | 55.76            | 64         | 0.3905 |
| 450     | 55.72            | 60.8       | 0.3750 |
| 500     | 53.93            | 59.75      | 0.3682 |
| 550     | 52.46            | 58.4       | 0.3502 |
| 600     | 52.16            | 56.59      | 0.3499 |

A comparison of all parameters is shown in Figure 2 and Figure 3.

![Figure 2. Head Comparison graph](image)

![Figure 3. Power comparison graph](image)
Conclusion

As a result of the experiment, it can be concluded that the values obtained using the bench practically correspond to the values obtained using computer simulation. However, with viscosities above 200cSt (200 mm² / s), the indicators have significant discrepancies, which can be observed in Figure 2 and Figure 3. This phenomenon can be explained for several reasons:

• In the computer model it is not possible to specify all the conditions under which the pump will work;
• In computer simulation, a pump model with a low coefficient of speed ns was used, such pumps are not intended for pumping highly viscous liquids;
• Also in the computer model, leakages between areas and leakages were not taken into account.

Thus, in the course of the experiment, the main assumptions for viscous tests using hydrodynamic modeling were identified. Were built comparative graphs for pressure and power.

Literature

[1] DOI: 10.1016/j.cnsns.2015.10.001. Lomakin, V.O., Kuleshov, M.S., Bozh’eva, S.M. Numerical Modeling of Liquid Flow in a Pump Station (2016) Power Technology and Engineering, 49 (5), pp. 324-327.

[2] DOI: 10.1007/s10749-016-0623-9. Lycheva, T.N., Lychev, S.A. Spectral decompositions in dynamical viscoelastic problems (2016) PNRPU Mechanics Bulletin, (4), pp. 120-150.

[3] DOI: 10.1016/S0142-727X(99)00078-8. W.-G. Li, “Effects of viscosity of fluids on centrifugal pump performance and flow pattern in the impeller,” International Journal of Heat and Fluid Flow, vol. 21, no. 2, pp. 207–212, 2000.

[4] DOI: 10.1007/s10010-006-0045-1. A. Nemdili and D. H. Hellmann, “Investigations on fluid friction of rotational disks with and without modified outlet sections in real centrifugal pump casings,” Forschung im Ingenieurwesen, vol. 71, no. 1, pp. 59–67, 2007.

[5] DOI: 10.1016/S0262-1762(00)87528-8. J. F. Gülich, “Pumping highly viscous fluids with centrifugal pumps—part 1,” World Pumps, vol. 1999, no. 395, pp. 30–34, 1999.

[6] DOI: 10.1115/GT201542014. K. Juckelandt and F.-H. Wurm, “Applicability of wall-function approach in simulations of turbomachines,” in Proceedings of ASME Turbo Expo 2015: Turbine Technical Conference and Exposition (GT ’15), June 2015.

[7] Stepanoff, A. J. Centrifugal and Axial Flow Pumps. Theory, Design and Application (1957) KRIEGER PUBLISHING COMPANY, pp. 308-317

[8] Lobanoff V. S. Ross R. R. Centrifugal Pumps: Design and Application (2013) Gulf Professional Publishing, pp. 305-310

[9] Hydraulic Institute (U.S.) Hydraulic Institute standards for centrifugal, rotary & reciprocating pumps (1983) Michigan University, pp.111

[10] Dickenson T. C. Pumping Manual (1995) Oxford: Elsevier Advanced Technology, pp. 36.

[11] Turzo, Z., Takacs, G. and Zsuga, J., "Equations Correct Centrifugal Pump Curves for Viscosity" (2000), Oil & Gas Journal, pp. 57-61

[12] Mathematical modeling of the mechanisms of volumetric hydraulic machines.S. Semenov and D. Kulakov 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012042

[13] Methodology of dynamic pumps testing on high-viscosity liquids. K. Dobrokhodov and A. Petrov 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012016

[14] Karassik, I. y R. Carter. Bombas centrífugas. Selección, operación y mantenimiento (1985). Compañía Editorial Continental, México, pp. 143

[15] Turino, I. M. “Determinación aproximada de las características de funcionamiento de una bomba centrifuga” (1994), Centro Azúcar, Cuba, pp.34

[16] A. T. Ippen, “The influence of viscosity on centrifugal pump performance,” Transactions of the American Society of Mechanical Engineers, vol. 68, no. 8, pp. 823–848, 1946
[17] W. G. Li and Z. M. Hu, “An experimental study on performance of centrifugal oil pump,” Fluids Machinery, vol. 25, no. 2, pp. 3–7, 1997.
[18] A Protopopov and D Bondareva 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012002
[19] A Protopopov and V Vigovskij 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012003
[20] A Petrov et al 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012040
[21] The effect of design parameters of the closed type regenerative pump the energy characteristics. N Isaev 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012026