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The Effect of Rotational Speed on the Performance of the Electric Submersible Pump

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A B S T R A C T

Electric Submersible Pump (ESP) is a centrifugal pump with several stages. It's a dependable and efficient industrial lifting tool for lifting different amounts of fluids, and it's known for its adaptability and reliability. ESPs are commonly used in onshore oil wells, as well as underground pumping and other applications. Pump's efficiency is evaluated under the control of various rotational speeds. CFD approach has been used in this study to analyze flow behaviors within the ESP at various flow speeds. ANSYS CFX program was approved to conduct and investigate single-stage simulation of the GN7000 pump. Also, the sensitivity of the network has been determined. The flow field has been solved by using SST, Unsteady-RANS model with different rotational velocities (2500-3000-3500 cycle in this case). The test used two types of fluids with varying viscosities. Findings revealed that the viscosity has an impact on efficiency as well. There is a high level of compatibility as compared to previous studies. This research presented a set of curves for efficiency and pressure and to know the extent of the effect of rotational speed on performance in general.

1. Introduction

ESPs are one of the most widely used synthetic lifting technologies in the industry because of its broad applicability in viscous oil flows Thomas et al., [1]. Fluid receives energy in the form of pressure, which raises to the surface in this method of pumping. ESP is divided into stages, each of them consists of a rotor that moves vertically, in additional to diffuser that is fixed (stationary). Due to a strong rotation exerted on it, the rotor provides the velocity of the fluid then its guided to the pump's outlet by the diffuser , in other words, to the next stage of the pump. For example, The rotor and the diffuser of REDA GN7000 style of centrifugal pump used in the lift oil are shown in **Fig. 1.**

In recent years, A huge number of studies have been conducted aimed at a better understanding of ESP such as, Estevam, [2] and Amaral, [3]. discovery of oil, make companies to increase their use of this tool along the world. Also, The electrostatic precipitator itself is more effective in marine applications than other synthetic lift systems

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Nomenclature

g	gravity acceleration (m s^{-2})	*Greek symbols*	
H	head (m)	μ	dynamic fluid viscosity $kgm^{-1}s^{-1}$
$Q_{des,w}$	the volumetric flow rate (m^3/s)	ρ	density kg m^{-3}
S_{Cor} S_{cfi}	the Coriolis and centrifugal effects	η	pump efficiency %
P	pressure (pa)	ϕ	specific capacity
Re_w	rotating Reynolds number	Γ	generic diffusion coefficient of the general transportation
D_h	hydraulic diameter (m)		
D_s	length scale of the pump geometries	*Subscripts*	
U	average peripheral or tangential flow velocity (m s^{-1})	i	inlet
Δp	pressure difference	o	outlet
t	time	Z	blade number
		D	diameter
Symbols		b	blade height
∂	Derivative	e	blade thickness
∇	Gradient	β	blade angle

It is necessary to know that pressure losses present in both rotor and diffuser at each point in order to test the ESP system's efficiency. The efficiency of the pump is determined by the output of each point. Pump output curves are crucial since they describe the pump's efficiency. The impact of various spin speeds on pressure differences, as well as the impact on the efficiency of the ESP and other factors that cause failure, will be discussed. It is critical to understand each category of failure in order to assess results. The Hydraulic Institute's pump curves are commonly used in the industry and are a valuable guide for the use of centrifugal pumps. However, since these curves are built with water as the test fluid, they are inadequate as a benchmark for ESP results. In oil industry, oil from different regions have different properties, viscosity has a direct impact on the velocity and the pressure losses in the ESP level, and thus on their efficiency. Several previous experiments on the effects of rotational velocity and viscosity have been conducted, and the findings of these studies have been published. Researchers have studied the effect of speed on performance for example, Yang Y. et al. [4], Zhou L. et al. [5], and others who studied the effect of viscosity on ESP performance like Amaral et al. [6], Gulich J.F [7], Barrios L.J. et al. [8], Paternost G.M. et al. [9], Zhu J. et al. [10], Ofuchi E.M. et al. [11], Patil A. and Morrison G. [12].

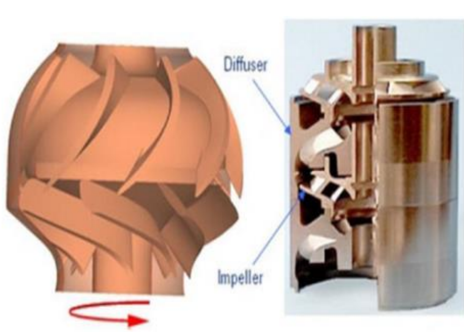

Figure 1- Vane diffuser and Impeller blade, one stage, [3].

One of the assumptions in this paper is that the temperature is constant and fluid is in one phase, and the test was done by both water and oil fluids. Noting that the viscosity of water is 1cp, and Thi Qar oil for one of its wells has a viscosity of 24.98cp at a temperature of 21◦C. When comparing this research with Stel's research, in the current research it has been found a good match through the values has obtained for the pressure difference of water at the available rotational speed (3500 rpm) and at Q/Qdes =1.5. The value of the pressure difference was (41,000pa) for stel et al. [13] and it was (-42,168pa) for the current research) with an error of 2.81% it is a good number for the purpose of investigation.

In the current study, an attempt has been performed to find the best performance of ESP in terms of (rotational speed, the losses that occur within a single stage, network sensitivity test and validation of the designed model with previous researchers).

2. Numerical Methods

2.1. Geometry model and mesh generation.

Model has been designed by using AutoCAD software for the GN7000 pump and it has been simulated by ANSYS20 to generate a grid mesh for it. The mesh used in this study and shown in **Fig. 2**. The whole field group is about 3,010,000 elements. The network is mainly composed of tetrahedral elements and is at the edges of the blades and the subfield interfaces. As well as the hexahedral elements used in the internal field. The sensitivity of the network was tested taking into account the following parameter (Control Body Scaling - Adjust Amplification - Face Tangle Control).

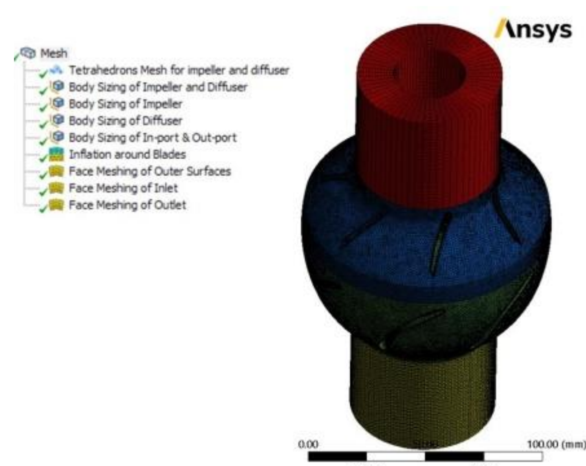

Figure 2- Control processes for mesh generation.

Fig.3 illustrates the control process in grid generation where the operating load is set to flow rate Q/Q_{des}=1and rotation speed of 3500 rpm to calculate the pressure difference values across the field. In **Fig.3** It has been noticed that the greater the number of elements of the grid by making improvements, the less the deviation in the pressure difference. Therefore, the closest value was obtained through the grid used in this research in comparison with the research of Stel et al. [13] and was adopted in this study.

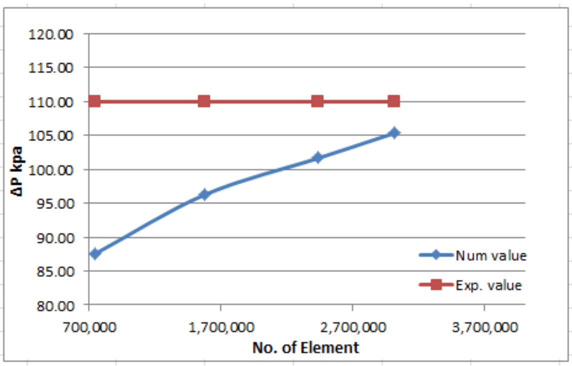

Figure 3- Mesh sensitivity test results; comparison between numerical results and experimental value [3].

2.2. Boundary conditions and main specifications

Fluids have been modelled by using ANSYS CFX software in a variety of ways, including geometry and grid forming. Fluid analysis problem has been solved by using pre-treatment based on Navier-Stokes's equations. The CFX program can solve unstable governing equations. The mass and momentum equations known as the Reynolds - Averaged Navier - Stokes's equations (U-RANS) can be expressed in the following general form:

$$\rho\left(\partial\phi/\partial t + \nabla.(\vec{V}\phi)\right) = \nabla.(\Gamma\nabla\phi) + S \qquad (1)$$

The pump has designed to spin at 3500 rpm (58.33 rev/sec) with n_{des}=3500 rpm. The producer's productivity is at its peak when working with water. It has a volumetric flow rate of $1.360x10^{-2}$, a mass flow rate of 13.6 kg/s, and a head of 9.6 m. That the designing volumetric flow rate will change if the impeller's spinning speed changes.;

$$Q_{des,w} = (n/3500)\times 1.36\times 10^{-2} \qquad (2)$$

There are four subdomains in the dominant domain. The in-port domain (the intake), Rotor domain (the impeller), the Stator domain (the diffuser), and the out-port domain (the discharge port) are the four domains, respectively. The inlet face has a continuous reference pressure, pref = 0 (Pa), with a turbulence slope and flow direction of zero. Facade surfaces are used to link sub-domains together (connecting fixed and rotating areas). Non-slip surfaces are required on all surfaces. A boundary state interface with a given discharge fluid flow rate value is known as the outlet surface.

3. Results and Discussion

3.1. Pressure difference

All simulation conditions have been summarized in **Table 1**; and all of these cases simulated for 2500, 3000 and 3500 rpm. By using MPI parallel calculation with 6 CPU processors with a PC of specifications (Intel Core^{TM} i9 CPU 3.5 GHz 32 GB RAM). The simulations need 33 runs where each run demands approximately 36 hrs.

Table.1 Summary of simulation conditions

Case	Working Fluid	Viscosity (cP)	Density (kg/m3)	Flow rate (kg/s)	Reynolds number
1	Pure Water	1	998.2	3.4	47,674
2	Pure Water	1	998.2	6.8	95,348
3	Pure Water	1	998.2	10.2	143,022
4	Pure Water	1	998.2	13.6	190,696
5	Pure Water	1	998.2	17	238,370
6	Pure Water	1	998.2	20.4	286,044
7	Oil	24.98	889	3.022	1,699.705
8	Oil	24.98	889	6.045	3,399.410
9	Oil	24.98	889	9.0678	5,099.116
10	Oil	24.98	889	12.0904	6,798.82
11	Oil	24.98	889	15.113	8,498.52

However, the **Fig. 4** shows the pressure difference with respect to flow rate values for water as a working fluid at three different impeller speeds. It has shown that as the angular velocity of impeller increases, the pressure difference also increases. This is because as impeller rotating became faster, the flow behaves as a blockage at the rotor zone. Thus, more flow force needs to penetrate this block of fluid, which lead to increase pressure difference across the pump stage. In addition, in the high flow rate cases, the effect of rotational speed degrades on pressure difference. It was found that the pressure changes for volumetric flow rate of Q/Q_{des} = 1.5 is 44.8141% and 32.0046% for 2500 rpm and 3000 rpm respectively lower than n=3500 rpm case. And in the case of Q/Q_{des} = 0.5, the pressure difference is 71.9994% and 37.298% for 2500 rpm and 3000 rpm respectively lower than n=3500 rpm case.

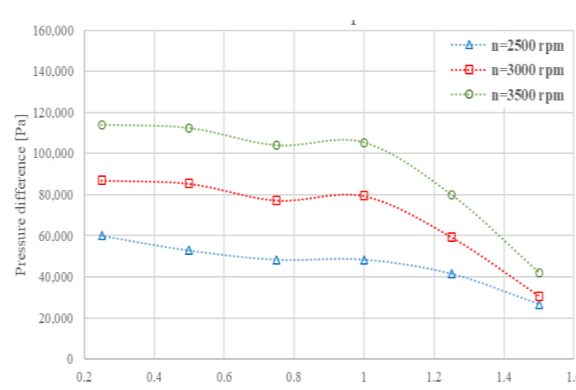

Figure 4-Comparison of pressure difference for water 1cp, at several rotor speeds.

The **Fig. 5** shows similar comparison for the 24.98 cp fluid. The same observations verified by look to these curves. The pressure difference proportion directly to the angular velocity of impeller.

It's evident from the previous estimates that as the flow rate increases, the velocity of the inlet flow rises with it. This rise in inlet velocity induces more shearing of the fluids and more energy loss. As a result, the pressure differential decreased in situations where the flow rate was higher. Also, in the following figure, it's obvious that for higher fluid viscosity, the pressure rise is lower. It is the same influence of flow rate increasing.

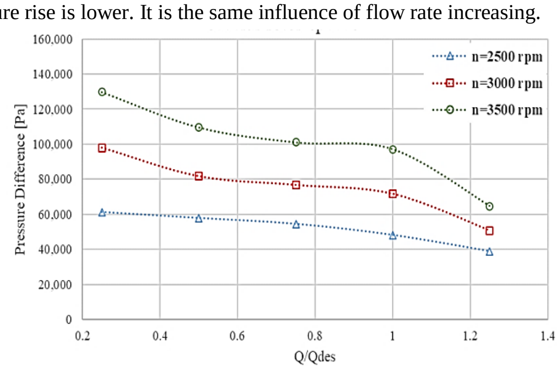

Figure 5- Comparison of pressure difference for oil 24.98 cp, at several rotor speeds.

Also, from the above comparison, it can be observed that the pressure rise is highest in the lowest flow rate condition for each fluid. As the flow rate become higher, the pressure difference decreased for fluid with operating condition or high-capacity condition.

For the case of Q/ Qdes,w = 1.5 at 3500 rpm, the pressure contour is shown in the **Fig.6.** The higher-pressure difference appears in the impeller region due to rotational motion. The secondary flow generated lead to high pressure difference around the blades. Therefore, the pressure difference in the impeller is approximately like the pressure change of the whole stage.

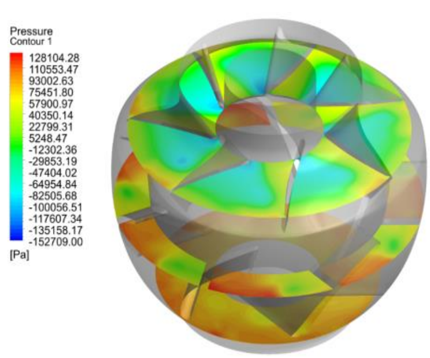

Figure 6- The pressure contour for the case of Q/ Qdes, w = 1.5 at 3500 rpm.

3.2. Analyze the results

In order to approximate the pump efficiency, the following equation was used:

$$\eta = (\rho\ g\ Q\ H)/T\omega = (Q\ \Delta p)/T\omega \qquad (3)$$

Q represents the flow rate (m3/s), H represents the head (m), represents the angular speed (rad/s), and T represents the torque (N.m). Torque that can be measured directly from the CFD-post function calculator in the ANCYC CFX program. The XY diagrams for the pump's performance in clear water and another fluid with a viscosity of 24.98 cp for various impeller speeds are published in the following figures. A comparison of two fluids (water and oil) at two different rotational speeds is seen in the diagram below (3000 and 3500 rpm). At the same rotational speed, the highest output point for various viscosity fluids is (3500 rpm). When working with water, Q / Qdes = 1 is the flow rate at the best performance stage. When treating a fluid with a viscosity of 24.98cp, the flow rate drops to Q / Qdes = 0.8 at the best Efficiency stage. When it is interacting with a lower rotational speed, the same explanation can be seen (3000 rpm). The best performance point has declined, as we can see. It might assumed that the rotational speed has a significant impact on efficiency and that the viscosity of the ESP is affected.

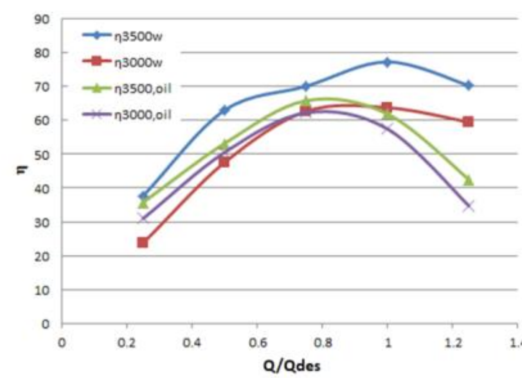

Figure 7- Best efficiency points are different for various rotational speed and viscosity fluids.

The best performance point shifts from Q/Qdes =0.8 at 3500 rpm to roughly Q/Qdes =0.68 at 2500 rpm, as seen in the graph above. This is because as the rotational speed reduces, so does the angular flow speed. For a constant volumetric flow rate, this resulted in the velocity vector in diffuser flow not being in the same direction. To achieve the same combination of axial and angular velocity in the diffuser component, the flow rate should be reduced as well. In the diffuser and out-port areas, this mixture resulted in blade-oriented flow. However, in the considered ESP model, the best performance point shift for both water and oil flow.

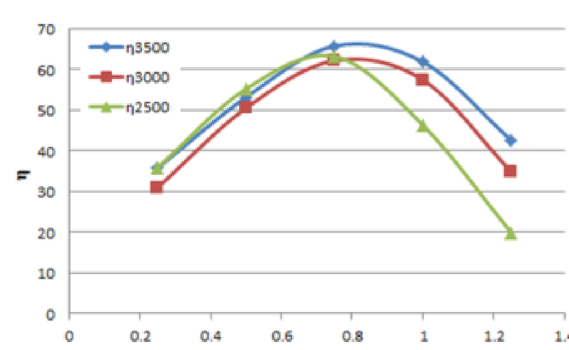

Figure 8- Efficiencies for three rotational speeds is explained in XY plot for oil flow.

3.3. Dimensionless analyses

It was determined to do a dimensionless analysis. The dimensionless numbers were used in this analysis to investigate the effect of viscosity on ESP performance. The equations below have been used to calculate the dimensionless numbers. The precise capability;

$$\Phi = \frac{Q}{\omega D_s^3} \tag{4}$$

Using Q is the stage's flow rate (m3/s), $\omega$is the impeller's angular speed (rad/s), and Ds is the impeller outlet mean diameter, which is the average of the inner and outer diameters. The value of Ds in this model is 0.0884 m. There's also the spinning Reynolds number to consider.

$$Re_w = \frac{\rho\omega D_s^2}{\mu} \tag{5}$$

where ρ is the operating fluid's density (kg/m3) and μ is the fluid's mechanics viscosity (kg/m.s). The rotating Reynolds numbers for all simulation cases are described in the **Table 2.** The rotational Reynolds number gives the same impression as the traditional Reynolds number.

Table.2. Rotating Reynolds numbers for all simulated cases.

Water			Oil		
n=2500	n=3000	n=3500	n=2500	n=3000	n=3500
2042161.47	2450597.4	2859030.3	72808.459	87370.2803	101931.994

The increasing of velocity (in this case angular velocity) led to increasing Re_w. The increase of viscosity results to reducing of Re_w value. However, the **Fig. 9** shows the efficiency versus specific capacity;

It has been shown that increasing the revolving Reynolds number increased pump efficiency. Points calculated from simulation data. Using Microsoft Excel, the curves were applied using a polynomial of the fourth order trend. These performance curves have a starting point that is roughly at the origin. Based on figure, the efficiency of ESP is a function of the specific capacity and the rotating Reynolds number.

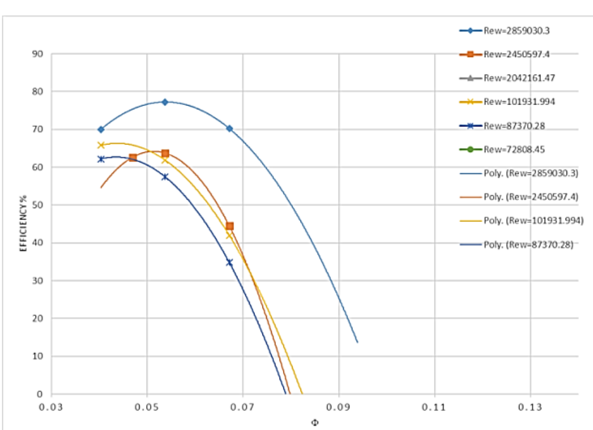

Figure 9- Efficiency versus specific capacity.

4. Conclusion

This research has been presented a numerical study of the GN7000 ESP that was designed and tested under different rotational velocities and flow rates to create specific operating conditions.

The conclusion includes the following aspects:

1. The arithmetic field was created for one stage with a high-quality structural network, and the sensitivity of the network was analyzed to determine the appropriate network and it was good when compared with previous research.
2. The results showed that the pressure difference with respect to the values of the fluid flow rate working at three different impeller speeds, that with increasing the rotational speed, the pressure difference increases.
3. In high flow rate situations, the effect of rotational velocity on the pressure difference decreases.
4. When comparing the rotational speed (3500-3000-2500 rpm), it was found that the best performance point for different viscous fluids with the same rotation speed is 3500rpm.

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