

Flow Pattern and Boosting Pressure Prediction of Electrical Submersible Pump (ESP) in Gas-Liquid Flow

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Abstract

The performance of widely used Electrical Submersible Pump (ESP) is significantly affected by gas entrainment, which is an encountered phenomenon in the petroleum industry. The boosting pressure of an ESP gradually degrades with the increase of inlet gas void fraction (GVF). The flow becomes unstable and a breakdown occurs when the flow pattern switches from bubble flow to intermittent flow. Therefore, an accurate ESP model is necessary to help design and operate the ESP system under gassy flow conditions. The gas-liquid two-phase tests of different ESPs were extensively carried out at the Tulsa University Artificial Lift Projects (TUALP) to investigate the complex flow behaviors. Then, a mechanistic model was proposed for the gas-liquid flow inside a rotating ESP, which captures the gas-liquid two-phase flow characteristics, including in-situ gas void fraction, flow pattern transition, bubble size, boosting pressure, etc.

In the new model, the pump head is calculated by subtracting recirculation head loss, turning head loss, friction head loss, and friction head loss from Euler head. The recirculation head loss is estimated by a new theory, the best-match flowrate Q_{BM} . It is generated by the mismatch of the velocity of real flowrates and Q_{BM} . The drag force coefficient correlations are selected and improved based on flow patterns and bubble sizes. Then, the mixture density and in-situ gas void fraction are calculated according to the force balance on gas bubbles. The proposed model is validated by experimental data of a radial-type ESP (TE2700) and two mixed-type ESPs (GC6100 and MTESP) in the TUALP database. The predicted ESP boosting pressures and flow patterns agree well with the corresponding experimental

measurements.

Keywords: Artificial Lift, Electrical Submersible Pump (ESP), Gas-Liquid Flow, Experiments, Mechanistic Modeling

1. Introduction

The electrical submersible pumps (ESPs), with more than 200,000 installations, are widely used in the petroleum industry, especially in the offshore fields with high production rates [1]. However, complex flow conditions, including high viscosity oil, gas, and sand, jeopardize ESP performance and reduce its lifespan. The internal multiphase flow structure is difficult to be characterized due to its complex geometry, resulting in obstacles in ESP modeling. The first experiment of gas-liquid flow in a centrifugal pump was conducted by Murakami and Minemura [2, 3], which is a visualization test with a semi-opened transparent impeller (rotor). The ESP's hydraulic head degrades with the increase of gas entrainment, especially when gas pockets are formed. The effect of pump hydraulic parameters on its boosting pressure, including pump types [4], stage number [5], rotational speed [6], inlet pressure [7], surfactant concentration [8], seal clearance [9], grooves on volute [10] were studied separately. At TULAP, two-phase performance experiments were extensively carried out for different ESPs, including GN7000 [11], GN4000 [4, 11], GC6100 [5, 12, 13, 6, 14], DN1750 [15], TE2700 [7, 1], and MTESP [16]. Specifically, gas-liquid visualization experiments were carried out for GC6100 [17, 14, 18].

With the understanding of gas-liquid two-phase flow characteristics inside ESPs from experimental studies, the theoretical work on calculating the ESP boosting pressure under both single-phase and gassy conditions has been conducted by researchers. Based on the ideal assumptions (incompressible, frictionless, infinite blades/vanes, static flow), Euler head has been incorporated in most existing models to estimate the boosting pressure of a centrifugal pump. The velocity triangles in an ESP impeller is shown in Figure 1, in which the global absolute velocity C is composed by impeller relative velocity W and peripheral velocity U . Figure 1 (b) and (c) show the triangle relationship between C , W , and U at the impeller inlet and outlet, where the subscript 1 and 2 represent inlet and outlet. C'_2 is the ideal absolute velocity assuming an infinite number of impeller blades.

Stepanoff (1958) introduced the Euler head (H_E) that developed based

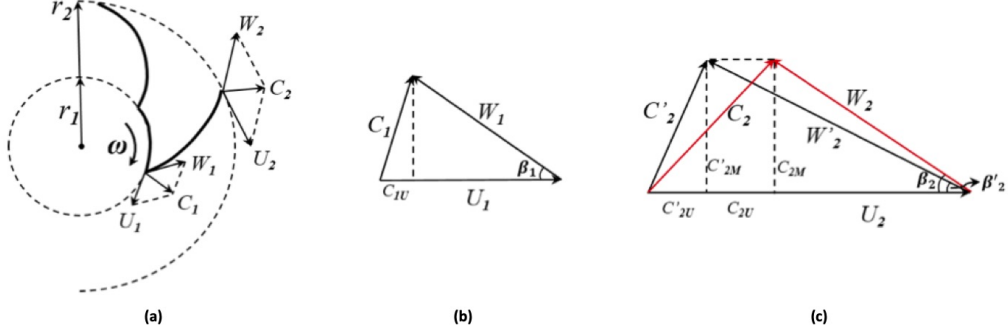


Figure 1: Velocity triangles in an ESP impeller, (a) impeller flow channel, (b) inlet velocity triangle, (c) outlet velocity triangle.

on the ideal angular momentum conservation equations in centrifugal pumps
as:

$$H_E = \frac{\vec{C}_2 \cdot \vec{U}_2 - \vec{C}_1 \cdot \vec{U}_1}{g} = \frac{C_2 U_{2U} - C_1 U_{1U}}{g} \quad (1)$$

where the subscript U represents the peripheral direction. According to the
velocity triangles in Figure 1, the Euler head can be re-written as:

$$H_E = \frac{U_2^2 - U_1^2}{2g} - \frac{W_2^2 - W_1^2}{2g} - \frac{C_2^2 - C_1^2}{2g} \quad (2)$$

and

$$H_E = \frac{\Omega^2(r_2^2 - r_1^2)}{g} - \frac{Q\Omega}{2\pi gh} \left(\frac{1}{\tan \beta_2} - \frac{1}{\tan \beta_1} \right) \quad (3)$$

where r is the radius of the impeller, h is the channel height, β is the blade
angle from the tangential direction. If β_1 is close to 90° or the fluids enter
the impeller without pre-rotation, Equation 3 can be written as

$$H_E = \frac{\Omega^2 r_2^2}{g} - \frac{Q\Omega}{2\pi gh \tan \beta_2} \quad (4)$$

Then, the relationship between the ideal Euler head and outlet blade
angle β_2 is shown in Figure 2 (a). The real pump head is calculated by
subtracting the head losses in Figure 1 (b) from the Euler head in Equation
4 as:

$$H = H_E - H_{friction} - H_{shock} - H_{leakage} - H_{recirculation} - H_{diffuser} - H_{disk} \quad (5)$$

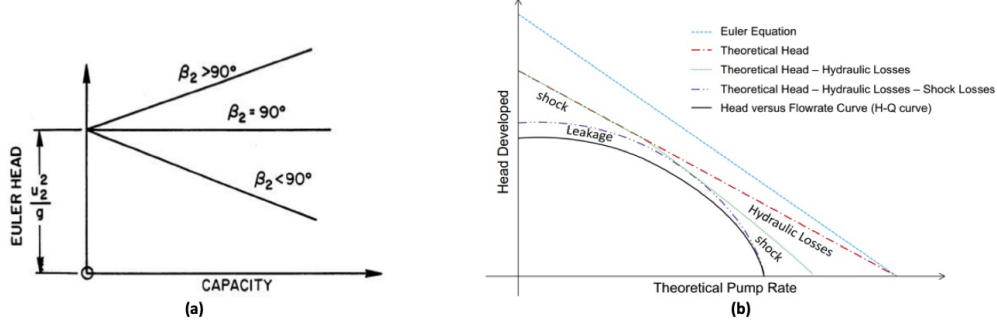


Figure 2: Schematic of head curves, (a) Euler head with different outlet blade angles, (b) actual head after deducting losses (Vieira et al., 2015).

The remaining difficulty is to determine different types of head loss and gas effect to mixture density in an impeller. For two-phase modeling of gas-liquid performance inside a rotating centrifugal pump, several one-dimensional (1D) two-fluid models have been proposed in the literature. In 1980, a 1D control volume method was proposed by Zakem to analyze the gas-liquid interaction effect in straight channel impellers. Similarly, Furuya [19] established an analytical model by considering pump geometry, GVF, slippage, and flow patterns. However, the fluid compressibility and condensation effect were neglected, which results in an average error of $\pm 30\%$ for $\text{GVF} < 20\%$ and $\pm 50\%$ for $\text{GVF} > 30\%$. Then a comprehensive investigation of ESP two-phase flow, including air-water and diesel- CO_2 , was carried out by Sachdeva et al. [20, 21], based on which a dynamic five-equation, 1D, two-fluid model was developed and validated. Besides, Minemura et al. [22] developed a two-phase model for centrifugal pumps based on energy changes from rotating impeller and stationary volute, which considers fluid viscosity effect and gas compressibility. Although pump geometry and momentum balance equations are included in Minemura et al. [22] and Sachdeva et al. [21] models, they are restricted to narrow experimental flow conditions and pump types. Based on their studies, Sun [23, 24] and improved correlations for head losses and drag coefficient, and developed a new two-phase model based on mixed-type GC6100 testing results at TUALP by solving a set of 1D conservation equations. However, the model cannot predict the boosting pressure of radial-type TE2700 under two-phase flow conditions with acceptable accuracy.

Although experimental and theoretical studies on ESP under gassy flow are introduced, most of them are only validated by their pump and testing conditions. The flow pattern transition, gas-liquid slippage, gas bubble size effect on in-situ gas void fraction and mixture density in ESPs are not well understood. Mechanistic modeling of ESP two-phase performance is still preliminary in the literature. Experimental investigations are insufficient on ESP two-phase flow behaviors including flow patterns and transition boundaries. The development and validations are also critical for the accuracy of model predictions. In this paper, a mechanistic model is proposed to predict the flow patterns and boosting pressure under gas-liquid flow inside a rotating ESP. The model, validated by the three pump curves in the TUALP database, including 5-inch mixed-type GC6100 [14], 5-inch radial-type TE2700 [1], and 4-inch mixed-type MTESP [16], can capture the multiphase flow characteristics, such as flow pattern transition, in-situ gas void fraction \aleph_G , bubble sizes, etc. The predicted ESP boosting pressure of three ESPs shows good agreement with the experimental data.

2. Model Development

This section discusses the mechanistic model for predicting ESP performance, including single-phase liquid and gas-liquid two-phase modeling, as well as the closure relationships for estimating the representative bubble size, the in-situ gas void fraction α_G , and drag coefficient, etc. Meanwhile, the comparisons of mechanistic model predictions with corresponding experimental data are also presented.

2.1. Single-phase Liquid Performance

Takacs [25] listed three head loss types in centrifugal pumps, namely hydraulic losses, shock losses, and leakage losses. The actual pump head is the result after subtracting all the head losses from Euler head. The hydraulic losses caused by fluid friction and diffusion losses inside impeller channels increase steadily with the liquid flow rate. The shock losses are negligible at the best efficient point (BEP) but increase at lower or higher liquid rates, which is due to sudden changes in flow direction at the inlet and outlet of the impeller. The leakage losses always exist as long as the liquids flow through the clearances between the rotating and stationary parts of the pump stage, including the impeller eye, balancing holes. However, the leakage losses diminish with the increased liquid flow rates due to lower backpressures.

104 In this study, the mechanistic model of ESP boosting pressure starts from
105 Euler equation and introduces a conceptual best match flow rate (Q_{BM}), at
106 which the flow direction at the impeller outlet matches the designed flow
107 direction. The recirculation head loss is calculated by the mismatch of the
108 velocity triangles compared to the Q_{BM} .

109 2.1.1. Liquid Flow Rate $Q_l < Q_{BM}$

110 Based on the velocity triangles in Figure 1, three terms at the right-hand
111 side (RHS) of Equation 2 are the hydraulic heads as a consequence of
112 centrifugal force, velocity change through the impeller as well as dynamic
113 effect, respectively. For each velocity component, its expression is discussed
114 below. The tangential velocity at the impeller inlet is given by

$$U_1 = R_1 \Omega \quad (6)$$

115 where R_1 is the radius of the impeller inlet, and Ω is the angular velocity
116 of the impeller. Similarly, the tangential velocity at the impeller outlet is
117 expressed as

118 2.1.2. Liquid Flow Rate $Q_l > Q_{BM}$

119 2.1.3. Friction Loss

120 2.1.4. Leakage Loss

121 2.2. Gas-Liquid Performance

122 2.2.1. Closure Relationships

123 3. Results and Discussions

Treatments	Response 1	Response 2
Treatment 1	0.0003262	0.562
Treatment 2	0.0015681	0.910
Treatment 3	0.0009271	0.296

Table 1: Table caption

124 3.1. Single-phase Flow Results

125 3.2. Gas/Liquid Two-phase Flow Results

126 3.2.1. Mapping Test Results

127 3.2.2. Surging Test Results

128 4. Conclusions

129

130 Nomenclature

131 General

132 g Gravitational Constant $6.67384 \times 10^{-11} N \cdot m^2/kg^2$

133 c Speed of light in a vacuum inertial system $299,792,458 m/s$

134 h Plank Constant $6.62607 \times 10^{-34} Js$

135 Greek Symbols

136 \mathbb{H} Quaternions

137 \mathbb{C} Complex Numbers $6.67384 \times 10^{-11} N \cdot m^2/kg^2$

138 \mathbb{R} Real Numbers

139 Subscripts

140 ρ Friction Index m

141 V Constant Volume m

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