EXPERIMENTS AND MODELING OF ESP PERFORMANCE WITH VISCIOUS OILS AND OIL-WATER EMULSIONS

by
Jianlin Peng

A thesis submitted in partial fulfillment of the requirements for the degree of Master of Science in the Discipline of Petroleum Engineering

The Graduate School
The University of Tulsa

2020
EXPERIMENTS AND MODELING OF ESP PERFORMANCE WITH VISCOUS OILS AND OIL-WATER EMULSIONS

by

Jianlin Peng

A THESIS

APPROVED FOR THE DISCIPLINE OF

PETROLEUM ENGINEERING

By Thesis Committee

Hong-Quan Zhang, Chair
Ovadia Shoham
Baojun Song
Haiwen Zhu
COPYRIGHT STATEMENT

Copyright © 2020 by Jianlin Peng

All rights reserved. No part of this publication may be reproduced, stored in a retrieval system, or transmitted, in any form or by any means (electronic, mechanical, photocopying, recording, or otherwise) without the prior written permission of the author.
ABSTRACT

Jianlin Peng (Master of Science in Petroleum Engineering)
Experiments and Modeling of ESP Performance with Viscous Oils and Oil-Water Emulsions
Directed by Dr. Hong-Quan Zhang
82, pp., Chapter 4: Conclusions and Recommendations
(238 words)

A General Electric (GE) TE-2700 14-stage radial type electrical submersible pump (ESP) was tested with a 3-inch closed flow loop under different viscous oil flow conditions. A pipe-in-pipe heat exchanger was used to cool down the temperature. The ISO-VG320 oil was used as a working fluid for single-phase oil tests. Tap water was added to create oil-water emulsions. At water fraction: 0 and 5%, and different rotational speeds (1800 rpm, 2400 rpm, 3000 rpm, and 3500 rpm), flowrates, pump head, and temperature were recorded, and the ESP performance was characterized. Mass flowrate and density were monitored using the mass flowmeter. Fluid samples were collected during the tests, and viscosity-temperature relationships for both single-phase oil, and oil-water emulsions were measured using a rotational rheometer. The ESP performance declined with the increase of fluid viscosity.

A mechanistic model was developed based on the Euler theory to predict ESP hydraulic performance. The head losses, including friction loss, turn loss, leakage loss, and recirculation loss,
were subtracted. The friction factor correlation in the ESP performance model was modified. Compared with the original mechanistic model, predictions by the improved model show a better agreement with the experimental data at low flow rates. Emulsion rheology was modeled by considering the effects of droplet size, friction, shear, and stage number with corresponding dimensionless numbers. Results agree well with the experimental data, but additional data are required to verify the model generality in the future study.
ACKNOWLEDGMENTS

I would like to thank Dr. Hong-Quan Zhang, my advisor, for having me as his master’s student and as a Research Assistant of Tulsa University Artificial Lift Projects (TUALP). Without his guidance, understanding, and support, I would not have accomplished my master’s study. His wholehearted devotion to and meticulous attitude towards academic research inspired me to become a qualified researcher in the past two and a half years as well as in the future.

I would like to express my gratitude to the TUALP team for their resourcefulness and help. I thank Project Engineer Mr. Bryan Sams, for his technical advice and prompt actions to take care of my experimental facilities. I also thank Ms. Donna Trankley for her professional administrative assistance and management. I would give special thanks to Dr. Haiwen Zhu for his help in facility repair, experiment, and thesis revision. I would also give many thanks to Dr. Jianjun Zhu for his advice on the modification of the mechanistic model. I am also grateful to Dr. Yi Shi for helping me improve the flow loop and collect experimental data.

My sincere appreciation goes to Dr. Ovadia Shoham and Dr. Baojun Song for being my thesis committee members.

Last, I would like to dedicate this work to my parents, Fei Peng, and Bin Huang. Their unlimited love and support always give me strength during my academic pursuit.
TABLE OF CONTENTS

COPYRIGHT STATEMENT ........................................................................................................ iii

ABSTRACT ................................................................................................................................. iv

ACKNOWLEDGMENTS .............................................................................................................. vi

TABLE OF CONTENTS .......................................................................................................... vii

LIST OF FIGURES ................................................................................................................... x

LIST OF TABLES ..................................................................................................................... xiv

INTRODUCTION ...................................................................................................................... 1

CHAPTER 1: LITERATURE REVIEW ....................................................................................... 3

1.1 Ippen ................................................................................................................................. 3
1.2 Stepanoff .......................................................................................................................... 4
1.3 Hydraulic Institute .......................................................................................................... 5
1.4 Amaral et al. .................................................................................................................... 5
1.5 Morrison et al. ................................................................................................................ 6
1.6 Phan et al. ....................................................................................................................... 7

CHAPTER 2: EXPERIMENTAL SETUP AND RESULTS ....................................................... 9
4.2 Recommendations ................................................................. 52

NOMENCLATURE ........................................................................... 54

BIBLIOGRAPHY ............................................................................ 60

APPENDIX A: EQUIPMENT AND INSTRUMENT SPECIFICATIONS ............. 68

APPENDIX B: CHANNEL DISTRIBUTIONS OF NI MODULES ..................... 78

APPENDIX C: PNEUMATIC CONTROL VALVE FLUCTUATION AND REPAIR ...... 80
LIST OF FIGURES

Figure 1.1 Empirical Pump Head Curve - Mixed Flow Type Pump................................. 7
Figure 1.2 Schematic of Pump Facility of Phan et al. (2017)........................................... 8
Figure 2.1 Schematic of TUALP High-Viscosity ESP Flow Loop........................................ 10
Figure 2.2 TUALP Gas-Liquid ESP Flow Loop................................................................. 10
Figure 2.3 TUALP High-Viscosity ESP Flow Loop............................................................ 11
Figure 2.4 TUALP High-Viscosity Flow Loop Discharge Port............................................ 11
Figure 2.5 TE-2700 ESP of TUALP High Viscosity Flow Loop........................................ 13
Figure 2.6 Catalog Pressure and Efficiency Curve from the Manufacturer........................ 14
Figure 2.7 Data Acquisition System of TUALP High-Viscosity ESP Flow Loop................ 15
Figure 2.8 Liquid Flow Rate Control of TUALP High-Viscosity ESP Flow Loop............... 15
Figure 2.9 ISO-VG320 Viscosity versus Temperature....................................................... 16
Figure 2.10 ISO-VG320 Shear Rate Tests at Different Temperatures................................. 17
Figure 2.11 5% Water Cut and Pure Oil Viscosity versus Temperature............................ 17
Figure 2.12 Shear Stress Test Comparison for 5% WC Emulsion and Pure Oil............... 18
Figure 2.13 Comparison of Current, Previous Data and Catalog Curve............................. 21
Figure 2.14 TE-2700 ESP Performance with ISO-VG320 Oil at 1,800 rpm ...................... 22
Figure 2.15 TE-2700 ESP Performance with ISO-VG320 Oil at 2,400 rpm ...................... 22
Figure 2.16  TE-2700 ESP Performance with ISO-VG320 Oil at 3,000 rpm ........................................ 23
Figure 2.17  TE-2700 ESP Performance with ISO-VG320 Oil at 3,500 rpm ............................... 23
Figure 2.18  TE-2700 ESP Performance at Oil Viscosity~240 cP ............................................... 24
Figure 2.19  TE-2700 ESP Performance with 5% WC Emulsion at 3,000 rpm .......................... 25
Figure 2.20  Samples of Work Fluids (a) ISO-VG320 Oil (b) 5% Water/Oil Emulsion .......... 26
Figure 2.21  Water Droplets in ISO-VG320 Oil under Microscope ............................................. 26
Figure 3.1  Velocity Triangles at Impeller Inlet and Outlet .......................................................... 30
Figure 3.2  Velocity Triangles without Inlet Rotation ................................................................. 33
Figure 3.3  Velocity Triangles at Impeller Outlet for $Q_{QLK} < Q_{BMP}$ ...................................... 34
Figure 3.4  Recirculation Effect in Impeller .................................................................................. 35
Figure 3.5  Velocity Triangles at Impeller Outlet for $Q_{QLK} > Q_{BMP}$ ...................................... 36
Figure 3.6  Balancing of Axial Thrust (Tuzson 2000) ................................................................. 40
Figure 3.7  Leakage Geometries in an ESP Stage ........................................................................ 41
Figure 3.8  Flow Chart of Modified Mechanistic Model .............................................................. 43
Figure 3.9  Comparison between Modified Model and Catalog ................................................. 45
Figure 3.10  Error between the Predicted Results and Catalog Curve ....................................... 45
Figure 3.11  TE-2700 ESP Comparison of Modified Model and Experimental Data at 1,800 rpm ........................................................................................................................................ 46
Figure 3.12  TE-2700 ESP Comparison of Modified Model and Experimental Data at 2,400 rpm ........................................................................................................................................ 47
Figure 3.13  TE-2700 ESP Comparison of Modified Model and Experimental Data at 3,000 rpm
Figure 3.14  TE-2700 ESP Comparison of Modified Model and Experimental Data at 3,500 rpm

Figure 3.15  TE-2700 ESP Comparison of Modified Model and Experimental Data at 3,000 rpm for 5% WC Emulsion

Figure 3.16  TE-2700 ESP Comparison of Modified Model and Previous Version against Experiment Data at 1,800 rpm

Figure 3.17  TE-2700 ESP Comparison of Modified Model and Previous Version against Experiment Data at 2,400 rpm

Figure 3.18  TE-2700 ESP Comparison of Modified Model and Previous Version against Experiment Data at 3,500 rpm

Figure A.1  Pressurization Port and Gas Discharge Valve

Figure A.2  Oil Injection Port

Figure A.3  Coriolis Flow Meter

Figure A.4  Temperature Sensor

Figure A.5  Temperature Sensor

Figure A.6  Data Acquisition Device

Figure A.7  Pressure Monitors

Figure A.8  Pneumatic Control Valve

Figure A.9  Cooling System

Figure A.10  Pipe-in-Pipe Heat Exchanger
Figure A.11  Fluid Flow Schematic inside the Heat Exchanger ................................................................. 76

Figure A.12  Rotational Viscometer ........................................................................................................ 77

Figure A.13  Water bath Temperature Control and Circulator ............................................................... 77

Figure C.1  The variation of volume flow rate when the valve is 0% closed ......................................... 82

Figure C.2  The variation of volume flow rate when the valve is 90% closed ....................................... 82

Figure C.3  The variation of volume flow rate when the valve is 0% and 60% closed ...................... 83

Figure C.4  The variation of volume flow rate when the valve is 0% and 80% closed ................... 83

Figure C.5  The variation of volume flow rate when the valve is 0% and 90% closed ................ 84

Figure C.6  Loop with Pneumatic Valve Uninstalled ........................................................................ 85
LIST OF TABLES

Table 1.1  Values of Morrison Number for Different Pump Types .............................................. 6
Table 2.1  Test Matrix .................................................................................................................. 20
Table 3.1  TE-2700 ESP Specifications ......................................................................................... 44
Table A.1  TUALP High-Viscosity ESP Flow Loop Equipment Specifications ......................... 68
Table A.2  TUALP High-Viscosity ESP Flow Loop Instrumentation Specifications ................. 69
Table A.3  TUALP High-Viscosity ESP Flow Loop DAQ Specifications .................................... 69
Table A.4  Coriolis Flow Meters Specifications ......................................................................... 71
Table A.5  Pipe-in-Pipe Heat Exchanger Design Data ................................................................. 71
Table B.1  cFP-AI-110 Module #1 Channel Dsicturbutions ...................................................... 78
Table B.2  cFP-AI-111 Module #2 Channel Dsicturbutions ...................................................... 78
Table B.3  cFP-AI-111 Module #3 Channel Dsicturbutions ...................................................... 79
Table B.4  cFP-AO-200 Module #4 Channel Dsicturbutions ..................................................... 79
Table C.1  Errors in the maximum/minimum volume flow rate and the average volume flow rate ............................................................................................................. 84
INTRODUCTION

The electrical submersible pump (ESP) is one of the most widely used artificial lift methods in the oil industry. Compared with other artificial lift methods, ESP is more suitable for high flow rates, and it is being adopted in offshore production systems. It is used to overcome the pressure loss and lift very high liquid flow rate. A stack of centrifugal pump stages is connected by a central shaft in the ESP. Each stage has a rotational impeller and a stationary diffuser. The impeller is locked by a key to the shaft, which is rotated by a submerged motor. As a result, the liquid is accelerated and then guided by a diffuser. This way, the electric energy is transformed into hydraulic pressure energy. Only the water performance curves (head, horsepower, and efficiency) are provided by the manufacturers. However, ESP performance is affected by fluid properties, including but not limited to viscosity, density, gas fraction, and interfacial surface tension. Therefore, an accurate prediction of the ESP performance is required to optimize the production system design and operation.

Viscous effects on ESP performance has been analyzed by many researchers using different methodologies. Some researchers conducted experiments on different ESPs with varying flow conditions. Analyzing the viscous effect on a limited number of ESPs is a simple task and can characterize the pump performance within the tested ranges. However, it is difficult to predict the performance of every ESP in the market because of wide viscosity and flow ranges. Some
researchers proposed empirical correlations based on experiments to avoid analyzing the complicated flow structures inside the ESP. These correlations lose accuracy in different pumps and flow conditions. CFD simulation can help understand the fluid behavior inside a pump stage, but it is very time-consuming, and the results are sometimes unreliable. Therefore, a mechanistic model, which is reliable and easy to be used, is derived in this study based on physical principles.

In the following, Chapter 1 is the literature review. Chapter 2 demonstrates the experiment facility, procedures, test matrix, and results. Chapter 3 presents the development, modification, and validation of the mechanistic model. Finally, Chapter 4 presents conclusions and recommendations.
CHAPTER 1

LITERATURE REVIEW

1.1 Ippen

Ippen (1946) directed more than 220 performance tests for oil viscosities up to 10,000 Saybolt universal second (SUS), which is around 1900 cP, with four distinctive single-stage centrifugal pumps. The pump head of different oil viscosities was measured under different rotational speeds. It is the first study of viscosity effects on pump performance in the laboratory. Ippen defined Reynolds number as

\[ R_D = 2620 \frac{N d_2^2}{\nu \cdot 10^5} \]  \hspace{1cm} (1.1)

where \( N \) is the rotational speed of the impeller in rpm, \( d_2 \) is the impeller diameter in ft, and \( \nu \) is the kinematic viscosity in cSt (centistokes).

The ratio of oil head to water head \( \frac{H_o}{H_w} \), the efficiency loss \( \frac{100 - e}{100} \), and the ratio of oil power input to the water power input \( \frac{BHP_o}{S_0 \cdot BHP_w} \) is corrected by oil specific gravity and plotted against \( R_D \). Pumps with \( R_D \) lower than 1,000 cannot be plotted using this method.
1.2 Stepanoff

Stepanoff (1949) conducted ESP experiments with fluid viscosities from 1 to 2,000 cSt, which is around 1800 cP. He introduced the impeller specific speed, $N_s$, to describe the pump characteristics under different types of fluid. $N_s$ is defined as

$$N_s = \frac{\sqrt{q_{BEP}N}}{H_{BEP}^{0.75}g^{0.75}}$$

(1.2)

where $q_{BEP}$ is pump capacity at BEP in GPM, $N$ is the impeller rotational speed in rpm, $H_{BEP}$ is the pump head at BEP in ft, and $g$ is the gravitational acceleration in $\text{ft/s}^2$. Thus, a relation can be written as

$$\frac{q_{vis}^{vis}}{q_{water}^{water}} = \left(\frac{H_{vis}^{vis}}{H_{water}^{water}}\right)^{1.5}$$

(1.3)

The left side of Equation (1.3) can be defined as the flow rate correction factor, $F_Q$, and the right side of Equation (1.3) can be identified as the head correction factor.

Stepanoff defined a Stepanoff Reynolds Number:

$$Re_{Stepanoff} = 6.0345 \frac{Nq_{vis}^{vis}}{\sqrt{H_{water}^{water}v}}$$

(1.4)

where $N$ is the rotational speed in rpm, $q_{vis}^{vis}$ is the viscous fluid flow rate at BEP in bpd, $H_{water}^{water}$ is the water head at BEP in ft, and $v$ is the liquid kinematic viscosity in cSt.

To obtain the hydraulic efficiency, the pump head correction factor and flow rate correction factor, as well as the Stepanoff Reynolds Number, were plotted. An iterative method was applied with an initial guess of the viscous fluid flow rate to obtain the pump performance from the plots.
1.3 Hydraulic Institute

An empirical correlation based on the catalog water curve was proposed by Hydraulic Institute (1955) for the centrifugal pump performance under viscous flow. Three correction factors are defined as

\[ C_Q = \frac{q_{vis}}{q_w}, \]  
\[ C_H = \frac{H_{vis}}{H_w}, \]  
and \[ C_\eta = \frac{\eta_{vis}}{\eta_w}. \]

where \( C_Q, C_H, \) and \( C_\eta \) are the correction factors of flowrate, head, and efficiency, \( q_{vis} \) and \( q_w, H_{vis} \) and \( H_w, \) and \( \eta_{vis} \) and \( \eta_w \) are the flowrates, heads, and efficiencies of viscous fluid and water flow, respectively.

The correction curves can be found in the charts provided by Hydraulic Institute in a broad flow rate range. However, stand-alone experiments are recommended if high accuracy is required.

1.4 Amaral et al.

Amaral et al. (2007) conducted experiments on a semi-axial ESP and a conventional radial pump. Glycerin and water with a viscosity range from 1020 to 67 cP by changing temperatures from 20 to 60°C were tested. According to their results, the pump affinity law can roughly capture the viscosity effect on pump performance. Compared with the experimental test curves, the accuracy of the predicted curve decreases with the increase of oil viscosity.
1.5 Morrison et al.

The conventional pump affinity laws were modified by Morrison et al. (2017) to consider the oil viscosity effect. The flow behavior of a mixed type pump with different fluid viscosities ranging from 2.4 to 400 cP was investigated by using the Computational Fluid Dynamics (CFD) method. The modified flow coefficient ($\varphi$) and head coefficient ($\Psi'$) along with Timer’s rotational Reynolds number ($Re_w$) are defined as:

$$
\varphi = \frac{Q}{\omega D_s^2},
$$

(1.8)

$$
\psi = \frac{\Delta p}{\rho D_s^2 \omega^2},
$$

(1.9)

and

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

(1.10)

These three dimensionless parameters follow a single universal curve:

$$
\Psi = C_1 \varphi Re_w^{-Mo} + C_2
$$

(1.11)

where $C_1$ and $C_2$ are empirical constants, and $Mo$ is the Morrison number which is related to the pump specific speed. The Morrison numbers of different pump types are listed in Table 1.1.

<table>
<thead>
<tr>
<th>Pump Type</th>
<th>$N_s$</th>
<th>Mo</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mixed flow</td>
<td>2758</td>
<td>0.066</td>
</tr>
<tr>
<td>Split vanes</td>
<td>3027</td>
<td>0.072</td>
</tr>
<tr>
<td>Semi-axial</td>
<td>3817</td>
<td>0.2</td>
</tr>
<tr>
<td>Helical axial</td>
<td>5284</td>
<td>0.05</td>
</tr>
</tbody>
</table>

As shown in Figure 1.1, the modified affinity law agrees well with the experimental tests. Since the Morrison number changes a lot for different types of pumps, further investigations are necessary to verify its accuracy on other pumps.
Phan et al. (2017) presented the effects of high viscosity oil on the performance of a single-stage centrifugal pump. The Conoco R&O Multipurpose 220 oil was used as a working fluid at three temperatures: 43°C (155 cP), 46°C (134 cP), and 49°C (115 cP), and three rotational speed: 3600, 3300, and 3000 rpm. The working fluid was stored in a tank, which has a heater at the bottom, to eliminate residual gas in the flow loop. The schematic of the pump facility is shown in Figure 1.2. According to their results, the homogeneous model is unable to predict head performance with acceptable accuracy.
Figure 1.2 Schematic of Pump Facility of Phan et al. (2017)
CHAPTER 2

EXPERIMENTAL SETUP AND RESULTS

The experimental facility, testing procedure, data acquisition system (DAQ), and experimental results for oil and oil-water emulsions are presented in this chapter.

2.1 Experimental Facility

The schematic of the experimental facility is shown in Figure 2.1. As shown in Figure 2.2, the previous TE-2700 gas-liquid flow loop by Zhu et al. (2017a and 2017b) is upgraded for high viscosity and oil-water emulsion tests. Zhang (2017) disconnected the air injection line and the gas-liquid separator and added a pipe viscometer, as shown in Figure 2.2. ESP performance was tested with water and ND20 oil with viscosity from 1 cp to 107 cp. To discharge gas trapped inside the loop more effectively, a ball valve was connected to the gas discharge port in Figure 2.4 (a). Later on, it was replaced by a pipe, as shown in Figure 2.4 (b), to change the discharge direction downwards to protect students from breathing in the hazardous oil mist when releasing the pressure. The flow loop has a capacity of 46.3 gallons and a maximum designed flow rate of 6,000 bpd. The detailed specifications and configurations are listed in Appendix A.
Figure 2.1 Schematic of TUALP High-Viscosity ESP Flow Loop

Figure 2.2 TUALP Gas-Liquid ESP Flow Loop
Figure 2.3 TUALP High-Viscosity ESP Flow Loop

Figure 2.4 TUALP High-Viscosity Flow Loop Discharge Port

with (a) Ball Valve (b) Elbow Pipe
2.1.1 Viscous Fluid Flow Loop

In this study, ESP TE2700, which was driven by a motor (North American H3650), was tested in a 3-inch stainless steel closed loop. The rotational speed of the motor was controlled by a variable speed drive (Hitachi L300P). The pressure increments were measured using four differential pressure transducers (Rosemount 3051S). The pressure was monitored by six absolute pressure transmitters (Rosemount 2051). Three-wire platinum resistance temperature detectors (RTDs) were used to measure the temperature at the ESP intake and the ESP output, and one J-thermocouple was used to record the temperature 15-ft downstream the ESP where the fluid flow was fully developed (Zhu 2019). One pneumatic control valve and one manual control valve were installed to control the liquid flowrate. Mass flowrate and density were monitored by a Coriolis flowmeter (Proline Promass 80E).

The flow loop was pressurized before the high viscosity oil test by a compressor (Kaeser CSD60) and controlled by an air pressure regulator to avoid cavitation. In the gas discharge section, a hollow cylinder (length: 7.75-in, diameter: 2.75-in) was submerged in the 3-in pipeline as a float to prevent the reverse gas entrainment.

The heat exchanger, in which ice water was pumped by a Dayton Stainless Steel Centrifugal Pump 2ZWT9A, was initially designed to control the loop temperature. However, it was hard to maintain a stable temperature due to the thickness of the pipe. Therefore, it was mainly used to cool down the flow loop after each high viscosity test.
2.1.2 ESP for Experiment

The TE-2700 used in the experiment is a 14-stage radial type ESP manufactured by General Electric (GE) with a BEP at 2,700 bpd and 3,500 rpm. The pump bench is shown in Figure 2.5. The pressure ports were drilled at stages 2–14. The differential pressure of each stage was measured, except for stage 1. The absolute pressures at pump inlet and outlet, as well as the pressure at each stage from stages 5-12, were measured to double-check the differential pressure measurements (Zhu et al. 2018a, Zhu et al. 2019a, Zhu et al. 2019b). The rotational speed and torque were measured by a torque sensor. However, only part of the torque data was collected in this study since the torque monitor lost its accuracy in the test. The pump catalog curve from the manufacturer is shown in Figure 2.6.

Figure 2.5 TE-2700 ESP of TUALP High Viscosity Flow Loop
2.1.3 Data Acquisition System

The data acquisition system was built by National Instrument (NI) modules. NI input modules (cFP-AI-111) were used to collect output analog signals (4–20 mA) from temperature and pressure transmitters, as well as the Coriolis flowmeter. Temperature transmitters (INOR IPAQ R520) were used to convert RTD and thermocouples’ output signals to 4–20 mA current signals. A NI output module (cFP-AO-200) was used in this study to generate input signals to control the pneumatic control valve and VSD (Zhu et al. 2019c). All modules were connected to a NI Ethernet interface (cFP-1804) and transferred to a data processing computer. The analog signals were scaled up to the engineering unit by NI LabVIEW for data processing.
As shown in Figure 2.7, the DAQ program was written in a graphic-programming language LabVIEW 2014 for acquiring data and controlling the flow rate (Zhu et al. 2018b). The flow rate was controlled by adjusting the closing percentage of the pneumatic control valve, as shown in Figure 2.8. When the pneumatic control valve was not available, the flow rate is controlled by a manual control valve.

Figure 2.7 Data Acquisition System of TUALP High-Viscosity ESP Flow Loop

Figure 2.8 Liquid Flow Rate Control of TUALP High-Viscosity ESP Flow Loop
2.2 Experimental Program

2.2.1 Testing Fluids

Tap water and lubricating oil were used as work fluids in this study. The oil ISO-VG320 used in this study has a higher viscosity range (50 cP to 1600 cP, as shown in Figure 2.9) than that of ND20 in the previous study. Lab tests with different shear rates at 30°C, 40°C, and 70°C, as shown in Figure 2.10, indicate that this oil behaves as a Newtonian fluid.

![ISO-VG320 Viscosity versus Temperature](image)

Figure 2.9 ISO-VG320 Viscosity versus Temperature
Figure 2.10 ISO-VG320 Shear Rate Tests at Different Temperatures

The emulsion was sampled from the loop and examined closely under a microscope. Figure 2.11 and Figure 2.12 show that the 5% water cut fluid has a similar property as the ISO-VG320 oil.

Figure 2.11 5% Water Cut and Pure Oil Viscosity versus Temperature
2.2.2 Experimental Procedure

2.2.2.1 Single-Phase Fluid Testing Procedure.

Before the high viscosity test, the loop needs to be filled with the working fluid, and the gas bubbles in the fluid need to be removed. Therefore, the flow loop was firstly filled by working fluid through the input port until the liquid level reached the upper part of the transparent input port, while the gas discharge valve and the injection valve were open. Then, the discharge valve and injection valve were closed, and the loop was pressurized 30 psig to avoid cavitation at the pump inlet (Zhu et al. 2019d, Zhu et al. 2018c). The pump was started at a low rotational speed for a short period until gas bubbles trapped in the liquid were separated in the discharge pipe. After the pump was stopped, the loop pressure was released to vent the separated gas in the discharge pipe. Previous steps were repeated until the fluid level in the transparent liquid input port was
stable, and the loop was ready for a high viscosity test.

After gas bubbles were removed by the steps as mentioned above, the flow loop was repressurized to 50 psig, and the rotational speed was increased to the designated point. The pump curve was recorded when the temperature was in the designed range. Pump rotational speed, temperatures, pressures, differential pressures, and flow rate were recorded every second. At each flow rate, fifty samples were collected to minimize the uncertainty. When the loop temperature was unable to be maintained by the heat exchanger, the pump was stopped, and the loop was cooled down by the heat exchanger. The pump may need to be stopped multiple times until a complete pump curve was accomplished (Zhu et al. 2018d).

2.2.2.2 Oil-Water Emulsion Testing Procedure

Emulsion experiments are very similar to the single-phase oil testing. The total loop volume, which was measured before the test, is 46.3 gallons. The water and oil volume were calculated accordingly. A stable emulsion mixture can be created by the tested ESP in a few minutes. When the readings in the mass flow meter were stable without large fluctuations, the pump curve was recorded similarly as for the single-phase oil tests.

2.2.3 Test Matrix

The test matrix is listed in Table 2.1. The experiments were interrupted by the winter conditions, followed by the COVID-19. Therefore, only the high viscosity oil test and the 5%
water-oil emulsion test were completed.

Table 2.1 Test Matrix

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Water Fraction (%)</th>
<th>ESP Rotational Speed (rpm)</th>
<th>Choke Opening (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tap Water</td>
<td>100</td>
<td>2400</td>
<td></td>
</tr>
<tr>
<td>ISO-VG320</td>
<td>0</td>
<td>1800, 2400, 3000, 3500</td>
<td>100, 50, 40, 30, 20, 15, 10, 9, 8, 7, 6, 5</td>
</tr>
<tr>
<td></td>
<td>5, 10, 15, 20, 25, 30, 35, 40, 45, 50</td>
<td>3500</td>
<td></td>
</tr>
</tbody>
</table>

2.3 Experimental Results

2.3.1 Sing-Phase Liquid Results

2.3.1.1 Water Performance Curve

TE-2700 ESP was tested with tap water at the rotational speed of 2,400 rpm. The pump curve is compared with the catalog curve and previous tests by Zhang (2017) in Figure 2.13. The tested pump curve agrees well with the catalog curve, which validates the experimental setup of this study.
2.3.1.2 Oil Performance Curve

TE-2700 ESP was tested with ISO-VG320 oil at four pump rotational speeds of 1,800 rpm, 2,400 rpm, 3,000 rpm, and 3,500 rpm. The viscosity of the original ISO-VG320 sample was comparable to the samples collected from the loop, as shown in Section 2.2.1. Then, the viscosity can be obtained from the curves measured by the rotational viscometer (Anton Paar RheolabQC).

The head curves at the rotational speeds of 1,800 rpm, 2,400 rpm, 3,000 rpm, and 3,500 rpm are shown in Figure 2.14, Figure 2.15, Figure 2.16, and Figure 2.17, respectively. Their trends are similar, and the pump head decreases with the increase of fluid viscosity.
Figure 2.14 TE-2700 ESP Performance with ISO-VG320 Oil at 1,800 rpm

Figure 2.15 TE-2700 ESP Performance with ISO-VG320 Oil at 2,400 rpm
Figure 2.16 TE-2700 ESP Performance with ISO-VG320 Oil at 3,000 rpm

Figure 2.17 TE-2700 ESP Performance with ISO-VG320 Oil at 3,500 rpm

Figure 2.18 shows the comparison of pump head at different rotational speeds (1,800 rpm, 2,400 rpm, and 3,500 rpm) while the oil viscosities are similar (240 cP). Compared with high flow
rates, head losses due to friction, turning, and recirculation at low flow rates are less significant. However, the head loss of an ESP becomes more significant at a higher flow rate.

![Figure 2.18 TE-2700 ESP Performance at Oil Viscosity≈240 cP](image)

**Figure 2.18 TE-2700 ESP Performance at Oil Viscosity≈240 cP**

2.3.2 Oil-Water Emulsion Results

2.3.2.1 5% Water Cut Emulsion

The head curves of different viscosities at the rotational speed of 3,000 rpm are shown in Figure 2.19. The trends are similar to those of single-phase oil results.
2.4 Sampling

To calculate the oil and water fractions and measure working fluid viscosity, the oil sample was taken after each experiment. The sample was collected at the oil discharge port of the loop. A beaker was used to record the volume of the sample. Figure 2.20 (a) shows the pure ISO-VG320 oil sample, and Figure 2.20 (b) shows the 5% water cut emulsion sample. Small fractions from the emulsion sample were taken and observed under a microscope, as shown in Figure 2.21. The viscosity of the water-oil emulsion was measured the same way as explained in Section 2.3.1. The oil sample was stored until the gas bubbles ultimately came out from the oil. Then, the volume of the oil was checked to estimate the gas fraction. The water-oil emulsion sample was stored until oil and water were completed separated. Then, the water cut was checked to ensure it matched the designed fraction.
Figure 2.20 Samples of Work Fluids (a) ISO-VG320 Oil (b) 5% Water/Oil Emulsion

Figure 2.21 Water Droplets in ISO-VG320 Oil under Microscope
CHAPTER 3

ESP PERFORMANCE MODELING AND RESULTS

This chapter presents the modification of the TUALP mechanistic ESP performance prediction model for viscous and emulsion fluids. The results were compared to the previous version and experimental data. The model was improved by considering the emulsion rheology as well as modifying the correlation factors for friction and the theoretical head.

3.1 Emulsion Rheology Model

According to Kokal (2005), emulsion rheology is determined by densities, viscosities, volume fraction and interfacial tension of phases, droplet characteristics, and shear rate. The model was incorporated by Zhu et al. (2018e, 2019e, 2019f, 2019g) to predict ESP performance.

Since the continuous phase inversion point is essential to predict the effective viscosity, the water fraction corresponding to this point is determined using the Brinkman (1952) model:

$$\mu_E = \frac{\mu_C}{\left(1 - \phi_D\right)^E} \quad (3.1)$$

where $\mu_C$ is continuous phase viscosity, $\phi_D$ is the volume fraction of the dispersed phase, and $E$ is a modified exponent acquired from experiments. The water fraction at the inversion point can be expressed as:

$$\phi_W = \frac{1}{1 + \tilde{\mu}^{1/E}} \quad (3.2)$$
where \( \tilde{\mu} = \frac{\mu_0}{\mu_w} \).

The droplet size distribution has a tremendous influence on the rheology of the emulsion. Since the distribution is hard to be obtained, the Weber number is used to estimate the mean droplet diameter:

\[
We = \frac{\rho_A v^2 l}{\sigma} \approx \frac{\rho_A Q^2}{\sigma V}
\]

(3.3)

where \( \rho_A \) is the average density, \( Q \) is the flow rate, \( V \) is the pump channel volume of one stage, and \( \sigma \) is the interfacial tension obtained from experiments.

Turbulence effect is taken into consideration by introducing a representative Reynolds number:

\[
Re = \frac{\rho_A v l}{\mu_A} \approx \frac{\rho_A Q}{\mu_A d}
\]

(3.4)

where \( \mu_A \) is the modeled emulsion viscosity, \( d \) is pump diameter.

Changing shear rate by changing pump rotational speed can lead to the change of effective viscosity of emulsion. The rotational speed is considered by using Strouhal number:

\[
St \approx \frac{f v}{Q}
\]

(3.5)

where \( f \) is the ESP rotational speed.

The final emulsion viscosity can be expressed with the factors above:

\[
\mu_A = C(\mu_E - \mu_M) + \mu_M
\]

(3.6)

where \( \mu_M \) is the mixture base viscosity defined as:

\[
\mu_M = \frac{\mu_w}{(1 - \phi_O \phi_{OE})^E}
\]

(3.7)

where \( \phi_{OE} = 1 - \left( \frac{\mu_w}{\mu_O} \right)^{1/E} \).

\( C \) is a coefficient representing the effects of droplet size, turbulence, shear, and stage number in
emulsion rheology:

\[ C = \frac{(N\text{WeRe})^n}{bSt^m} \]  

(3.8)

where \( N \) is stage number from the ESP inlet. The exponents \( m \) and \( n \) need to be determined by experiments. According to Hattan (2018), the model can be extended to other types of pumps.

### 3.2 ESP Performance Model

The mechanistic model is developed based on Euler’s equation for centrifugal pump. Head losses, including recirculation loss, friction loss, turn loss, leakage loss, diffuser loss, and disk loss, are considered (Zhu et al. 2019h, Zhu et al. 2019i). Although the previous model can predict ESP performance with high accuracy, special considerations on low flow rates, high viscosities, and low rotational speeds are still required to improve the model.

#### 3.2.1 Euler’s Equation for Centrifugal Pumps

Euler’s Equation is applied to a two-dimensional system that filled with single-phase, incompressible ideal fluid. The velocities of the pump inlet and outlet are shown in Figure 3.1.
The external torque acting on the impeller for the fluid to flow can be derived from Newton’s second law of motion:

$$\tau = \dot{m}(R_2 C_{2U} - R_1 C_{1U}) \quad (3.9)$$

where $R_1$ is the radius of the inlet and $R_2$ is the radius of the outlet, $C_{1U}$ is the fluid tangential velocity at the impeller inlet and $C_{2U}$ is the fluid tangential velocity at the impeller outlet.

The tangential velocity of the impeller at the inlet and the outlet can be expressed as:

$$U_1 = R_1 \Omega \quad (3.10)$$

$$U_2 = R_2 \Omega \quad (3.11)$$

where $\Omega$ is the angular velocity of the impeller to be obtained from pump rotational speed, $N$:

$$\Omega = \frac{2\pi N}{60} \quad (3.12)$$
Then, the shaft power can be calculated by adopting the equations above as:

\[ P_2 = \rho Q (U_2 C_{2U} - U_1 C_{1U}) \]  \hspace{1cm} (3.13)

The hydraulic power, \( P_{hyd} \), can be written as:

\[ P_{hyd} = H_E \rho g Q \]  \hspace{1cm} (3.14)

Assuming the flow has no losses, Equation (3.13) and Equation (3.14) are equivalent. Then, the Euler’s Equation for the centrifugal pump can be expressed as:

\[ H_E = \frac{u_2 C_{2U} - u_1 C_{1U}}{g} \]  \hspace{1cm} (3.15)

Euler’s head can also be rewritten based on the velocity trigonometry:

\[ H_E = \frac{u_2^2 - u_1^2}{2g} + \frac{w_2^2 - w_1^2}{2g} + \frac{c_2^2 - c_1^2}{2g} \]  \hspace{1cm} (3.16)

where the first term on the right side of Equation (3.16) is the static head as a result of centrifugal forces, the second term is the static head as a result of velocity change, and the third term is the dynamic head.

The fluid absolute velocity is defined as the meridional velocity. The meridional velocity at the impeller inlet is expressed as:

\[ C_{1M} = \frac{Q + Q_{LK}}{(2\pi R_1 - Z_1 T_B) y_{11}} \]  \hspace{1cm} (3.17)

where \( Q \) is the flow rate, \( Q_{LK} \) is the leakage flow rate, \( Z_1 \) is the impeller blade number, \( T_B \) is the blade thickness projected to the radial direction, and \( y_{11} \) is the impeller inlet height.

The meridional velocity at the impeller outlet is expressed as:

\[ C_{2M} = \frac{Q + Q_{LK}}{(2\pi R_2 - Z_1 T_B) y_{12}} \]  \hspace{1cm} (3.18)

where \( y_{12} \) is the impeller outlet height.

The relative velocity at the impeller inlet is expressed as:
\[ W_1 = \frac{C_{1M}}{\sin \beta_1} \] (3.19)

where \( \beta_1 \) is the blade angle from tangential at impeller inlet.

The relative velocity at the impeller outlet is expressed as:

\[ W_2 = \frac{C_{2M}}{\sin \beta_2} \] (3.20)

where \( \beta_2 \) is the blade angle from tangential at impeller outlet.

The fluid absolute velocity at the impeller inlet is then expressed as:

\[ C_1 = \sqrt{C_{1M}^2 + \left( U_1 - \frac{C_{1M}}{\tan \beta_1} \right)^2} \] (3.21)

The fluid absolute velocity at the impeller outlet is then expressed as:

\[ C_2 = \sqrt{C_{2M}^2 + \left( U_2 - \frac{C_{2M}}{\tan \beta_2} \right)^2} \] (3.22)

The fluid tangential velocity at the impeller inlet can be expressed as:

\[ C_{1U} = U_1 - W_1 \cos \beta_1 \] (3.23)

The fluid tangential velocity at the impeller outlet can be expressed as:

\[ C_{2U} = U_2 - W_2 \cos \beta_2 \] (3.24)

Equation (3.15) can be rewritten as:

\[ H_E = \frac{U_2(U_2 - W_2 \cos \beta_2) - U_1(U_1 - W_1 \cos \beta_1)}{g} \] (3.25)

Assuming no tangential fluid velocity at the impeller inlet, then \( C_{1U} = 0 \) and \( C_1 = C_{1M} \), as shown in Figure 3.2. Euler’s equation can be expressed as:

\[ H_E = \frac{U_2^2}{g} - \frac{U_2 C_{2M}}{\tan \beta_2} \] (3.26)
3.2 Effective Velocity at Impeller Outlet

When the direction of the fluid absolute velocity and the designated flow direction are the same, the BMP is reached. However, if mismatch happens, an effective velocity is introduced when the flowrate is different from the BMP flow rate (Zhu et al. 2019j, Zhu et al. 2019k). Figure 3.3 shows the velocity triangles when the flow rate is lower than that at the BMP.
Figure 3.3 Velocity Triangles at Impeller Outlet for \( Q+Q_{LK} < Q_{BMP} \)

The fluid flow velocity at the impeller outlet is expressed as:

\[
C_{2F} = C_{2B} \frac{Q}{Q_{BMP}} \tag{3.27}
\]

where \( C_{2B} \) is the absolute fluid velocity at the impeller outlet at the BMP.

The shear velocity, if \( C_{2F} \) is higher than \( C_2 \), can be expressed as:

\[
V_S = U_2 \frac{Q_{BMP}-(Q+Q_{LK})}{Q_{BMP}} \tag{3.28}
\]

The projected velocity, \( C_{2P} \), can be derived as:

\[
C_{2P} = \frac{C_2^2+C_{2F}^2-V_S^2}{2C_{2F}} \tag{3.29}
\]

Fluid circulation takes place in the impeller because of the shear effects, as shown in Figure 3.4. As a result, the kinetic energy is not fully converted to static pressure. Shear velocity, impeller channel size, and fluid viscosity are the dominant factors for recirculation flow. Therefore, the
Reynolds number is introduced to estimate the recirculation head loss:

\[ Re_c = \frac{\rho V_S D_C}{\mu} \]  \hspace{1cm} (3.30)

where \( D_C \) is the representative impeller channel width at the outlet, and can be calculated:

\[ D_C = \frac{2\pi R_2}{Z_I} \sin \beta_2 - T_B \]  \hspace{1cm} (3.31)

![Figure 3.4 Recirculation Effect in Impeller](image)

A correlation developed from experimental data is expressed as:

\[ C_{2E} = C_{2F} + \sigma (C_{2P} - C_{2F}) \]  \hspace{1cm} (3.32)

where \( \sigma \) is the velocity reduction factor, which can also be named the slip factor:

\[ \sigma = \left( \frac{\mu_W}{\mu_O} \right)^{0.1} \frac{1}{1 + 0.02 Re_c^{0.2}} \]  \hspace{1cm} (3.33)

If the pump in-situ flow rate is higher than that at the BMP, as shown in Figure 3.5, the shear velocity can be expressed as:
\[ V_S = U_2 \left( \frac{Q+Q_{LK} - Q_{BMP}}{Q_{BMP}} \right) \]  

\[ (3.34) \]

The effective velocity, \( C_{2E} \), is now expressed as:

\[ C_{2E} = \frac{c_{2E}^2 + c_{2F}^2 - \frac{V^2}{2}}{2c_{2F}^2} \]  

\[ (3.35) \]

The effective Euler head thus can be expressed as:

\[ H_{EE} = H_E + \frac{c_{2E}^2 - c_{2}^2}{2g} \]  

\[ (3.36) \]
3.2.3 Head Losses

3.2.3.1 Recirculation Loss

Based on the derivations of Equation (3.36), the recirculation head loss can be expressed as:

\[ h_{\text{recirculation}} = \frac{c_2^2 - c_{2i}^2}{2g} \quad (3.37) \]

3.2.3.2 Friction Loss

The previous model treated the fluid inside the impeller and the diffuser as channel flow without considering other factors that contribute to the friction loss. Only Churchill (1977) equations, which is developed for circular, straight, stationary pipe, are used to calculate the friction factor:

The Churchill equation for the friction factor can be expressed as:

\[ f = 8 \left[ \left( \frac{8}{A_{\text{Re}}} \right)^{12} + \frac{1}{(A+B)^{1.5}} \right]^{1/12} \quad (3.38) \]

where \( A \) and \( B \) can be expressed as:

\[ A = \left[ 2.457 \ln \left( \frac{1}{\frac{7}{\text{Re}^{0.5}} + 0.27 \frac{\varepsilon}{D}} \right) \right]^{16} \quad (3.39) \]

\[ B = \left( \frac{37.530}{\text{Re}} \right)^{16} \quad (3.40) \]

The Reynolds number can be expressed as:

\[ Re = \frac{d_H Q_L \rho_L}{2 \pi r \mu_L \sin \beta} \quad (3.41) \]

Sun and Prado (2003) proposed a correlation of friction factor in rotating ESPs. In ESP, the
friction factor needs to consider the cross-section shape effect ($F_{\gamma}$), the pipe curvature effect ($F_{\beta}$), and the rotational speed effect ($F_{\omega}$). The cross-section shape effect ($F_{\gamma}$) is expressed as:

$$F_{\gamma} = \left[ \frac{2}{3} + \frac{11}{24} l_L (2 - l_L) \right]^{-1}$$ \hspace{1cm} Re \leq 2300 \hspace{1cm} (3.42)$$

$$F_{\gamma} = \left[ \frac{2}{3} + \frac{11}{24} l_L (2 - l_L) \right]^{-0.25}$$ \hspace{1cm} Re > 2300 \hspace{1cm} (3.43)$$

where $l_L$ is the aspect ratio of the rectangular cross-section for liquid defined as:

$$l_L = \frac{\min(a_L, b_L)}{\max(a_L, b_L)}$$ \hspace{1cm} (3.44)$$

where $a_L$ is the channel width, and $b_L$ is the channel height for the impeller or diffuser.

The critical Reynolds number is introduced to take care of the pipe curvature effect as:

$$N_{Re} = 2 \times 10^4 \times \left( \frac{R_c}{r_H} \right)^{-0.32}$$ \hspace{1cm} \frac{R_c}{r_H} < 860 \hspace{1cm} (3.45)$$

$$N_{Re} = 2300$$ \hspace{1cm} \frac{R_c}{r_H} \geq 860 \hspace{1cm} (3.46)$$

where $R_c$ is the radius of curvature along a channel, $r_H$ is the hydraulic radius based on the hydraulic diameter:

$$R_c = \frac{1}{\sin \beta - \frac{d_H}{r_H} \frac{1}{\frac{r_H}{\tan \beta}}}$$ \hspace{1cm} (3.47)$$

$$r_H = \frac{d_H}{2}$$ \hspace{1cm} (3.48)$$

The pipe curvature effect can then be concluded as:

For laminar effect ($Re < N_{Re}$)
\( F_\beta = 0.266 \text{Re}^{0.389} \left( \frac{R_c}{r_H} \right)^{-0.1945} \)  \hspace{1cm} (3.49)

For turbulent effect \((Re > N_Re)\)

\[
F_\beta = \begin{cases} 
    \left( \frac{\text{Re}}{r_H} \right)^{-2} & \text{Re} \left( \frac{R_c}{r_H} \right)^{-2} \geq 300 \\
    0.092 \left( \frac{\text{Re}}{r_H} \right)^{-2} + 0.25 & 300 \geq \text{Re} \left( \frac{R_c}{r_H} \right)^{-2} > 0.034 \\
    1 & \text{Re} \left( \frac{R_c}{r_H} \right)^{-2} \leq 0.034 
\end{cases} \hspace{1cm} (3.50)
\]

The rotational Reynolds number, \( \text{Re}_\omega \), is introduced to take care of the rotational speed effect defined by:

\[
\text{Re}_\omega = \frac{\omega d_H^2 \rho_l}{\mu_l} \hspace{1cm} (3.51)
\]

The rotational speed effect, \( F_\omega \), can then be expressed as:

\[
F_\beta = \begin{cases} 
    1 & \frac{\text{Re}_\omega}{\text{Re}} < 1 \\
    0.942 + 0.058 \left( \frac{\text{Re}_\omega^2}{\text{Re}} \right)^{0.282} & 1 \leq \frac{\text{Re}_\omega^2}{\text{Re}} < 15 \\
    0.942 \left( \frac{\text{Re}_\omega^2}{\text{Re}} \right)^{0.05} & \frac{\text{Re}_\omega^2}{\text{Re}} \geq 15 
\end{cases} \hspace{1cm} (3.52)
\]

Thus, the friction factor is calculated as:

\[
f_{\gamma \beta \omega} = F_\gamma F_\beta F_\omega f \hspace{1cm} (3.53)
\]

The friction loss can then be calculated by:

\[
h_{\text{friction}} = \frac{f_{\gamma \beta \omega} Q^2}{8gD_H \pi^2 \rho_m^2 \sin^3 \beta_m \frac{r_2 - r_1}{r_1 r_2}} \hspace{1cm} (3.54)
\]

### 3.2.3.3 Turn Loss

Turning loss takes place when the fluid flow direction changes through the impeller and diffuser. The turning head loss in the impeller can be expressed as:

\[
H_{TI} = f_{TI} \frac{V_t^2}{2g} \hspace{1cm} (3.55)
\]
The turning head loss in the diffuser can be expressed as:

\[ H_{TD} = f_{TD} \frac{V_D^2}{2g} \]  

(3.56)

where \( f_{TI} \) and \( f_{TD} \) are the local drag coefficients based on experimental data.

Then, the total turning loss can be expressed as:

\[ h_{\text{shock}} = f_{TI} \frac{V_I^2}{2g} + f_{TD} \frac{V_D^2}{2g} \]  

(3.57)

3.2.3.4 Leakage Loss

In an ESP, the reverse flow goes through the secondary leakage flow passage, including balance holes and the clearance between the impeller and the diffuser, as shown in Figure 3.6. The leakage geometries in an ESP stage is shown in Figure 3.7.

![Figure 3.6 Balancing of Axial Thrust (Tuzson 2000)](image)
Figure 3.7 Leakage Geometries in an ESP Stage

The head loss through the leakage passage can be expressed as:

\[ h_{\text{leakage}} = H_{IO} - \frac{u_z^2 - u_{LK}^2}{2g} \]  \hspace{1cm} (3.58)

where \( H_{IO} \) is the head increase across the impeller, \( U_{LK} \) is the tangential velocity due to the impeller rotation at the leakage. The head increase across the impeller can be calculated as:

\[ H_{IO} = H_{EE} - H_{FI} - H_{TI} \]  \hspace{1cm} (3.59)

The tangential velocity due to the impeller rotation at the leakage is:

\[ U_{LK} = R_{LK} \Omega \]  \hspace{1cm} (3.60)

where \( R_{LK} \) is the radius of the leakage clearance.

As shown in Figure 3.7, the secondary flow passage is complicated. Therefore, friction, sudden expansion, and sudden contraction are considered to obtain the total head loss through the leakage area:
\[ h_{\text{leakage}} = f_{\text{CON}} \frac{v_L^2}{2g} + f_{\text{EXP}} \frac{v_L^2}{2g} + f_{\text{LK}} \frac{v_L^2 L_G}{2g S_I} \quad (3.61) \]

where \( L_G \) is the leakage channel length, \( S_I \) is the leakage width, and \( f_{\text{CON}} \) and \( f_{\text{EXP}} \) are the local hydraulic loss factors due to contraction and expansion. A \( f_{\text{CON}} = 0.5 \) and \( f_{\text{EXP}} = 1.0 \) are used.

\[ V_L = \sqrt{\frac{2g h_{\text{leakage}}}{f_{\text{LK}} S_I^2 + 1.5}} \quad (3.62) \]

Then, the leakage flow rate,

\[ Q_{\text{LK}} = 2\pi R_{\text{LK}} S_L V_L \quad (3.63) \]

### 3.2.4 Correction Factors for Theoretical Head

The slip factor, \( \delta_s \), is introduced to recalculate the theoretical pump head \( H_{th} \) to investigate the mismatch between the real outflow velocity and the designed outflow velocity. The Euler head with pre-rotation can be written as

\[ H_E = \frac{\Omega^2 r_L^2}{g} - \frac{Q \Omega}{2 \pi g h \tan \beta_2}, \quad (3.64) \]

where \( \delta_s \) in the previous model is proposed by Wiesner (1967) as

\[ \delta_s = 1 - \sqrt{\sin \frac{\beta_2}{Z_1}}^{0.7}, \quad (3.65) \]

\( \delta_s \) in this study is modified by Zhang (2017) as:

\[ \delta_s = 1 - \sqrt{\sin \frac{\beta_2}{Z_1}}^{1.5 \left( \frac{N_{S,ref}}{N_S} \right)^{0.4}}, \quad (3.66) \]

where \( N_{S,ref} \) is the reference specific speed.

The theoretical head \( H_{th} \) can then be calculated as
$$H_{th} = \delta_s \frac{\Omega^2 r_2^2}{g} - \frac{Q\Omega}{2\pi gh \tan \beta_2}$$ (3.67)

### 3.3 Mechanistic Model Setup

With all the losses determined, the mechanistic model of ESP performance prediction can be established following the flow chart, as shown below.

![Flow Chart of Modified Mechanistic Model](image)

**Figure 3.8 Flow Chart of Modified Mechanistic Model**

The model is coded in Python in order to obtain a better prediction efficiency with a much
less calculation time. The fluid properties, pump geometries, and operating conditions are input into the code as parameters for the calculations of the model. The pump geometries are shown in Table 3.1.

**Table 3.1 TE-2700 ESP Specifications**

<table>
<thead>
<tr>
<th>ESP</th>
<th>Component</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Impeller</td>
<td>Blade Number</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>Tangential Blade angle at Inlet (°)</td>
<td>19.5</td>
</tr>
<tr>
<td></td>
<td>Tangential Blade angle at Outlet (°)</td>
<td>24.7</td>
</tr>
<tr>
<td></td>
<td>Blade Thickness (mm)</td>
<td>2.72</td>
</tr>
<tr>
<td></td>
<td>Channel Length (mm)</td>
<td>76.0</td>
</tr>
<tr>
<td></td>
<td>Inlet Channel Height (mm)</td>
<td>12.2</td>
</tr>
<tr>
<td></td>
<td>Outlet Channel Height (mm)</td>
<td>7.84</td>
</tr>
<tr>
<td></td>
<td>Inlet Radius (mm)</td>
<td>17.5</td>
</tr>
<tr>
<td></td>
<td>Outlet Radius (mm)</td>
<td>56.1</td>
</tr>
<tr>
<td>Diffuser</td>
<td>Vane Number</td>
<td>9</td>
</tr>
<tr>
<td></td>
<td>Channel Length (mm)</td>
<td>87.1</td>
</tr>
<tr>
<td></td>
<td>Partition Wall Thickness (mm)</td>
<td>4.48</td>
</tr>
<tr>
<td></td>
<td>Inlet Radius (mm)</td>
<td>54.7</td>
</tr>
<tr>
<td></td>
<td>Outlet Radius (mm)</td>
<td>22.0</td>
</tr>
</tbody>
</table>

The model is compared with the catalog curve provided by the manufacturer at 3,500 rpm and 3,000 rpm. As shown in Figure 3.9 and Figure 3.10, the error between the predicted results and the catalog curve is less than 10%.
Figure 3.9 Comparison between Modified Model and Catalog

Figure 3.10 Error between the Predicted Results and Catalog Curve
3.4 Mechanistic Model Validation

In this section, the mechanistic model is validated by experimental data of TE-2700 with a broad range of viscosity at the rotational speed of 1,800 rpm, 2,400 rpm, 3,000 rpm, and 3,500 rpm as shown in Figure 3.11 to Figure 3.14. The improvement of the modified model, which adopts new correlation factors, is analyzed by comparing it with the previous model predictions. The validation covers viscosities ranging from 1 to 400 cP, and the results agree well with the experimental tests. Figure 3.15 is the comparison between experimental data and model predictions for the oil-water emulsion with a water cut of 5%. The emulsion is treated as a single-phase liquid, with the volumetric average density of oil and water. The results match the model in general.

Figure 3.11 TE-2700 ESP Comparison of Modified Model and Experimental Data at 1,800 rpm
Figure 3.12 TE-2700 ESP Comparison of Modified Model and Experimental Data at 2,400 rpm

Figure 3.13 TE-2700 ESP Comparison of Modified Model and Experimental Data at 3,000 rpm
The new model is compared with the previous version against the experimental data, as shown in Figure 3.16 to Figure 3.18. Both models can predict the pump head with high accuracy.
Compared with the previous version, the improved model is more accurate at lower flow rates.

Figure 3.16 TE-2700 ESP Comparison of Modified Model and Previous Version against Experiment Data at 1,800 rpm

Figure 3.17 TE-2700 ESP Comparison of Modified Model and Previous Version against Experiment Data at 2,400 rpm
Figure 3.18 TE-2700 ESP Comparison of Modified Model and Previous Version against Experiment Data at 3,500 rpm
CHAPTER 4

CONCLUSIONS AND RECOMMENDATIONS

The fluid viscosity effect and emulsions effect on ESP performance are investigated experimentally and modeled mechanistically in this study. The main conclusions and recommendations are summarized below.

4.1 Conclusions

4.1.1 Experimental Study

1. The reliability of the experimental setup in this study was validated by comparisons with the catalog curve and the previous experimental data collected by Zhang (2017).

2. The TE-2700 ESP was tested with ISO VG320 oil at rotational speeds of 1800, 2400, 3000, and 3500 rpm. The test oil viscosities are 60 to 400 cP at 1800 rpm, 80 to 290 cP at 2400 rpm, 40 to 130 cP at 3000 rpm, and 50 to 190 cP at 3500 rpm.

3. The fluid viscosity is affected by loop temperature, and the viscosity-temperature relationship is measured using a rotational viscometer with collected samples.

4. Pump head decreases with the increase of oil viscosity, which is caused by higher energy loss due to increased shear stress.
5. The relationship between the pump head and flow rate becomes more linear at high fluid viscosity than that of low fluid viscosity.

### 4.1.2 Mechanistic Modeling

1. The previous TUALP ESP mechanistic model has been improved by replacing the friction head loss correlation and theoretical head correlation. Cross-section shape effect ($F_r$), the pipe curvature effect ($F_p$), and the rotational speed effect ($F_o$) are included in the new friction correlation. A new slip factor correlation is added to calculate the theoretical head.

2. The hydraulic pressure increase predicted by the new mechanistic model with water as the working fluid agrees well with the catalog curve.

3. The old mechanistic model and the new mechanistic model are compared against experimental data at different rotational speeds. Compared with the prediction of the previous model, the modified model matches the experimental data and trend better at a lower flow rate.

### 4.2 Recommendations

The following suggestions are made for the future ESP high viscosity and water-oil emulsion test and modeling:

1. Oil with higher viscosity can be tested to validate the proposed mechanistic model in the broader viscosity range.

2. The torque sensor needs to be replaced to obtain accurate torque and efficiency curve.
3. At the beginning of the test, gas may be carried by working fluid. Therefore, the liquid input port should be extended to ensure that gas will be separated in this section.

4. The pipe-in-pipe heat exchanger needs to be improved to maintain the loop temperature better.
NOMENCLATURE

BEP  best efficiency point

BMP  best match point

DAQ  data acquisition system

ESP  electrical submersible pump

PV   pipe viscometer

VSD  variable speed drive

$A_{SD}$  diffuser channel total wall area, $L^2$, m$^2$

$A_{SI}$  impeller channel total wall area, $L^2$, m$^2$

$C_1$  absolute fluid velocity at impeller inlet, $L/T$, m/s

$C_{1M}$  meridional velocity at impeller inlet, $L/T$, m/s

$C_{1U}$  fluid tangential velocity at impeller inlet, $L/T$, m/s

$C_2$  absolute fluid velocity at impeller outlet, $L/T$, m/s

$C_{2B}$  absolute fluid velocity at impeller outlet corresponding to BMP, $L/T$, m/s

$C_{2E}$  effective velocity at impeller outlet, $L/T$, m/s

$C_{2F}$  fluid velocity outside impeller, $L/T$, m/s

$C_{2M}$  meridional velocity at impeller outlet, $L/T$, m/s

$C_{2P}$  projected velocity, $L/T$, m/s
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{2U}$</td>
<td>fluid tangential velocity at impeller outlet, L/T, m/s</td>
</tr>
<tr>
<td>$C_D$</td>
<td>drag coefficient</td>
</tr>
<tr>
<td>$C_H$</td>
<td>head correction factor</td>
</tr>
<tr>
<td>$C_q$</td>
<td>flowrate correction factor</td>
</tr>
<tr>
<td>$C_\eta$</td>
<td>efficiency correction factor</td>
</tr>
<tr>
<td>$\bar{a}$</td>
<td>average droplet sizes, L, m</td>
</tr>
<tr>
<td>$d$</td>
<td>impeller diameter, L, m</td>
</tr>
<tr>
<td>$D_C$</td>
<td>representative impeller channel width at outlet, L, m</td>
</tr>
<tr>
<td>$D_D$</td>
<td>diffuser representative (hydraulic) diameter, L, m</td>
</tr>
<tr>
<td>$D_I$</td>
<td>impeller representative (hydraulic) diameter, L, m</td>
</tr>
<tr>
<td>$D_L$</td>
<td>leakage diameter, L, m</td>
</tr>
<tr>
<td>$dP$</td>
<td>differential pressure, $M/(LT^2)$, Pa</td>
</tr>
<tr>
<td>$f$</td>
<td>friction factor</td>
</tr>
<tr>
<td>$f_{FD}$</td>
<td>friction factor in diffuser</td>
</tr>
<tr>
<td>$f_{FI}$</td>
<td>friction factor in impeller</td>
</tr>
<tr>
<td>$f_{LK}$</td>
<td>leakage friction factor</td>
</tr>
<tr>
<td>$f_{TD}$</td>
<td>local drag coefficient in diffuser</td>
</tr>
<tr>
<td>$f_{TI}$</td>
<td>local drag coefficient in impeller</td>
</tr>
<tr>
<td>$F_\gamma$</td>
<td>cross-section shape effect</td>
</tr>
<tr>
<td>$F_\beta$</td>
<td>pipe curvature effect</td>
</tr>
<tr>
<td>$F_\omega$</td>
<td>rotational speed effect</td>
</tr>
</tbody>
</table>
\( h \) channel height, L, m
\( H \) pump head, L, m
\( H_{BEP} \) head at BEP, L, m
\( H_E \) Euler’s head, L, m
\( H_{EE} \) effective Euler’s head, L, m
\( H_{FD} \) head loss due to friction in diffuser, L, m
\( H_{FI} \) head loss due to friction in impeller, L, m
\( H_{IO} \) head increase across impeller, L, m
\( H_{LK} \) pressure head difference across leakage, L, m
\( H_{oil} \) pump head with oil, L, m
\( H_{TD} \) head loss due to turn in diffuser, L, m
\( H_{TI} \) head loss due to turn in impeller, L, m
\( H_{vis} \) pump head with viscous fluid, L, m
\( H_{water} \) pump head with water, L, m
\( L_D \) diffuser channel length, L, m
\( L_{LK} \) leakage channel length, L, m
\( L_I \) impeller channel length, L, m
\( N \) rotational speed, 1/T, rpm
\( N_s \) specific speed
\( P \) pressure, M/(LT^2), Pa
\( P_2 \) shaft power, ML^2/T^3, kg⋅m^2/s^3
\( P_{hyd} \) hydraulic power, \( ML^2/T^3 \), \( kg \cdot m^2/s^3 \)

\( Q \) volumetric flowrate, \( L^3/T \), \( m^3/s \)

\( q_{BEP} \) flowrate at BEP, \( L^3/T \), \( m^3/s \)

\( Q_{BMP} \) volumetric flowrate at BMP, \( L^3/T \), \( m^3/s \)

\( Q_{LK} \) leakage volumetric flowrate, \( L^3/T \), \( m^3/s \)

\( q_{vis} \) viscous fluid flowrate, \( L^3/T \), \( m^3/s \)

\( q_{water} \) water flowrate, \( L^3/T \), \( m^3/s \)

\( R_1 \) radius of impeller inlet, \( L \), \( m \)

\( R_2 \) radius of impeller outlet, \( L \), \( m \)

\( R_D \) Reynolds number by Ippen

\( Re \) Reynolds number

\( Re_C \) recirculation effect Reynolds number

\( Re_D \) Reynolds numbers in diffuser

\( Re_I \) Reynolds numbers in impeller

\( Re_L \) leakage Reynolds number

\( Re_{Stepanoff} \) Reynolds number by Stepanoff

\( R_{LK} \) radius corresponding to leakage, \( L \), \( m \)

\( S_L \) leakage width, \( L \), \( m \)

\( St \) Strouhal number

\( t \) time, \( T \), \( s \)

\( T \) temperature, \( ^\circ C \)
\( T \) blade thickness, L, m

\( U_1 \) impeller tangential velocity at inlet, L/T, m/s

\( U_2 \) impeller tangential velocity at outlet, L/T, m/s

\( U_{LK} \) tangential velocity due to impeller rotation at leakage, L/T, m/s

\( v \) velocity, L/T, m/s

\( V \) volume, L\(^3\), m\(^3\)

\( V_D \) representative fluid velocity in diffuser, L/T, m/s

\( V_I \) representative fluid velocity in impeller, L/T, m/s

\( V_L \) fluid velocity at leakage, L/T, m/s

\( V_{ol_D} \) diffuser channel volume, L\(^3\), m\(^3\)

\( V_{ol_I} \) impeller channel volume, L\(^3\), m\(^3\)

\( V_S \) shear velocity, L/T, m/s

\( W_1 \) relative velocity with respect to impeller at inlet, L/T, m/s

\( W_2 \) relative velocity with respect to impeller at outlet, L/T, m/s

\( We \) Weber number

\( y_{I1} \) impeller inlet height, L, m

\( y_{I2} \) impeller outlet height, L, m

\( Z_I \) impeller blade number

**Greek Symbols**

\( \eta \) efficiency
\( \mu \)  
Dynamic viscosity, \( \text{M}/(\text{LT}), \text{Pa} \cdot \text{s} \)

\( \mu_{\text{eff}} \)  
Effective viscosity, \( \text{M}/(\text{LT}), \text{Pa} \cdot \text{s} \)

\( \beta_1 \)  
Blade angle from tangential at impeller inlet, \( ^\circ \)

\( \beta_2 \)  
Blade angle from tangential at impeller outlet, \( ^\circ \)

\( \eta_{\text{vis}} \)  
Efficiency with viscous fluid

\( \eta_{\text{water}} \)  
Efficiency with water

\( \nu \)  
Kinematic viscosity, \( \text{L}^2/\text{T}, \text{m}^2/\text{s} \)

\( \rho \)  
Density, \( \text{M}/\text{L}^3, \text{kg}/\text{m}^3 \)

\( \sigma_s \)  
Slip factor

\( \omega \)  
Angular velocity, \( 1/\text{T}, 1/\text{s} \)

\( \Omega \)  
Angular speed, \( \text{L}/\text{T}, \text{m}/\text{s} \)

\( \phi \)  
Flow coefficient

\( \psi \)  
Head coefficient
BIBLIOGRAPHY


## APPENDIX A

### EQUIPMENT AND INSTRUMENT SPECIFICATIONS

Table A.1 TUALP High-Viscosity ESP Flow Loop Equipment Specifications

<table>
<thead>
<tr>
<th>Equipment</th>
<th>Model</th>
<th>Capacity</th>
<th>Purpose</th>
</tr>
</thead>
<tbody>
<tr>
<td>ESP Pump</td>
<td>GE Oil &amp; Gas Wood Group TE-2700</td>
<td>BEP: 2700 bpd, 3,500 rpm</td>
<td>Testing Bench</td>
</tr>
<tr>
<td>Electric Motor</td>
<td>North American H3650</td>
<td>50 hp</td>
<td>Drive Motor</td>
</tr>
<tr>
<td>Air Compressor</td>
<td>Kaeser CSD60</td>
<td>186 cfm, 217 psi</td>
<td>Air Source</td>
</tr>
<tr>
<td>Air Pressure Regulator</td>
<td>Speedaire 4ZM22</td>
<td>300 psi Max Inlet Pressure, 150 psi Max Outlet Pressure</td>
<td>Air Pressure Regulation</td>
</tr>
<tr>
<td>Variable Speed Drive</td>
<td>Hitachi L300P</td>
<td>50 hp</td>
<td>Altering Rotational Speed</td>
</tr>
<tr>
<td>ESP Thrust Chamber</td>
<td>Schlumberger REDA NO.88AB1- LT</td>
<td>-</td>
<td>Thrust Bearing Box</td>
</tr>
<tr>
<td>Liquid Pneumatic Control Valve</td>
<td>Fisher Body ED Actuator 657 Positioner 582i</td>
<td>-</td>
<td>Liquid Flow Rate Control</td>
</tr>
<tr>
<td>Water Pump</td>
<td>Dayton Stainless Steel Centrifugal Pump 2ZWT9A</td>
<td>0.5 hp</td>
<td>Water Circulation</td>
</tr>
<tr>
<td>Water Tank</td>
<td>Value Brand T-0300-059</td>
<td>300 gal</td>
<td>Water Storage</td>
</tr>
<tr>
<td>Water Pneumatic Control Valve</td>
<td>Fisher Body V100 Actuator 1052 Positioner 3622</td>
<td>-</td>
<td>Water Flow Rate Control</td>
</tr>
</tbody>
</table>
### Table A.2 TUALP High-Viscosity ESP Flow Loop Instrumentation Specifications

<table>
<thead>
<tr>
<th>Instrument</th>
<th>Model</th>
<th>Range</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermocouple</td>
<td>Thermo Electric TCMSC83077875</td>
<td>0 to 1600 °F</td>
<td>±0.75%</td>
</tr>
<tr>
<td>Resistance Temperature Detector</td>
<td>Omega PR-11-2-100-1/8-18-E</td>
<td>-200 to 600 °C</td>
<td>±0.15°C</td>
</tr>
<tr>
<td>Temperature Transmitter</td>
<td>INOR IPAQ R520</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Absolute Pressure Transmitter</td>
<td>Emerson Rosemount 2051</td>
<td>0 to 500 psi</td>
<td>±0.1%</td>
</tr>
<tr>
<td>Differential Pressure Transmitter</td>
<td>Emerson Rosemount 3051S</td>
<td>-10 to 50 psig</td>
<td>±0.1%</td>
</tr>
<tr>
<td>Pipe Viscometer</td>
<td>Emerson Rosemount 3051S</td>
<td>-250 to 250 psig</td>
<td>±0.1%</td>
</tr>
<tr>
<td>Coriolis High Flow Rate Meter</td>
<td>Proline Promass 80E</td>
<td>0 to 6615 lb/min</td>
<td>Mass Flow: ±0.2%</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Volume Flow: ±0.2%</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Density: ±0.0005 g/cm³</td>
</tr>
<tr>
<td>Coriolis Low Flow Rate Meter</td>
<td>Micro Motion CMF200</td>
<td>0 to 1600 lb/min</td>
<td>Mass Flow: ±0.1%</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Volume Flow: ±0.1%</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Density: ±0.0005 g/cm³</td>
</tr>
</tbody>
</table>

### Table A.3 TUALP High-Viscosity ESP Flow Loop DAQ Specifications

<table>
<thead>
<tr>
<th>Device</th>
<th>Features</th>
</tr>
</thead>
<tbody>
<tr>
<td>Data Processing Computer</td>
<td>Dell Optiplex 9020, i7-4770 CPU @ 3.4 GHz, RAM: 16GB, HD: 1TB</td>
</tr>
<tr>
<td>National Instruments cFP-AI-110</td>
<td>• Eight analog voltage or current input channels</td>
</tr>
<tr>
<td></td>
<td>• Eight voltage input ranges: 0–1 V, 0–5 V, 0–10 V, ±60 mV, ±300 mV, ±1 V, ±5 V, and ±10 V</td>
</tr>
<tr>
<td></td>
<td>• Three current input ranges: 0–20, 4–20, and ±20 mA</td>
</tr>
<tr>
<td></td>
<td>• 16-bit resolution</td>
</tr>
<tr>
<td></td>
<td>• Three filter settings: 50, 60, and 500 Hz</td>
</tr>
<tr>
<td></td>
<td>• 250 Vrms CAT II continuous channel-to-ground isolation, verified by 2,300 Vrms dielectric withstand test</td>
</tr>
<tr>
<td></td>
<td>• −40 to 70 °C operation</td>
</tr>
<tr>
<td></td>
<td>• Hot-swappable</td>
</tr>
<tr>
<td>National Instruments cFP-AI-111</td>
<td>• Sixteen single-ended analog current input channels</td>
</tr>
<tr>
<td></td>
<td>• Three input ranges: ±20, 0–20, and 4–20 mA</td>
</tr>
<tr>
<td></td>
<td>• 16-bit resolution</td>
</tr>
</tbody>
</table>
| National Instruments cFP-AO-200 | • Three filter settings: 50, 60, and 500 Hz  
• Hot-swappable  
• 2300 $V_{\text{rms}}$ transient overvoltage protection  
• –40 to 70 °C operation |
| National Instruments cFP-1804 | • Eight 0–20 or 4–20 mA outputs  
• 0.5 mA over ranging  
• 12-bit resolution  
• Up to 1 kΩ load impedance (with 24 V loop supply)  
• Indicators for open current loops  
• Short-circuit protection  
• 2300 $V_{\text{rms}}$ transient overvoltage protection between the inter-module communication bus and the I/O channels  
• –40 to 70 °C operation  
• Hot plug-and-play |
| National Instruments cFP-CB-1 | • Network interface: 10 BaseT and 100 BaseTX Ethernet, IEEE802.3, 10/100 Mbps  
• One RS-232 (DCE) serial port, 300 to 115200 bps  
• 11 to 30 VDC, 20W  
• 2300 $V_{\text{rms}}$ transient overvoltage protection  
• –40 to 70 °C operation |
| National Instruments cFP-CB-1 | • cFP-CB-1 is designed for general-purpose and hazardous voltage operation with all Compact FieldPoint I/O modules  
• 36 terminals available  
• Tie-wrap anchors for wires  
• Color-coded V and C terminals for voltage supply and common connections  
• –40 to 70 °C operation |
<table>
<thead>
<tr>
<th></th>
<th>High Flow Meter</th>
<th>Low Flow Meter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model</td>
<td>Promass 80E</td>
<td>CMF200M</td>
</tr>
<tr>
<td>Brand</td>
<td>Endress+Hauser</td>
<td>Emerson</td>
</tr>
<tr>
<td>Meter Size (inch)</td>
<td>3</td>
<td>2</td>
</tr>
<tr>
<td>Accuracy</td>
<td>0.20%</td>
<td>0.25%</td>
</tr>
<tr>
<td>Accurate Measurement Range (bpd)</td>
<td>&gt;1700</td>
<td>&gt;150</td>
</tr>
<tr>
<td>Pressure Drop at 300 cP (psi)</td>
<td>12.2 at 10,000 bpd</td>
<td>13.7 at 2,100 bpd</td>
</tr>
<tr>
<td>Pressure Drop at 700 cP (psi)</td>
<td>18.8 at 10,000 bpd</td>
<td>25.6 at 2,100 bpd</td>
</tr>
<tr>
<td>Pressure Drop at 1000 cP (psi)</td>
<td>23.0 at 10,000 bpd</td>
<td>34.1 at 2,100 bpd</td>
</tr>
</tbody>
</table>

**Table A.5 Pipe-in-Pipe Heat Exchanger Design Data**

<table>
<thead>
<tr>
<th>Fluid</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Transfer Media</td>
<td>Water</td>
</tr>
<tr>
<td>Volume Flow Rate (bpd)</td>
<td>1940</td>
</tr>
<tr>
<td>Inlet Temperature (°C)</td>
<td>0</td>
</tr>
<tr>
<td>Outlet Temperature (°C)</td>
<td>1</td>
</tr>
<tr>
<td>Density (kg/m³)</td>
<td>1001</td>
</tr>
<tr>
<td>Specific Heat (W/K)</td>
<td>4129</td>
</tr>
<tr>
<td>Viscosity (cP)</td>
<td>1.7</td>
</tr>
<tr>
<td>Thermal Conductivity (W/m·K)</td>
<td>0.570</td>
</tr>
<tr>
<td>Pressure Drop (psi)</td>
<td>0.00</td>
</tr>
<tr>
<td>Log Mean Temp Difference (°C)</td>
<td>34.25</td>
</tr>
<tr>
<td>Heat Transfer Rate (BTU/hr)</td>
<td>50375</td>
</tr>
<tr>
<td>Heat Transfer Area (ft²)</td>
<td>21</td>
</tr>
<tr>
<td>Length (ft)</td>
<td>27</td>
</tr>
</tbody>
</table>
Figure A.1 Pressurization Port and Gas Discharge Valve

Figure A.2 Oil Injection Port
Figure A.3 Coriolis Flow Meter

Figure A.4 Temperature Sensor
Figure A.5 Temperature Sensor

Figure A.6 Data Acquisition Device
Figure A.7 Pressure Monitors

Figure A.8 Pneumatic Control Valve
Figure A.9 Cooling System

Figure A.10 Pipe-in-Pipe Heat Exchanger

Figure A.11 Fluid Flow Schematic inside the Heat Exchanger
Figure A.12 Rotational Viscometer

Figure A.13 Water bath Temperature Control and Circulator
APPENDIX B

CHANNEL DISTRIBUTIONS OF NI MODULES

Table B.1 cFP-AI-110 Module #1 Channel Distributions

<table>
<thead>
<tr>
<th>Channel</th>
<th>Description</th>
<th>Measurement Location</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>Rotational Speed</td>
<td>ESP Shaft</td>
</tr>
<tr>
<td>2</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Not Used</td>
<td></td>
</tr>
</tbody>
</table>

Table B.2 cFP-AI-111 Module #2 Channel Distributions

<table>
<thead>
<tr>
<th>Channel</th>
<th>Description</th>
<th>Measurement Location</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Differential Pressure</td>
<td>ESP Stage 6-7</td>
</tr>
<tr>
<td>1</td>
<td>Temperature</td>
<td>ESP Stage 14</td>
</tr>
<tr>
<td>2</td>
<td>Temperature</td>
<td>ESP Intake</td>
</tr>
<tr>
<td>3</td>
<td>Mass Flow Rate</td>
<td>Promass 80E</td>
</tr>
<tr>
<td>4</td>
<td>Fluid Density</td>
<td>Promass 80E</td>
</tr>
<tr>
<td>5</td>
<td>Mass Flow Rate</td>
<td>CMF200M</td>
</tr>
<tr>
<td>6</td>
<td>Fluid Density</td>
<td>CMF200M</td>
</tr>
<tr>
<td>7</td>
<td>Differential Pressure</td>
<td>ESP Stage 4-14</td>
</tr>
<tr>
<td>8</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>Differential Pressure</td>
<td>ESP Stage 4-5</td>
</tr>
<tr>
<td>13</td>
<td>Differential Pressure</td>
<td>ESP Stage 5-6</td>
</tr>
<tr>
<td>14</td>
<td>Temperature</td>
<td>Water Tank</td>
</tr>
<tr>
<td>15</td>
<td>Temperature</td>
<td>Coriolis Meter</td>
</tr>
</tbody>
</table>
Table B.3 cFP-AI-111 Module #3 Channel Dsicturbutions

<table>
<thead>
<tr>
<th>Channel</th>
<th>Description</th>
<th>Measurement Location</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Absolute Pressure</td>
<td>ESP Stage 8</td>
</tr>
<tr>
<td>6</td>
<td>Absolute Pressure</td>
<td>ESP Stage 9</td>
</tr>
<tr>
<td>7</td>
<td>Absolute Pressure</td>
<td>ESP Stage 10</td>
</tr>
<tr>
<td>8</td>
<td>Absolute Pressure</td>
<td>ESP Stage 11</td>
</tr>
<tr>
<td>9</td>
<td>Absolute Pressure</td>
<td>ESP Stage 12</td>
</tr>
<tr>
<td>10</td>
<td>Absolute Pressure</td>
<td>ESP Intake</td>
</tr>
<tr>
<td>11</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>13</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>14</td>
<td>Not Used</td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>Not Used</td>
<td></td>
</tr>
</tbody>
</table>

Table B.4 cFP-AO-200 Module #4 Channel Dsicturbutions

<table>
<thead>
<tr>
<th>Channel</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Liquid Control Valve</td>
</tr>
<tr>
<td>1</td>
<td>Variable Speed Drive</td>
</tr>
<tr>
<td>2</td>
<td>Water Control Valve</td>
</tr>
<tr>
<td>3</td>
<td>Not Used</td>
</tr>
<tr>
<td>4</td>
<td>Not Used</td>
</tr>
<tr>
<td>5</td>
<td>Not Used</td>
</tr>
<tr>
<td>6</td>
<td>Not Used</td>
</tr>
<tr>
<td>7</td>
<td>Not Used</td>
</tr>
</tbody>
</table>
There were some problems with the viscosity experiment devices. The most important one was the pneumatic control valve. The seal in the pneumatic control valve was broken, thus causing a leak even if the valve is fully closed, leading to a large fluctuation at relatively high mass flowrate before the seal replacement.

To test the sensitivity of the pneumatic control valve, the mass flow rate in a certain period at a certain valve opening was collected by the data acquisition system. In this test, 600 points were collected in 10 minutes, which means one point is acquired per second. The valve is regulated to be 0% closed (Qmax), 60% closed (0.4 Qmax), 80% closed (0.2 Qmax) and 90% closed (0.1 Qmax). The mass flow rate is assumed to be proportional to the valve opening degree. The collected mass flowrate is transferred to volume flowrate.

The figures below show the variation of the volume flowrate in 10 minutes. Figure C.1 shows the variation of volume flow rate when the valve is fully open (0% closed). A large fluctuation occurs during the time. The maximum value is about 2500 bpd while the minimum value is about 1227 bpd. The average value is 1728.77 bpd.

Figure C.2 shows the variation of volume flow rate when the valve is 90% closed. The volume flow rate is steady during the time.
Figure C.3 shows the comparison of volume flowrate variation when the valve is 0% closed and 60% closed. There is almost no difference between the two situations. Furthermore, the average volume flowrate is almost the same.

Figure C.4 shows the comparison of volume flowrate variation when the valve is 0% closed and 80% closed. There are no obvious differences between the two situations. Moreover, the average volume flowrate is 1728.77 bpd and 1611.87 bpd separately.

Figure C.5 shows the comparison of volume flowrate variation when the valve is 0% closed and 90% closed. Obvious differences can be seen between the two situations. This means only when the valve opening degree is more than 80% closed, the valve can be relatively sensitive while regulating.

From the previous experiments, it is confirmed that there is no big difference in the average volume flow rate when the valve is regulated from 0% closed to 80% closed. However, the average volume flowrate changes significantly during the process, from 80% closed to 100% closed off the valve.

The errors between the maximum/minimum volume flow rate and the average volume flow rate are shown in Table C.1. The biggest error can be about 30% at a high-volume flowrate while it reduces to about 17% at a low volume flow rate. Since the temperature is continuously increasing during the experiment (the water to cooling the oil was not used since the temperature was below zero), the viscosity of oil decreases during the time. Although the variation of viscosity will affect the volume flowrate, it cannot deny the fact that there are some problems with the valve.
Figure C.1 The variation of volume flow rate when the valve is 0% closed

Figure C.2 The variation of volume flow rate when the valve is 90% closed
Figure C.3 The variation of volume flow rate when the valve is 0% and 60% closed

Figure C.4 The variation of volume flow rate when the valve is 0% and 80% closed
Figure C.5 The variation of volume flow rate when the valve is 0% and 90% closed

Table C.1 Errors in the maximum/minimum volume flow rate and the average volume flow rate

<table>
<thead>
<tr>
<th>Volume flow rate</th>
<th>Qmax (bpd)</th>
<th>Qmin (bpd)</th>
<th>Ave (bpd)</th>
<th>Error max-ave (%)</th>
<th>Error min-ave (%)</th>
<th>Average viscosity (cP)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Qmax (0% closed)</td>
<td>2250</td>
<td>1500</td>
<td>1728.77</td>
<td>30.1503381</td>
<td>13.23310793</td>
<td>114.38</td>
</tr>
<tr>
<td>0.4 Qmax (60% closed)</td>
<td>2250</td>
<td>1500</td>
<td>1739.08</td>
<td>29.37875198</td>
<td>13.74749868</td>
<td>78.76</td>
</tr>
<tr>
<td>0.2 Qmax (80% closed)</td>
<td>2000</td>
<td>1250</td>
<td>1611.87</td>
<td>24.07948532</td>
<td>22.45032168</td>
<td>65.21</td>
</tr>
<tr>
<td>0.1 Qmax (90% closed)</td>
<td>1500</td>
<td>1170</td>
<td>1278.41</td>
<td>17.3332499</td>
<td>8.480065081</td>
<td>48.4</td>
</tr>
</tbody>
</table>

Some conclusions can be made about the pneumatic control valve:

1. The fluctuation is significant when the volume flow rate is high. Moreover, it becomes
steady when the volume flow rate is low.

2. The average volume flowrate changes little when the valve opening degree ranges from 0% closed to 80% closed and changes significantly when it ranges from 80% closed to 100% closed.

When the pneumatic valve was uninstalled and sent to be repaired, the flow rate is regulated by a manual valve. However, it is hard to close the manual valve at high rotational speed or low flowrate. Because of the pressure upstream, the valve is vibrating at high rotational speed and low flowrate. As a result, the maximum rotational speed is 2400 rpm. Figure C.6 shows how the experiment was conducted without the pneumatic control valve. The section of the pneumatic control valve was replaced by a temporary pipe to connect the flowmeter and the manual control valve to make sure the experimental facility remained a closed loop.

Figure C.6 Loop with Pneumatic Valve Uninstalled