44TH TURBOMACHINERY & 31ST PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 14 – 17 2015
GEORGE R. BROWN CONVENTION CENTER

HIGH VISCOSITY TEST OF A CRUDE OIL PUMP

Brigitte Le Fur
CETIM - Centre Technique des Industries Mécaniques
74, route de la Jonelière
BP 82617, 44326 Nantes, France

Cato Knutsen Moe
EUREKA PUMPS
Magnus Poulssons vei 7
NO-1366 Lysaker, Norway

Fabien Cerru
CETIM - Centre Technique des Industries Mécaniques
74, route de la Jonelière
BP 82617, 44326 Nantes, France

Dr. Brigitte Le Fur is now retired from Cetim, where she worked as a lead engineer in the Fluid and Flow Technologies Department. She attended ENSEEIHT in France where she received her engineering degree and PhD in fluid mechanics. She worked at Cetim to design test facilities for complex equipment using multi-phase flow and viscous fluids. She is the author of numerous papers on pumps and viscous flows.

Cato Knutsen Moe is a Senior Specialist Engineer at Eureka Pumps Engineering Department. He joined Eureka Pumps in 2009 and moved from the Application & Product Development Department to his current role in 2013. He received his MSc degree at NTNU, Norwegian University of Science and Technology. Previously Mr. Moe worked for Kværner Eureka and AkerSolutions Subsea on several EPC- R&D projects within the Topside and Subsea Oil & Gas industry. He held various positions there and specialized in pump hydraulic design and development.

Dr. Fabien Cerru is project and research engineer at CETIM in the Fluid and Flow Technologies Department. His organization is responsible for simulating and testing process equipment, e.g. valves, pumps, in a wide variety of conditions. Dr. Cerru attended ENSEEIHT in France where he received an engineering degree in fluid mechanics. He obtained his MSc at University of Manchester, U.K., in Thermal Power and Fluid Engineering and his PhD in Mechanical Engineering at Imperial College London, U.K. Previously, Dr. Cerru worked as a process and safety engineer at AREVA in the design of nuclear power plants.

ABSTRACT

In 2014 a full scale high viscosity pump performance test was performed for a 2-stage vertically suspended (VS1) API610 pump. The high viscosity test scope was part of an ongoing EPC project and exclusively conducted for industrial purposes. The objective was to simulate real field operation with crude oil and qualify the pump for the given field conditions and requirements. Furthermore, the reason for conducting this test scope was the uncertainties in published methods of predicting high viscosity fluid pump characteristics, limited available literature, hence reduce field operation risks.

The official test scope was divided into two main scenarios:
1) Start-up test at minimum. flow and a viscosity of 3075 cP,
2) Pump performance test with an intermittent duty point (reduced flow) of 850 m3/h-125 m (3654 USgpm- 410 ft), and a viscosity of 1075 cP.
Internal tests with various viscosities from 480 cP to 3075 cP were also done in preparation for the official tests.

Measured results were then compared to published methods for pump characteristics viscosity correction. The objective for these additional analyzes was to evaluate possible alternative approaches in order to improve uncertainties with viscous prediction and ultimately reduce the number of tests/test scope in future projects. It should be emphasized that this comparison is simply for discussion within the industry and not for validation of published methods of viscosity correction for pump characteristics.

The test results provide valuable insight on the effects of high viscosity on centrifugal pump performance curves, allowing a more accurate rating and possibly a better optimization of the process equipment, e.g. electrical motor, valves, cooling system, by improving the flow characteristics predictions.

INTRODUCTION

Crude oil viscosity variation with fluid temperature is affecting pump requirements: The driver must be able to start at the highest viscosity, as well as operate within the given requirements and over the applicable viscosity range. Due to uncertainties related to estimating the effect of the viscosity on the pump performance, extensive laboratory testing performed by Cetim was part of the EPC project scope in order to verify the pump performance.
Official test scope was divided into two main scenarios to qualify the pump for the given field conditions and requirements:

- Start-up test at minimum flow and a viscosity of 2057 cP,
- Pump performance test with an intermittent duty point (reduced flow) of 850 m^3/h-125 m (3654 USgpm- 410 ft), and a viscosity of 1075 cP.

Internal tests with various viscosities from 1 cP up to 2057 cP were also performed in preparation for the official tests.
For certain pumps and specific speeds, the Hydraulic Institute provides viscosity correction factors based on pump performance curves for water, e.g. ANSI/HI 9.6.7 (2010). These guidelines apply to the tested crude oil pump.
The first Section presents the main criteria for the pump design. Then the test loop design is explained. The third part presents the test results and their comparison with the Hydraulic Institute predictions. However, the third part is limited and only for discussion within the industry and not for validation of the viscosity prediction methods.

PUMP AND MOTOR

With a satisfying pump characteristic and specific speed, a well proven design and several years in operation for both crude and Seawater Service, the pump evaluated and chosen for crude oil service was well suited. With this in mind, no extra- or special design measures were taken with respect to the pump hydraulics. However, special attention was made for the following points:

- Pump Characteristics established and satisfactory. This was achieved through previous performance tests from EPC projects.
- The uncertainties related to prediction of Pump Characteristics for viscous flow. Several published methods were investigated, both empirical and loss analysis approach.
- Due to the uncertainties related to prediction of Pump characteristics for viscous flow, a conservative approach for power- and torque rating on the electrical motor-, but also head was imperative. Sufficient head would be secured through increased impeller outlet diameter. With a maximum potential impeller outlet diameter of 530 mm (20.9 inch) and the actual outlet diameter of 446 mm (17.6 inch) this was considered satisfactory.

The main parameters for the given pump are presented in the following bullet points:

- Standard 2 stage Centrifugal pump, API 610 vertically Suspended (VS1).
- Total length in operation / Viscosity test:
  51.5 m/ 8.6 m (168.9 ft/ 28.2 ft). The pump length was reduced for practical reasons by removing riser sections.
- Total weight (dry) operation/viscosity test:
  11400 kg/ 7100 kg (25132.7/ 15652.8 lb)
- Specific speed (water) nq: 47 (rpm, m3/s, m), i.e. Ns=2428 US customary units
- Mechanical seal System: Plan 53B
- Squirrel-cage induction motor rating:
  780 kW at 60 Hz (1046 BHP)
- Speed:
  1783 rpm at 60Hz, 1490 rpm at 50 Hz
- Rated/Normal Operating Point:
  1080 m3/h-125 m (4755.1 USgpm- 410.1 ft),
  viscosity: 181 cP, Re = 4.3×10^4
- Intermittent Duty point Reduced flow:
  850 m3/h- 125 m (3654.4 USgpm- 410.1 ft),
  viscosity: 1075 cP, Re = 7.3×10^3
- Intermittent start-up at minimum flow:
  viscosity: 2057 cP, Re = 3.8×10^3

Also, see the pump sectional arrangement drawing in reduced length for test setup, Figure 1.

Rated/Normal Operating Point (181 cP):
The rated flow and head together with rotational speed determined the selection of a pump with suitable hydraulic/size, specific speed and number of stages. Due to the low viscosity, hence minor viscosity correction on a well proven pump characteristic, this task was considered to be quite accurate. No official performance test was conducted for this viscosity.

Intermittent start-up (2057 cP):
With the highest viscosity, the pump was required to start at minimum flow. Power consumption decreases together with flow and would therefore not reach the electrical motor power rating. Initially, the pump torque was considered critical for the electrical motor. However, comparing the pump with the motor torque, it was evident that the margin was satisfactory even when considering the uncertainties related to prediction of pump characteristic for high viscous flow. Torque curves are not presented.
To verify the required start-up at viscosity 2057 cP a full-scale test was conducted. This was one of the two official test scenarios for this EPC project.

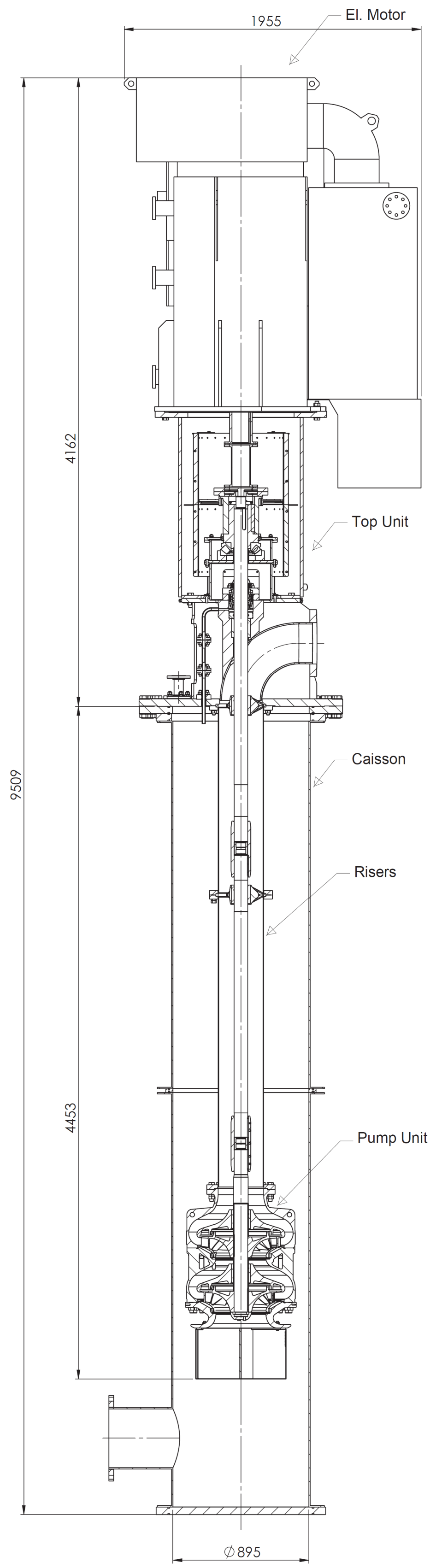

Figure 1: General arrangement of pump test configuration (by courtesy of EUREKA PUMPS AS)

Intermittent Duty point, Reduced flow (1075 cP):
Because of the significantly increased viscosity (in relation to normal operating point), flow, head and efficiency would decrease significantly and the pump would no longer be able to deliver the same flow at the required head of 125 m (410.1 ft), hence the name "reduced flow".
To verify the required head at reduced flow, the pump characteristic was established through a full-scale viscosity performance test. This was the second of two official test scenarios conducted for this EPC project.

Pump Characteristic Viscosity Correction:
The Intermittent duty point (Reduced flow) gave the highest power consumption and was decisive for rating the electrical motor.
Due to the high viscosity, the uncertainties related to prediction of pump characteristic were evident. Three different published methods were investigated, including both empirical- and loss analysis approach:

- American National Standard for Centrifugal Pumps, Std. No. ANSI/HI 1.1-1.5 (1994). Empirical.
- American National Standard for Centrifugal Pumps, Std. No. ANSI/HI 9.6.7 (2010). Empirical.
- Centrifugal Pumps, 2nd Ed. by J. F. Gülich (sec. 13.1). Loss Analysis, including empirical coefficients.

Note: For ANSI/HI 9.6.7 (2010) the equations for calculating correction coefficients are identical.
In Table 1 the calculated viscous correction coefficients are presented for B.E.P. Head and flow coefficients approximately coincide. However, the efficiency coefficient is significantly lower for HI 1.1-1.5 (1994) approach, hence the power is considerably higher.

Viscous Correction Results B.E.P				
Speed [rpm] 1783	Density [kg/m^3] 936	Density [lb/ft^3] 58.4	Viscosity [cP] 1075.0	
SI units	**Q_{Visc}** [m^3/h]	**H_{Visc}** [m]	**P_{Visc}** [kW]	**η_{Visc}** [%]
Water (1000 kg/m^3)	*1296*	*117*	*576*	*71.8*
HI 1.1-1.5 (1994)	*1169*	*104*	*787*	*39.2*
HI 9.6.7 (2010)	*1156*	*104*	*680*	*45.2*
Loss Analysis (Centr. Pumps, J.F Gülich)	1163	105	648	48.1
US customary units	**Q_{Visc}** [gal/min]	**H_{Visc}** [ft]	**P_{Visc}** [BHP]	**η_{Visc}** [%]
Water (62,4 lb/ft^3)	5704	384	773	71.8
HI 1.1-1.5 (1994)	5145	342	1056	39.2
HI 9.6.7 (2010)	5088	343	912	45.2
Loss Analysis (Centr. Pumps, J.F Gülich)	5120	345	869	48.1
Viscous Correction Coefficients:	C_Q [-]	C_H [-]	C_P [-]	C_η [-]
HI 1.1-1.5 (1994)	*0.90*	*0.89*	*1.37*	*0.55*
HI 9.6.7 (2010)	*0.89*	*0.89*	*1.18*	*0.63*
Loss Analysis (Centr. Pumps, J.F Gülich)	*0.90*	*0.90*	*1.12*	*0.67*

Table 1: Comparison of Correction Factors based on HI Correction Factor and Loss Analysis

The same observation is done outside the B.E.P, see Figure 2. For the Intermittent Duty point (Reduced flow), the power consumption varied from (see Figure 2):

- 770 kW (1033 BHP), HI 1.1-1.5 (1994)
- 620 kW (831 BHP), Centrifugal Pumps, 2nd Ed. By J. F. Gülich.

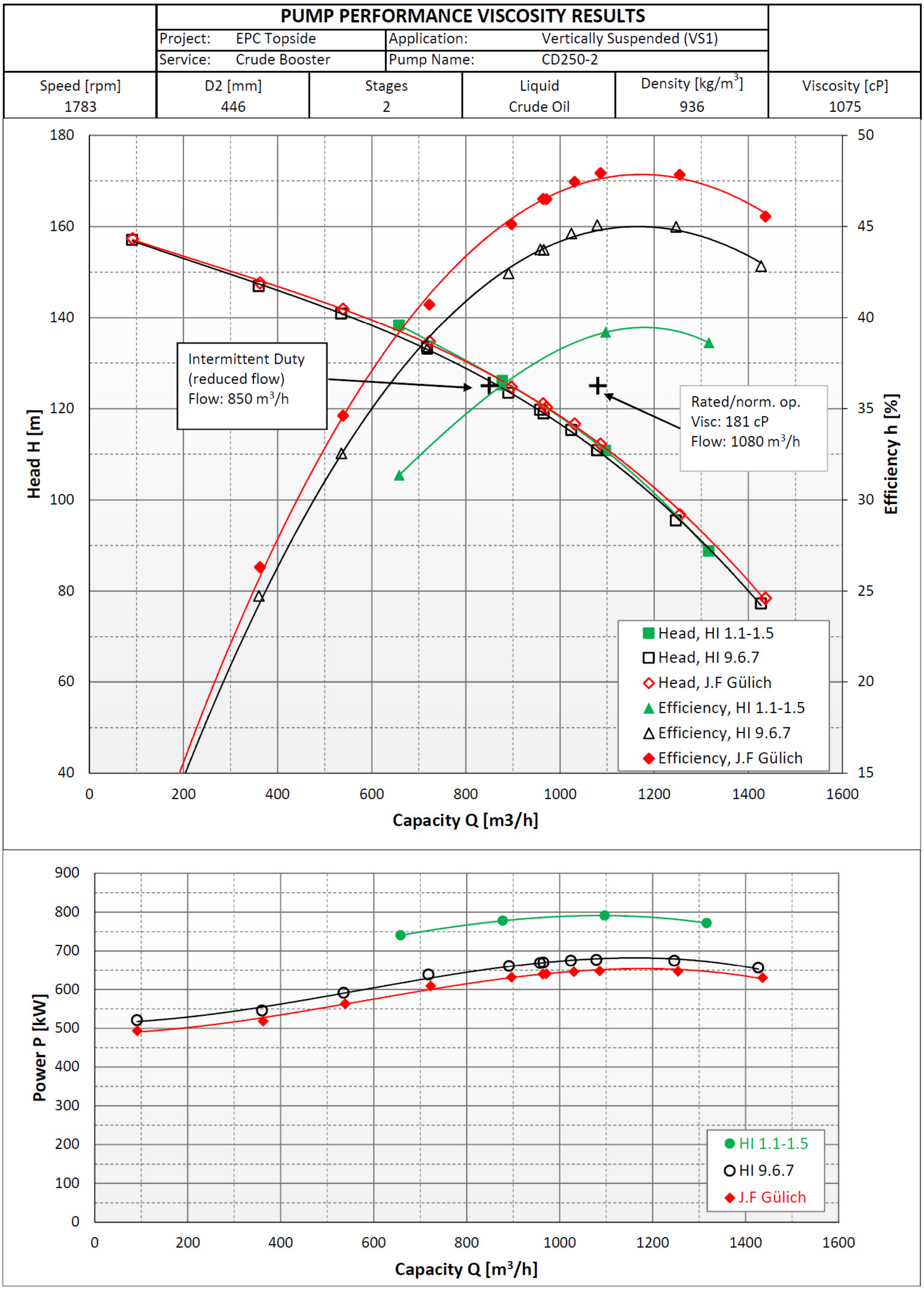

Figure 2: Estimated performance curve with corrections for viscosity 1075 cP (by courtesy of EUREKA PUMPS AS)

This is a variation of approximately 20%. However it should be emphasized that the loss analysis equations from Centrifugal Pumps, 2nd Ed. by J. F. Gülich incorporates empirical coefficients for head, flow and efficiency with an allowable variation range (maximum, minimum and mean). By increasing or decreasing these values, so will the correction coefficient, to make it possible for a more- or less conservative approach for predicting viscous pump characteristics.
Considering the high viscosity, the large variation in correction coefficients and the fact that a conservative approach was decided to use, ANSI/HI 1.1-1.5 (1994) was the preferred method when rating the electrical motor. It should be noted that this was an older HI version for predicting the pump characteristics for viscous flow.

Test results compared with Hydraulic Institute predictions:
In addition to test results presentation, a comparison with ANSI/HI- 1.1-1.5 (1994) and 9.6.7 (2010) methods of viscosity correction for pump characteristics is presented. It is simply for discussion and not for validation of the Hydraulic Institute's methods.
The loss analysis in Centrifugal Pumps, 2nd Ed, J. F. Gülich is simply left out to limit the amount of data presented, but also because of the large deviation for the power prediction.

TEST LOOP ENGINEERING

Test conditions
The purpose was to design a test loop that could carry out the official test scope consisting of two main scenarios:

1) Start-up test at high viscosity and minimum flow:

Motor was started DOL. Power supply was 4000 V, 60 Hz, 3ph, using portable power generators. For this test, oil viscosity was approx. 3075 cP at the start-up. Viscosity control was not required after starting.

2) Intermittent Duty point, reduced flow:

Motor was started DOL under the same conditions as for 1). For this performance test, the oil viscosity was controlled at about 1075 cP.

Additional performance tests
Internal tests with various viscosities from 480 cP up to 2057 cP were also performed in preparation for the official tests. Initially a performance test was conducted using water. The 60 Hz power generator was used during this test. Three additional performance tests at viscosities of 485 cP, 1085 cP and 1639 cP were conducted, but with lower rotational speed using the 50 Hz, 3 ph power source from the CETIM's facilities.

Test loop design
The most important design basis for the test loop was the highly viscous test fluid (more specifically: oil). Thus the pump was put in a closed test loop. The pump was submerged vertically in a caisson integrated in the closed test loop.
The loop, including control valves, flow-meter, heat exchanger and so forth was designed according to pump dimensions, flow and capacity. The test loop was equipped with instrumentation and auxiliary systems necessary for meeting the relevant parameter targets.

Cooling system selection
Designing the cooling system for a viscosity of 1075 cP, with a motor power consumption of 780 kW was the main challenge. During the performance test, the purpose was to maintain a stable oil temperature to keep a constant viscosity and comparable running conditions.
Water was selected as cooling fluid. CETIM has a 1,000 m^3 (353,147 ft^3) water basin at ambient temperature (typically 64.4°F/ 18°C). A plate heat exchanger was able to meet head losses and cooling requirements.
During testing, all the circulated oil went through the heat exchanger. The heat exchanger cooling water was pumped from the water basin at ambient temperature. The oil temperature was then controlled by adjusting the cooling water flow rate using a control valve.
The heat exchanger design would influence the selection of test oil significantly, because the temperature difference between the cargo oil and the cooling water had a significant influence on heat exchanger performance. Calculations were based on pump nominal flow rate.

Oil selection
A mineral oil was selected as it is a Newtonian fluid. For safety reasons, high oil temperatures were avoided. Two viscosity values were required for testing: 2057 cP and 1075 cP. The viscosity was controlled by the temperature of the fluid. It was necessary to have about 10°C (50°F) difference in the oil temperatures to achieve those two different viscosities. Also, the heat exchanger performance required an adequate temperature difference between tank oil and cooling water. This has given a temperature of about 28-30°C (82.4-86°F) for 2057 cP and about 38-40°C (100.4-104°F) for 1075 cP.
The ISO VG1000 selected oil had a viscosity of 2059 cP at 29.6°C (85.3°F) and 1075 cP at 38.1°C (100.6°F). When selecting the type of oil, it was also important to evaluate its density, especially for the start-up test, as required power is directly linked to density. At 2057 cP the specified crude oil density for field operation was 936.0 kg/m^3 (58.4 lb/ft^3, SG of 0.936). The ISO VG1000 density for 2057 cP was 892.0 kg/m^3 (55.7 lb/ft^3, SG of 0.892). Prior to the pump test, a specialized laboratory established the oil viscosity and density as function of temperature. A mineral oil with higher density and correct viscosities in this temperature range (28-40°C/82-104°F) has not been identified.
Table 2 presents the density and viscosity variation with temperature for the ISO VG 1000 oil that was selected.

Temperature		Density		Viscosity	
°C	°F	kg/m^3	lb/ft^3	cSt	cP
15.0	59.0	898.5	56.1	8477.6	7617.1
20.0	68.0	896.3	56.0	5267.8	4721.3
25.0	77.0	894.0	55.8	3381.2	3022.8
29.6	85.3	891.9	55.7	2308.6	2059.1
30.0	86.0	891.8	55.7	2235.7	1993.7
35.0	95.0	889.5	55.5	1519.2	1351.3
38.1	100.6	888.1	55.4	1210.9	1075.4
40.0	104.0	887.3	55.4	1058.5	939.2
100.0	212.0	860.3	53.7	54.9	47.2

Table 2: ISO VG 1000 oil characteristics

When the test loop design was frozen the required volume, including all auxiliaries, was 10 m^3 (353 ft^3).

Instrumentation and control

The following equipment was selected to meet the test requirements:

- A Coriolis flowmeter connected on the discharge line.
- Control valve(s) – on the discharge line –to adjust the discharged flow rate, hence establish the performance curve.
- A heat exchanger to control fluid temperature and maintain constant viscosity of 1075 cP during performance test.

The flowmeter selection was based on viscous flow rate (Q) up to 1300 m^3/h (4535 USgpm) at 1075 cP and the head loss generated, taking into account the pump total head of 95 m at 1300 m^3/h (311.7 ft at 5723 USgpm). The choice was a Coriolis flowmeter, ND250. The uncertainty associated with this measurement at the BEP is ±0.57%.
The control valve selection was driven by the head loss for the required viscosity (1075 & 2057 cP) and the maximum flow of 1300 m^3/h (5723 USgpm). Final choice was a ”Monovar” ND400 control valve. Other valves were also installed in the test loop. However these were only for isolation purposes in case of leakage or for dismantling parts in the test loop.
To measure the pump total head (H), pressure tappings were mounted at a distance of 2D from the discharge flange. The measuring sleeve had the same diameter as the pump discharge. Suction pressure was measured at the bottom of the caisson, near the suction bell. Total head was measured by means of pressure transducers fit for highly viscous fluid. The uncertainty associated with the total pump head measurement was lower than 1% at BEP. The head losses in the discharge column are small compared to the total head of 125 m (430 ft) from the pump.
Pump rotational speed (N) was measured with a proximity sensor. The pulses from the proximity sensor are counted by an electronic frequency meter with an uncertainty of 0.3%
Pump power input (P) was measured from motor power input and from motor efficiency for each operating point.
Electric motor power input was measured following IEC 60034 by a three phase power analyzer, using the three watt meters method. The power analyzer also provides voltage and frequency input measurements. Before the run test, the resistance of the supply cables and the stator resistance were measured. Also, a test of the motor under zero load was performed. Therefore, the total magnetic, mechanical losses and Joule’s heat losses were accounted for before start-up. Power measurement was corrected with a temperature measurement of the stator resistance. Therefore, the maximum uncertainty associated with the pump power input was about 2.5% giving an overall pump efficiency uncertainty of 2.69% at most.
To record the test fluid temperature and check the viscosity a temperature sensor was placed upstream of the test loop caisson.
The required voltage, frequency and power (4000 Volt, 60 Hz, 780 kW/1046 BHP) are supplied by a dedicated power generator to meet DOL start requirement for the motor. Due to the frequency used, generators and transformers were rented. The additional tests run at lower speed used the CETIM electrical facilities with a 50 Hz frequency and an input voltage of 3300V.

Test loop operating procedures and experience feedback

The following points have been taken into account in the test loop design.
Due to variation in ambient temperature affecting the test oil temperature and viscosity, pre-heating was evaluated. Low temperature/high viscosity would challenge the DOL start-up of the pump due to high power consumption and torque and high temperature/low viscosity would require cooling of the test oil that could be time-consuming.
Thus, a special auxiliary loop was installed to heat or to cool the oil in the main test loop.
When filling the test loop, ambient temperature was quite high and oil was naturally pre-heated to about 25°C (77°F). To increase oil temperature up to 29.6°C (85.3°F) or 38.1°C (100.6°F), the pump was run with 50 Hz, 3300V using CETIM electric facilities. The heat exchanger was switched off.
While running the pump the oil temperature increased from 25°C (77°F) to required temperature due to dissipated energy in the oil. This alternative method of increasing the temperature turned out to be quite effective. Furthermore, using a variable speed converter for pre-heating the oil would prevent the power consumption exceeding the motor power rating even for very low ambient temperatures/high viscosities, hence power consumptions. The planned auxiliary loop was not used because of ideal weather conditions.
Another important point was to have retention vats under the whole test loop in case of leakage.
During the design of the test loop, attention was paid to oil expansion with increasing temperature. An expansion tank was installed at an appropriate location in the test loop.
Last, but not least, all parts of the test loop and also the pump had to be cleaned for decommissioning. The more viscous the test oil, the harder it was to clean the test loop and equipment.
General arrangement drawings and pictures of the test loop are presented in Figures 3 to 6.

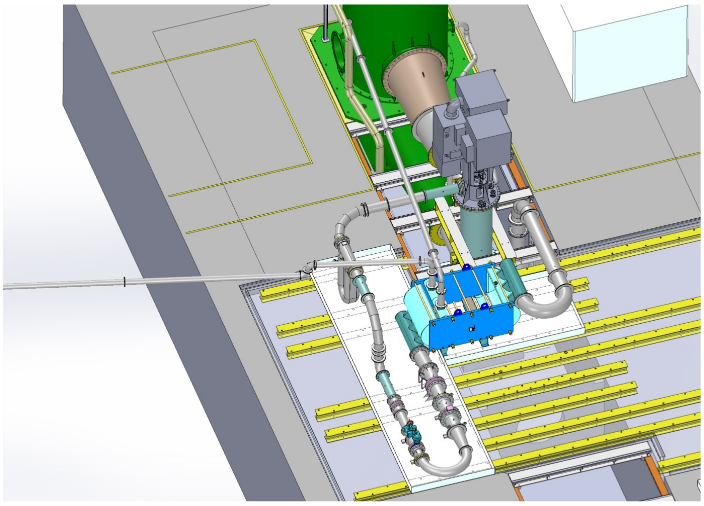

Figure 3: Test loop general arrangement set-up

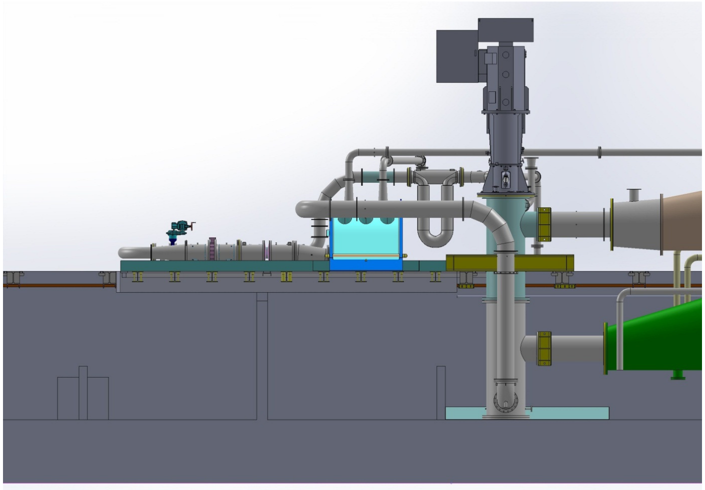

Figure 4: Test loop general arrangement set-up – side view

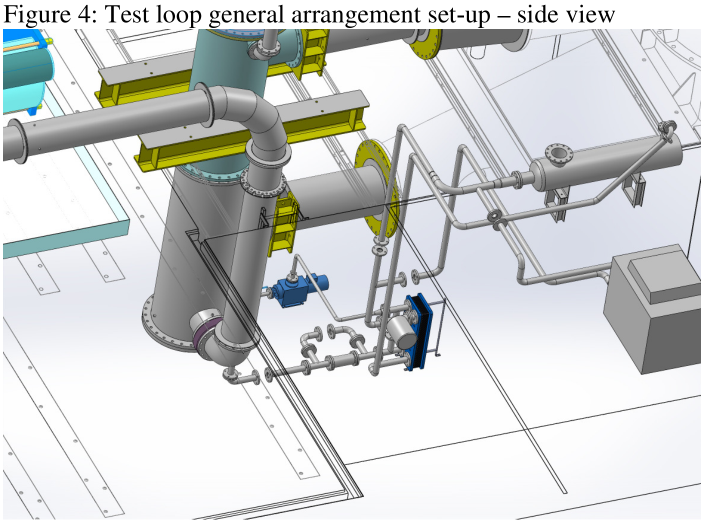

Figure 5: Auxiliary loop general arrangement set-up

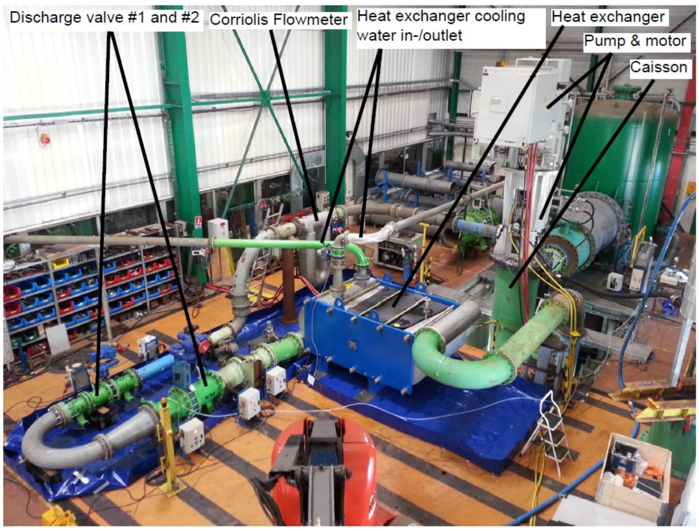

Figure 6: Test loop, by courtesy of EUREKA PUMPS AS - CETIM

TEST RESULTS

The test results are presented below. It should be noted that two different rotational speeds were used to test the pump and to evaluate the rotational speed effects on the results:

- 60 Hz leading to 1783 rpm
- 50 Hz leading to 1490 rpm

Test results presented below concern:

- Test performance curves of the pump with water at 60 Hz (Figure 7)

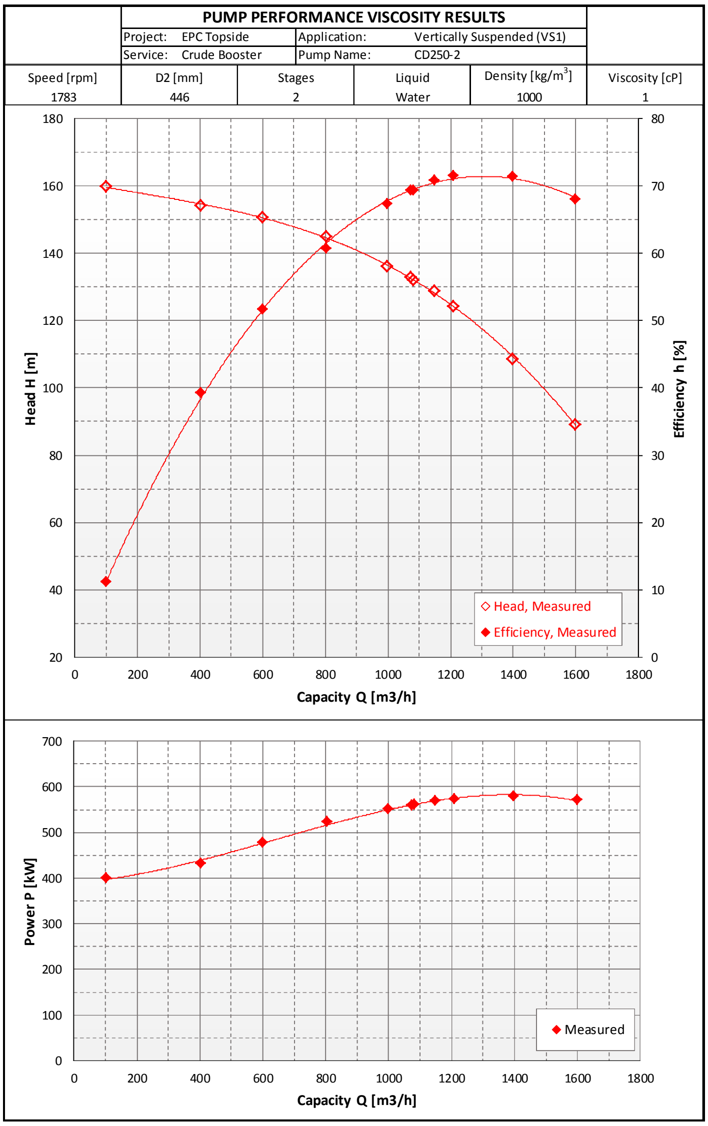

Figure 7: Test performance curves with water at 22°C, 1 cP at 60 Hz

Further results are presented below

- Cold start with oil (3075 cP and 55.56 lb/ft^3/ 890 kg/m^3 , SG of 0.890 as density) at 60 Hz (Figure8)

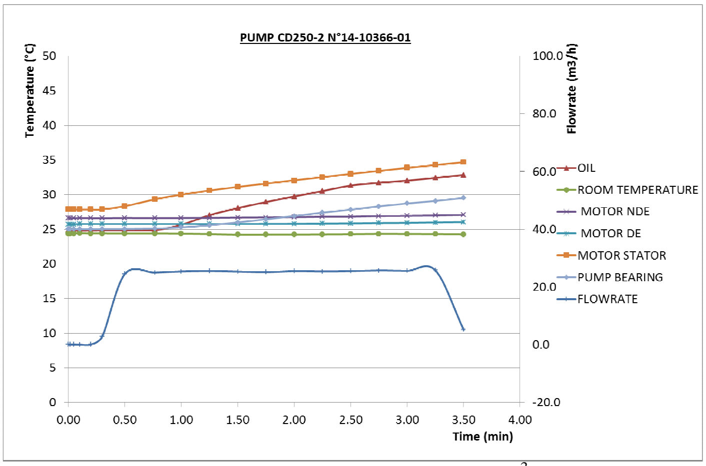

Figure 8: Cold start with oil at 3075 cP, 890 kg/m^3, 60Hz

The performance characteristics of the pump using various viscosities and rotational speed are then presented. Hydraulic Institute correction factors (ANSI/HI 9.6.7 (2010) (Figure 9) and HI 1.1-1.5 (1994)) apply to the pump tested as the calculated values of the B coefficient from HI is within the applicable range (1<B<40). The results from these corrections are plotted alongside the test results. The table below presents a summary of all tests performed including figures and tables presenting the results." (Please note that the table is not a summary of the results.)

Speed (rpm)/Freq (Hz)	Mean viscosity (cP)	Objective	Figure	Table
1783 rpm/ 60 Hz	1 cP	Power, H, η vs. Q	7	
1783 rpm/ 60 Hz	3075 cP	Cold start	8	
1783 rpm/ 60 Hz	1085.8 cP	Power, H, η vs. Q HI comparison	9	3
1490 rpm/ 50 Hz	1085.8 cP	Power, H, η vs. Q HI comparison	10	4
1490 rpm/ 50 Hz	1639.3 cP	Power, H, η vs. Q HI comparison	11	5
1490 rpm/ 50 Hz	485.5 cP	Power, H, η vs. Q HI comparison	12	6

In all tables, BEP was obtained using an interpolation curve, which is a polynomial equation fitted to the test points. The symbols used in the figures present actual measurement points for viscous fluid or HI viscosity corrected points based on water as test fluid represented in Figure 7.

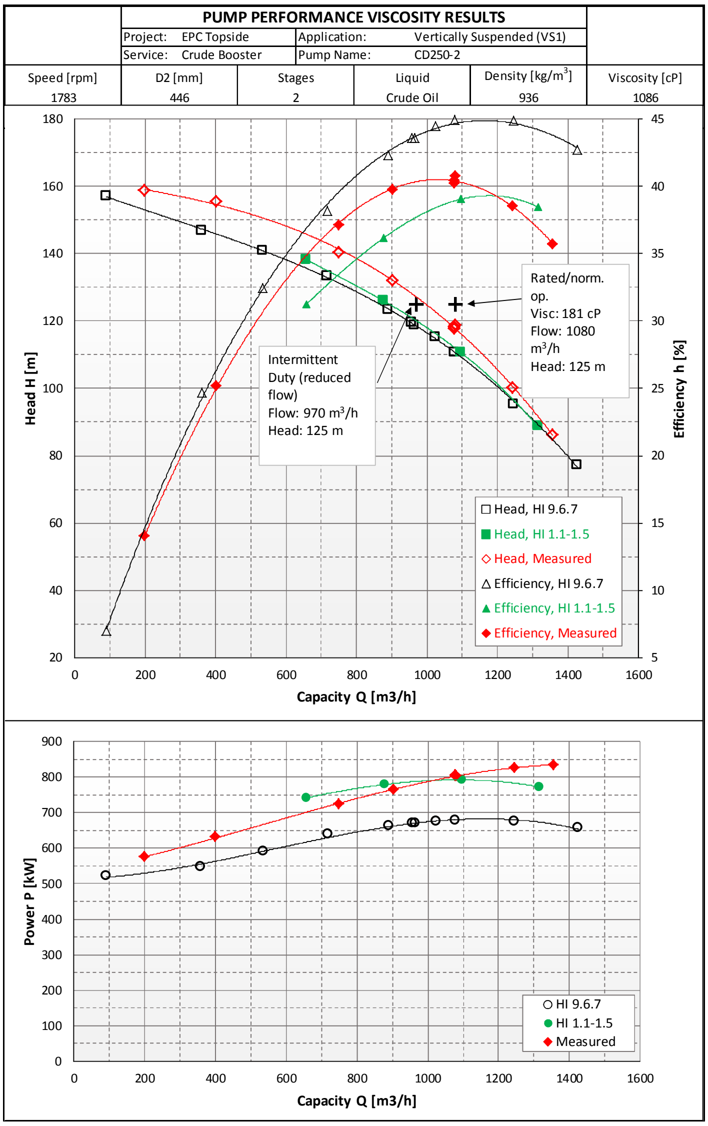

Figure 9: Test performance curves at 1085.8 cP-60 Hz compared with HI 9.6.7 (2010) and HI 1.1-1.5 (1994) predictions (by courtesy of EUREKA PUMPS AS – CETIM)

Test Result compared with HI Visc. Corr. Calc. at B.E.P				
Speed [rpm] 1783	Density [kg/m^3] 936	Density [lb/ft^3] 58.4		Viscosity [cP] 1085.8
SI units	Q_{Visc} [m^3/h]	H_{Visc} [m]	P_{Visc} [kW]	η_{Visc} [%]
Water (1000 kg/m^3)	1296	117	576	71.8
HI 1.1-1.5 (1994)	1167	104	789	39.3
ANSI/HI 9.6.7 (2010)	1155	104	681	45.1
Performance Test results (Ref. chart):	1020	123	790	40.7
US customary units	Q_{Visc} [gal/min]	H_{Visc} [ft]	P_{Visc} [BHP]	η_{Visc} [%]
Water (62,4 lb/ft^3)	5704	384	773	71.8
HI 1.1-1.5 (1994)	5139	342	1057	39.3
ANSI/HI 9.6.7 (2010)	5084	342	913	45.1
Performance Test results (Ref. chart):	4491	404	1059	40.7
Viscous Correction Coefficients:	C_Q [-]	C_H [-]	C_P [-]	C_η [-]
HI 1.1-1.5 (1994)	0.90	0.89	1.37	0.55
ANSI/HI 9.6.7 (2010)	0.89	0.89	1.18	0.63
Performance Test results (Ref. chart):	0.79	1.05	1.37	0.57
Ratio between Measured - & Calc. Values	Q_{Test}/Q_{Calc}	H_{Test}/H_{Calc}	P_{Test}/P_{Calc}	η_{Test}/η_{Calc}
HI 1.1-1.5 (1994)	0.87	1.18	1.00	1.04
ANSI/HI 9.6.7 (2010)	0.88	1.18	1.16	0.90

Table 3 Associated to Figure 9

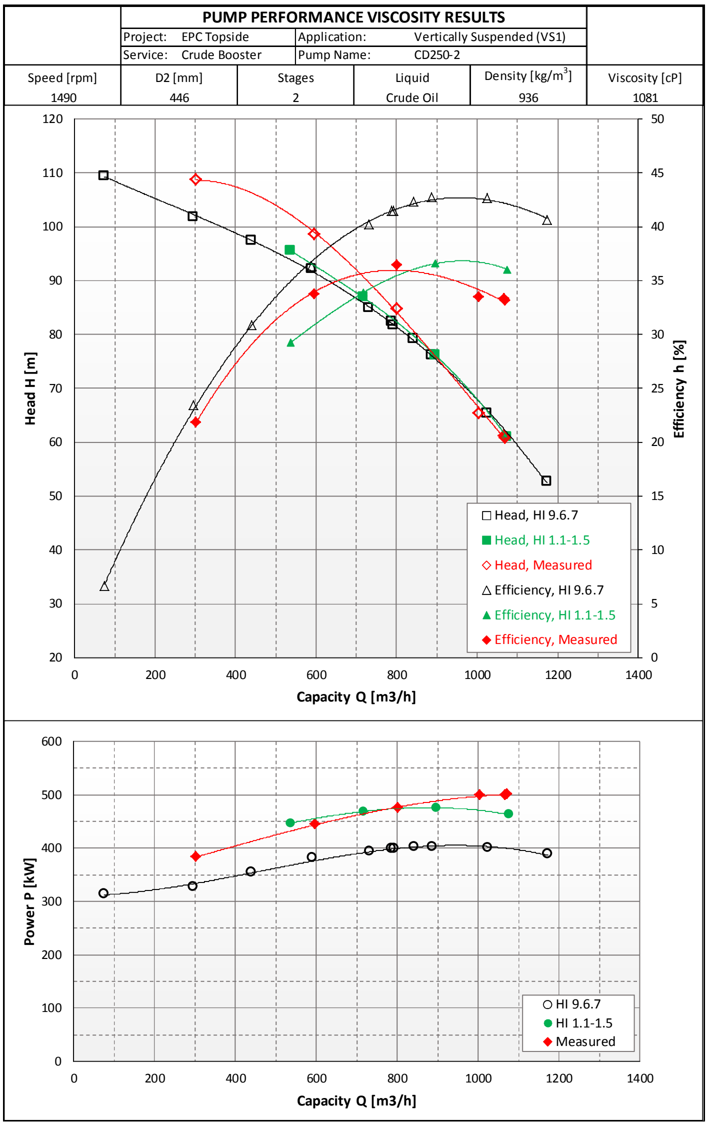

Figure 10: Test performance curves at 1081 cP-50 Hz compared with HI 9.6.7-2010 and HI 1.1-1.5 (1994) predictions (by courtesy of EUREKA PUMPS AS – CETIM)

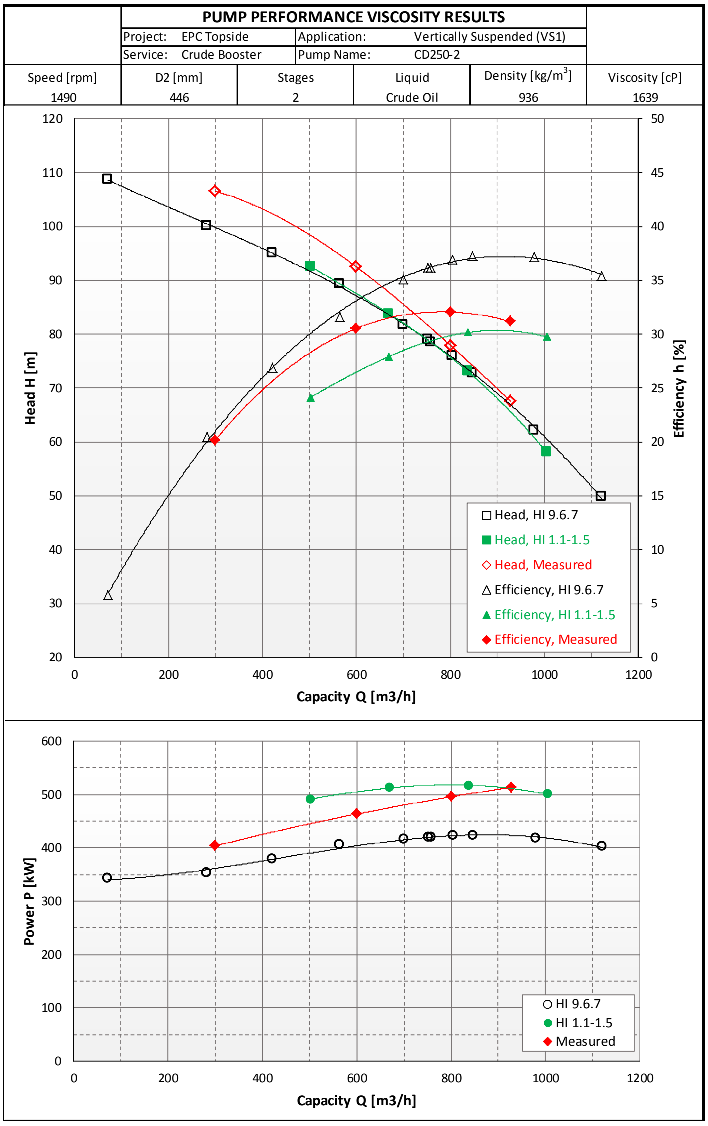

Figure 11: Test performance curves at 1639 cP-50 Hz compared with HI 9.6.7-2010 and HI 1.1-1.5 (1994) predictions (by courtesy of EUREKA PUMPS AS – CETIM)

Test Result compared with HI Visc. Corr. Calc. at B.E.P				
Speed [rpm] 1490	Density [kg/m³] 936	Density [lb/ft³] 58.4		Viscosity [cP] 1081.3
SI units	Q_{Visc} [m³/h]	H_{Visc} [m]	P_{Visc} [kW]	η_{Visc} [%]
Water (1000 kg/m³)	1083	82	336	71.8
HI 1.1-1.5 (1994)	954	72	473	36.8
ANSI/HI 9.6.7 (2010)	949	72	405	42.9
Performance Test results (Ref. chart):	795	85	472	36.4
US customary units	Q_{Visc} [gal/min]	H_{Visc} [ft]	P_{Visc} [BHP]	η_{Visc} [%]
Water (62,4 lb/ft³)	4767	268	451	71.8
HI 1.1-1.5 (1994)	4202	235	635	36.8
ANSI/HI 9.6.7 (2010)	4180	235	543	42.9
Performance Test results (Ref. chart):	3500	279	633	36.4
Viscous Correction Coefficients:	C_Q [-]	C_H [-]	C_P [-]	C_η [-]
HI 1.1-1.5 (1994)	0.88	0.88	1.41	0.51
ANSI/HI 9.6.7 (2010)	0.88	0.88	1.20	0.60
Performance Test Results:	0.73	1.04	1.40	0.51
Ratio between Measured - & Calc. Values	Q_{Test}/Q_{Calc}	H_{Test}/H_{Calc}	P_{Test}/P_{Calc}	η_{Test}/η_{Calc}
HI 1.1-1.5 (1994)	0.83	1.19	1.00	0.99
ANSI/HI 9.6.7 (2010)	0.84	1.18	1.17	0.85

Table 4 Associated to Figure 10

Test Result compared with HI Visc. Corr. Calc. at B.E.P				
Speed [rpm] 1490	Density [kg/m³] 936	Density [lb/ft³] 58.4		Viscosity [cP] 1639.3
SI units	Q_{Visc} [m³/h]	H_{Visc} [m]	P_{Visc} [kW]	η_{Visc} [%]
Water (1000 kg/m³)	1083	82	336	71.8
HI 1.1-1.5 (1994)	903	68	513	30.4
ANSI/HI 9.6.7 (2010)	908	69	425	37.4
Performance Test results (Ref. chart):	765	80	490	32.1
US customary units	Q_{Visc} [gal/min]	H_{Visc} [ft]	P_{Visc} [BHP]	η_{Visc} [%]
Water (62,4 lb/ft³)	4767	268	451	71.8
HI 1.1-1.5 (1994)	3974	222	687	30.4
ANSI/HI 9.6.7 (2010)	3998	225	569	37.4
Performance Test results (Ref. chart):	3368	262	657	32.1
Viscous Correction Coefficients:	C_Q [-]	C_H [-]	C_P [-]	C_η [-]
HI 1.1-1.5 (1994)	0.83	0.83	1.52	0.42
ANSI/HI 9.6.7 (2010)	0.84	0.84	1.26	0.52
Performance Test Results:	0.71	0.98	1.46	0.45
Ratio between Measured - & Calc. Values	Q_{Test}/Q_{Calc}	H_{Test}/H_{Calc}	P_{Test}/P_{Calc}	η_{Test}/η_{Calc}
HI 1.1-1.5 (1994)	0.85	1.18	0.96	1.05
ANSI/HI 9.6.7 (2010)	0.84	1.17	1.15	0.86

Table 5 Associated to Figure 11

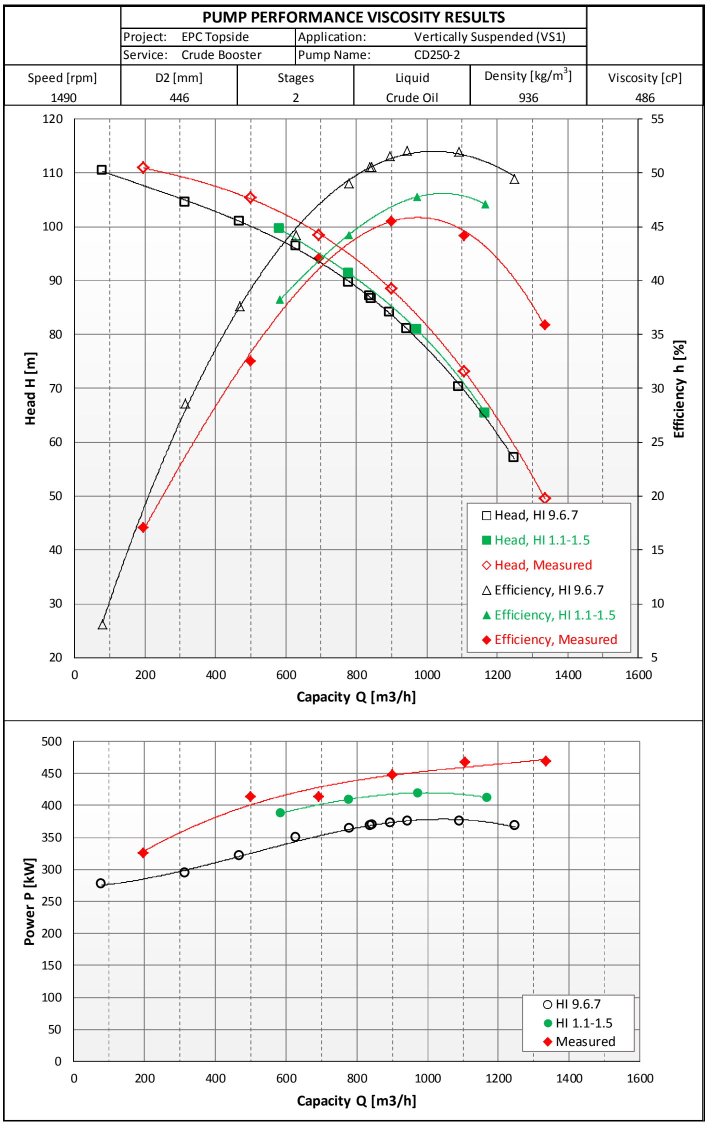

Figure 12: Test performance curves at 486 cP-50 Hz compared with HI 9.6.7-2010 and HI 1.1-1.5 (1994) predictions (by courtesy of EUREKA PUMPS AS – CETIM)

Test Result compared with HI Visc. Corr. Calc. at B.E.P				
Speed [rpm] 1490	Density [kg/m³] 936	Density [lb/ft³] 58.4		Viscosity [cP] 485.5
SI units	Q_{Visc} [m³/h]	H_{Visc} [m]	P_{Visc} [kW]	η_{Visc} [%]
Water (1000 kg/m³)	1083	82	336	71.8
HI 1.1-1.5 (1994)	1036	76	419	48.1
ANSI/HI 9.6.7 (2010)	1011	76	377	52.3
Performance Test results (Ref. chart):	975	84	452	45.5
US customary units	Q_{Visc} [gal/min]	H_{Visc} [ft]	P_{Visc} [BHP]	η_{Visc} [%]
Water (62,4 lb/ft³)	4767	268	451	71.8
HI 1.1-1.5 (1994)	4561	250	562	48.1
ANSI/HI 9.6.7 (2010)	4451	251	505	52.3
Performance Test results (Ref. chart):	4293	274	606	45.5
Viscous Correction Coefficients:	C_Q [-]	C_H [-]	C_P [-]	C_η [-]
HI 1.1-1.5 (1994)	0.96	0.93	1.25	0.67
ANSI/HI 9.6.7 (2010)	0.93	0.93	1.12	0.73
Performance Test results:	0.90	1.02	1.34	0.63
Ratio between Measured - & Calc. Values	Q_{Test}/Q_{Calc}	H_{Test}/H_{Calc}	P_{Test}/P_{Calc}	η_{Test}/η_{Calc}
HI 1.1-1.5 (1994)	0.94	1.10	1.08	0.95
ANSI/HI 9.6.7 (2010)	0.96	1.09	1.20	0.87

Table 6 Associated to Figure 12

To summarize these test results, Figure 13 is plotted. It gives a comparison at BEP between measured values and HI predictions according to ANSI/HI 9.6.7 (2010) and ANSI/HI 1.1-1.5 (1994) as a function of the viscosity and for a speed of 1490 rpm.

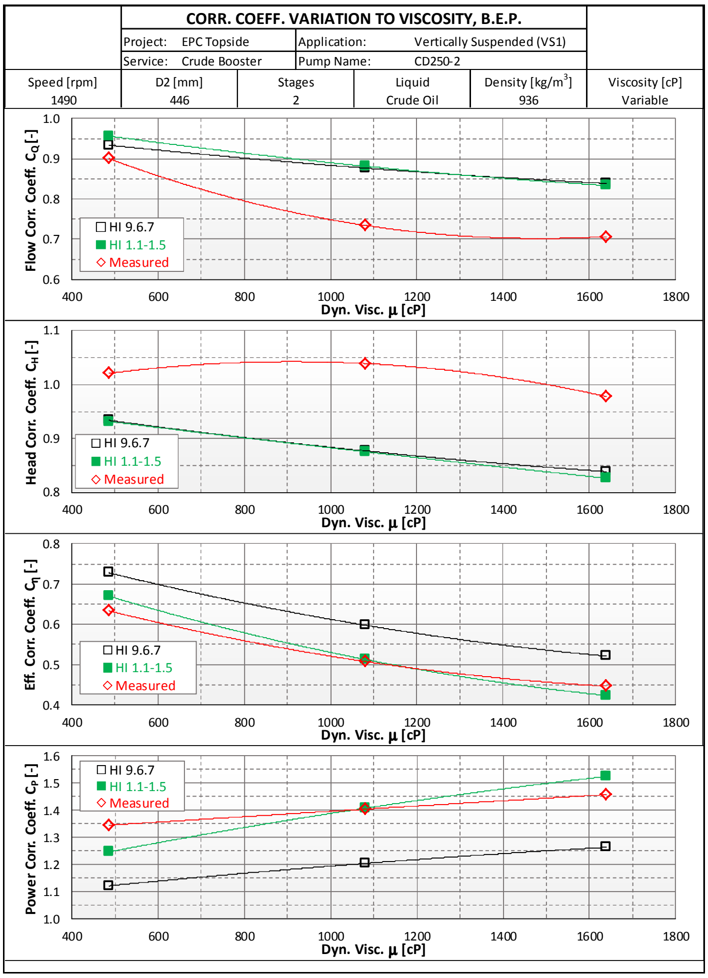

Figure 13: Deviation between Test Results and ANSI/HI 9.6.7 (2010) & 1.1-1.5 (1994). Predictions at BEP (100% being agreement between tests and predictions) (by courtesy of EUREKA PUMPS AS – CETIM)

DISCUSSION

Intermittent start-up (2057 cP) test:

Naturally, no performance curve exists as this test is a start-up at minimum flow. However, Figure 8 presents the actual values recorded for the 4 minutes test. The most important point to notice for this test is the actual start-up viscosity of 3075 cP. This is significantly higher than the actual requirement of 2057 cP. As stated in the "Pump & Motor" section, the rated torque for this motor was eventually no concern as it is significantly higher than the calculated pump torque. This is also demonstrated by the successful start-up test at 3075 cP.

Intermittent Duty point, Reduced flow (1075 cP) test:
With a required head of 125 m (410 ft), the read off flow from the test results in Figure 9 is actually 970 m^3/h (4271 USgpm). This is 14% higher that the estimated value of 850 m^3/h (3742 USgpm) from the predicted pump characteristics in Figure 2. This is also indicated in Table 3 when comparing the estimated correction coefficient values (B.E.P.) for flow and head with the actual ones.

An important observation in Figure 9 is the actually measured power consumption of 770 kW (1033 BHP), just below the rated motor power of 780 kW (1046 BHP) and significantly higher than the predicted power from ANSI/HI 9.6.7 (2010). It is therefore reason to believe the specified rated motor power would be insufficient if based on predictions from ANSI/HI 9.6.7 (2010) or a loss analysis based on J.F. Gülich, see Figure 2. Predicted power from ANSI/HI 1.1-1.5 (1994) is quite accurate compared to the real power consumption.

Test results comparison with HI prediction:

- A clear observation in Figures 9-12 is the over-prediction of efficiency (or under-prediction of power) for ANSI/HI 9.6.7 (2010). The same consistency is observed for the flow and head that is over- and under- predicted respectively. It results in a flow/head curve below the actual measured curve and an efficiency curve with B.E.P located at higher flow than the actual measured curve. This observation is for both ANSI/HI 9.6.7 (2010) & 1.1-1.5 (1994) and the flow/head curve prediction is almost identical as well. This is also presented by the correction coefficients in Tables 3-6. Furthermore, the very same head and flow prediction is observed for the loss analysis based on J.F. Gülich in Figure 2. However, this loss analysis prediction is limited to a viscosity of 1075 cP only. Hence it is unknown if consistent for other viscosities. ANSI/HI 1.1-1.5 (1994) is the oldest prediction method used for comparison, but with a far smaller power/efficiency deviation than ANSI/HI 9.6.7 (2010) and Gülich, particularly for viscosity of 1075/1085/1081 cP, see Figures 2, 9-12 and Tables 1, 3-6. For the real B.E.P flow in Figures 9-10, the power consumption is identical. Furthermore, the B.E.P C_p in Tables 3 & 4 is also identical when compared to the real C_p based on measured results. Interestingly, these observations are identical for both 50- and 60 Hz rot. speed and viscosity of 1085/1081 cP, cf. Figures 9 & 10. Hence, for this particular case it indicates an accurate power prediction when rotational speed varies.

- Figure 13 presents the same deviations between the predicted and measured flow characteristics as described above but for varying viscosity values. An interesting observation is the C_H and C_Q increasing accuracy with decreasing viscosity when compared to measured values. However, this is only the case for the viscosity range 485-1081 cP. Above 1081 cP the deviation seem to be approx. constant. The C_H calculated from the measured values are: C_H(486 cP) = 1.02 and C_H(1081 cP) = 1.04, see Table 4 and Table 6 in conjunction with Figure 13. Surprisingly the measured C_H values are above 1 at viscosity equal to- or less than 1081 cP. An explanation for this effect could be the reduction in B.E.P flow values when corrected for viscous flow. If this is significant and in combination with low viscous head loss the value $C_H = H_{Visc}/ H_{Water}$ can be greater than 1.

- The C_P and C_η from ANSI/HI 9.6.7 (2010) follow the same curve slope when compared to the measured values. ANSI/HI 1.1-1.5 (1994) does not follow the same gradient, but crosses at viscosity of 1081 cP having no deviation at this point compared to the measured value, which was fortunate when deciding the motor rating. Further, the power is under-predicted at viscosity 486 cP and over-predicted at 1639 cP. The deviation is also larger at viscosity 486 cP. The opposite would be expected as the prediction seems to be more inaccurate with higher viscosity.

- To emphasize, the test results are from an EPC project and are therefore limited. Hence, further work and additional experimental data is highly recommended in order to confirm and explain the above observations, especially for the discrepancies between the measured- and predicted pump characteristics.

CONCLUSIONS

A full scale high viscosity pump performance test was completed for a 2-stage vertically suspended (VS1) API610 pump. The test scope was part of an ongoing EPC project and exclusively conducted for industrial purposes. Test data presented is for both 50- and 60 Hz rotational speed and for a viscosity range of 1 to 3075 cP. The objective was to simulate real field operation with crude oil and to qualify the pump for the given field conditions and requirements in order to minimize field risks and associated costs. In addition, the laboratory tests were conducted due to published methods uncertainties and limited open literature concerning pump performance in viscous oils.

Pump performance viscosity predictions from HI and Gülich are also presented and compared with test results. The aim was to evaluate alternative future approaches in order to improve prediction uncertainties and ultimately reduce the number of tests/test scope in future projects. The comparison is simply for discussion within the industry and not for validation of published methods of pump performance predictions with viscous fluids. Comparison with HI and Gülich's loss analysis shows discrepancies for all performance parameters consisting of flow, head and power/efficiency. A major observation from this paper is the scatter between the power consumption observed in the test results and the results from loss analyses or HI prediction from 1994 and 2010. The HI correction from

1994 was selected because it was the most conservative power correction when rating the electrical motor.

Viscous performance tests will allow optimizing the selection of process equipment, especially for motor sizing for which ANSI/HI 9.6.7 (2010) predictions seems to under-predict the required power. Further tests regarding variation of viscosity, range of flow rate and a systematic variation of the stage number would be required to analyze more precisely the observed discrepancies. In cases where installation cost, maintenance cost and loss of production are critical in financial terms and where the uncertainties regarding performance predictions are high, realistic laboratory testing with real fluids, real operating points and specific design of the pump may be necessary to mitigate the potential risks.

To conclude, the testing of the pump performance under realistic conditions was positive. The design of the test loop provided adequate conditions regarding viscosity of fluids to evaluate the pump performance. Both official tests (start-up and intermediate duty) were successful. Important experience regarding building/designing the test stand and the execution of the tests was gained.

NOMENCLATURE

B	=Parameter used in the HI viscosity corrections [-]	
BEP	= Best Efficiency Point	
D_2	= Impeller outlet diameter	ft /m
DOL	= Direct On Line	
EPC	= Engineering, Procurement, Construction	
H	= Total head	ft /m
H_{Visc}	= Viscous total head	ft /m
n_q	= Specific speed $n_q = N\,Q^{0.5}/\,H^{0.75}$	[-]
N	= Shaft speed	
P	= Pump shaft input power	BHP /kW
P_{Visc}	= Shaft input power with viscous fluid	BHP /kW
$P_{w\,Inp}$	= Pump shaft input power with water	BHP /kW
Q	= Flow rate	USgpm /m^3/h
Q_{Visc}	= Flow rate with viscous fluid	USgpm /m^3/h
Re	= Reynolds number $Re=\omega r_2^2/\nu$	[-]
r_2	= Impeller outlet radius	ft /m
SG	= Specific gravity	
η	= Pump efficiency	[-]
η_{Visc}	= Pump efficiency with viscous fluid	[-]
μ	= Dynamic viscosity	cP
ω	= Angular velocity of shaft	rpm

REFERENCES

Hydraulic Institute, ANSI/HI 9.6.7-2010, *Effects of Liquid Viscosity on Rotodynamic (Centrifugal and Vertical) Pump Performance*

ANSI/ Hydraulic Institute, 1.1-1.5, 1994, *Centrifugal Pump Design and Applications*

J.F. Gülich, 2014, *Centrifugal Pumps*, 2nd Ed, New York, Springer-Verlag

IEC 60034, 2014, *International standard on rotating electrical machine,*

ACKNOWLEDGEMENTS

We would like to thank our customers and end user together with EUREKA PUMPS AS and CETIM teams who have worked on this project. Thanks also to Simen Holler (EUREKA PUMPS), Rune Osmundsen (EUREKA PUMPS), Arne Nilsen (EUREKA PUMPS) and all the suppliers for their support.