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EFFECT OF VISCOSITY AND TWO PHASE LIQUID – GAS FLUIDS ON THE PERFORMANCE OF MULTI-STAGE CENTRIFUGAL PUMPS

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ABSTRACT
The paper reports on developmental research on the effects of viscosity and two phases, liquid – gas fluids on ESPs which are multi stage centrifugal pumps for deep bore holes.

The test facility work was performed using pumps with ten or more stages moving fluids with viscosity from 2 to 2500 cP at various speed, intake pressure and Gas Void Fractions (GVF). For safety considerations the injected gas was restricted to nitrogen or air.

The results are a series of curves representing the performance degradation of the pump. Note that in some cases the pump performances actually improved with increasing viscosity. The resulting information will allow a better understanding and more accurate prediction of performance than has been previously available. The data indicates a significant difference in performance correction when compared to the information available from the Hydraulics Institute.

INTRODUCTION

Electrical Submersible Pumps (ESP) is a multistage pump in several sections, each section having up to 300 stages, running at variable speed from 2300 to 4100 rpm (40 to 70 Hz). Radial and mixed flow centrifugal pump stage designs are generally used in ESP. ESP is used in water well and downhole applications for lifting fluids from several hundred to few thousand feet depth. Due to well diameter constrains, ESP outside diameter ranges from 86mm to 260mm (3.38" to 10.25").

Centrifugal pumps have been used to pump viscous fluid for more than a century. The friction loss resulting from the fluid viscosity degrades the ideal performance of a pump. As the viscosity is increased, the maximum flow capacity of the pump is reduced. The location best efficiency point (BEP) on the performance curves shifts to the left and the required brake horsepower (BHP) increases. In some cases with small increases in viscosity, the performance of the pump may appear to increase. This improvement is due to reduction of leakage
losses within the pump stages offsetting the performance degradation due to the friction losses. (Fig 1 & 2)

VISCOSITY CORRECTION FACTORS FOR SINGLE PHASE FLUID

Stepanoff [1] and Ippen [2] were some of the earlier authors to understand and to analyze the effect of viscosity on centrifugal pump performance. Stepanoff [1] concluded that (a) "The affinity laws hold for all viscosities with less accuracy than those for water. Usually efficiency is better at higher speed as horsepower increases less than the cube of the speed", and (b) "At constant speed and variable viscosity, the head-capacity (flow rate) decreases as the viscosity increases, but the head at zero capacity (flow rate) remains essentially the same."

Since then, several methods [3–4], based on experimental test data, were developed to predict the performance of the pump handling viscous fluid from water performance characteristics. Validity of the results was limited to the test data range and type of the pump tested.

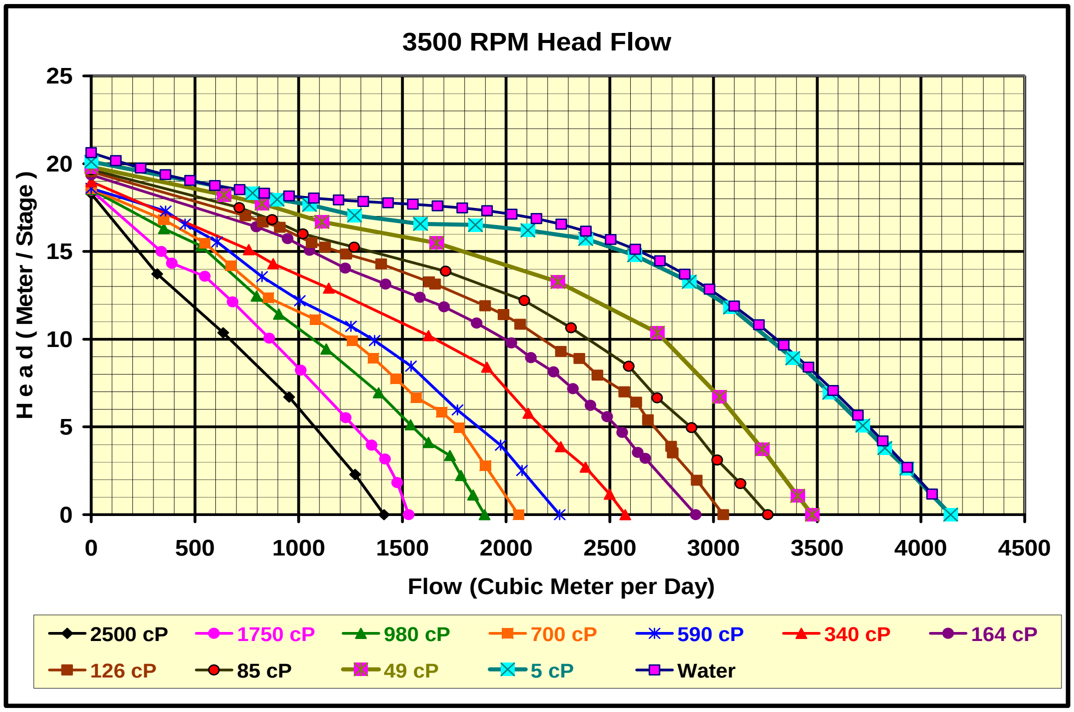

Fig.1, Head Flow data of a Pump at 3500 RPM and various viscosities

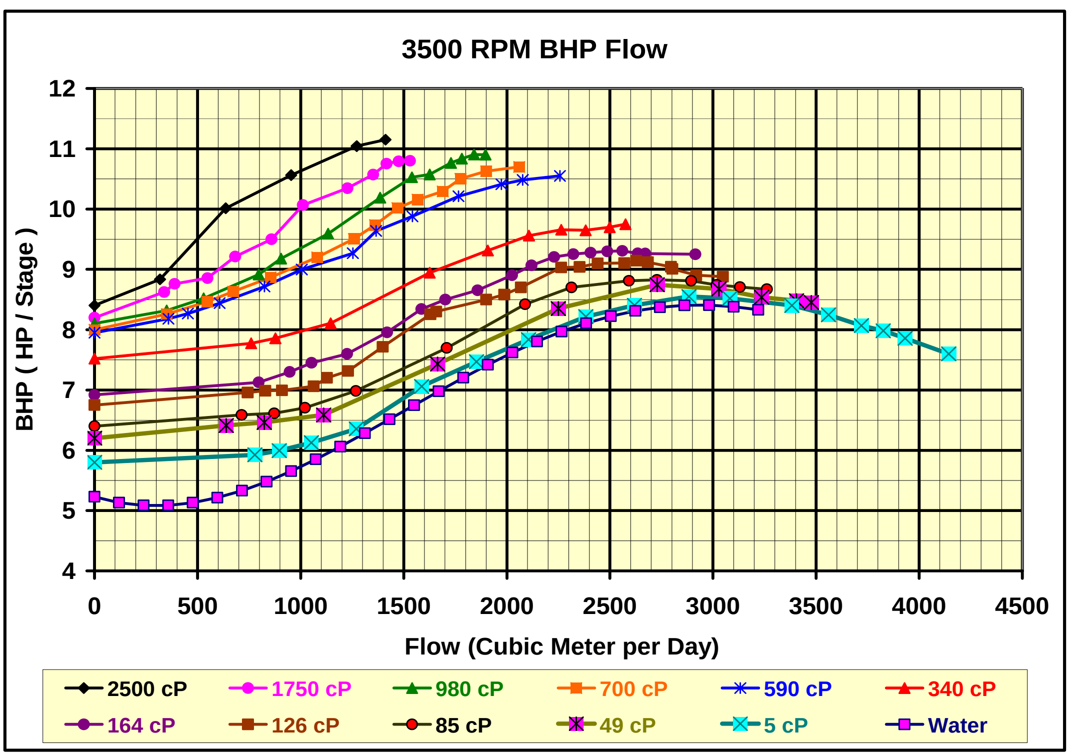

Fig.2, BHP Flow data of a Pump at 3500 RPM and various viscosities

The Hydraulic Institute [5-6] developed a method, for the prediction of performance of the pump in viscous fluid having viscosity up to 2200 cSt, based on test data of more than 10,000 pumps. The effect of speed on the viscous performance was not considered in this method. Correction factors were developed for Flow (Cq), Head (Ch), Efficiency (Cn) and BHP (Cbhp), from actual test data and then used for calculating viscous flow as follows:

$$Q_{vis} = C_q \bullet Q_w \quad \textbf{(Eq. 1)}$$

$$H_{vis} = C_h . \bullet H_w \quad \textbf{(Eq. 2)}$$

$$\eta_{vis} = C_n \bullet \eta_w \quad \textbf{(Eq. 3)}$$

$$bhp_{vis} = \frac{Q_{vis} \bullet H_{vis} \bullet S}{Const.(3960) \bullet \eta_{vis}} \quad \textbf{(Eq. 4)}$$

or

$$bhp_{vis} = C_{bhp} \bullet bhp_w \quad \textbf{(Eq. 5)}$$

The specific speed is defined as:

$$N_s = \omega \bullet \frac{Q_w^{1/2}}{H_w^{3/4}} \quad \textbf{(Eq. 6)}$$

In 2004, the Hydraulic Institute [6] published a revised method for calculation of correction factors, which also includes the speed effect. The database consists of mostly single stage volute pumps in a specific speed range from 60 to 3000 and viscosity up to 3000 cSt.

Comparisons of this method with the actual test data of viscous flow performance of multistage ESP pumps, showed an error margin of 30 to 40%. The error was larger at the lower flow rates than at BEP flow rate. The reason for the inaccuracy may be that the pumps used were mostly single stage volute pumps running at lower rpm.

Caicedo [7] has concluded that the Hydraulic Institute correlation will not work for ESP viscous pump performance and interpolation of actual test data should be used for performance prediction.

Gulich [8] has suggested another approach based on loss analysis. Gulich added a few more details in Stepanoff's [1] conclusions (c) "Mixing losses at the impeller and diffuser inlet and exit are often considered as little dependent on the Reynolds Number; (d) Disk friction losses grow with decreasing Reynolds Number or increasing viscosity; and (e) mechanical losses are essentially independent from the viscosity of the fluid pumped." Gullich concluded that disk friction and friction losses are the main cause for the performance deterioration by the viscous fluid.

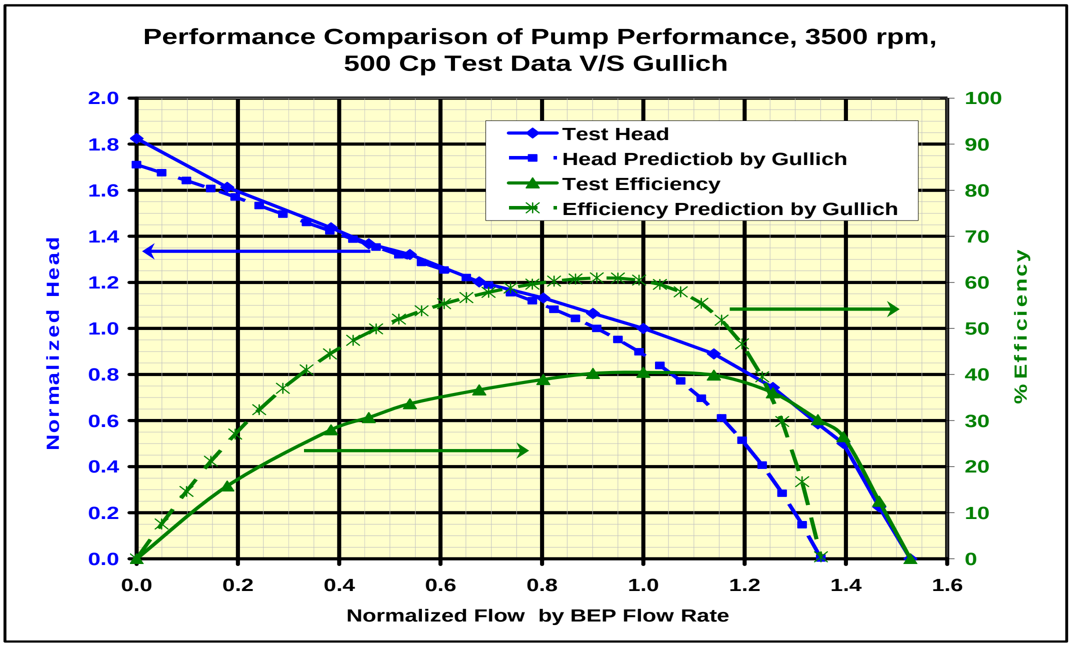

Fig.3, Comparison of Gullich method with actual test data of Multistage ESP pump

His analysis did not consider the effect of viscous fluid on volumetric losses through running clearance. Flow correction factors derived from this method do not agree with the flow correction factors derived for an ESP multi stage pump test data. Comparison of test data and prediction by Gulich are shown in figure 3. Gulich has used data for a single stage pump having both open and closed impellers, a specific speed range of 300 to 2500 and viscosity up to 4300 cSt.

VISCOSITY CORRECTION FACTORS FOR SINGLE PHASE FLUID & ESP

Electrical Submersible Pumps (ESP) have been used for lifting fluids out of the oil reservoir since 1930 [9 -12]. ESP is a multistage pump in several sections, each section having up to 300 stages, running at variable speed of 2300 to 4100 rpm (40 to 70 Hz).

Bearden [9] briefly noted the modifications required of an ESP for harsh oil application. Patterson [10] conducted a full scale test of an ESP running in viscous fluids up to 225 cSt in controlled conditions and concluded that an ESP can handle viscous fluid with reasonable performance. Carpenter [12] reported ESPs having longer run life in oil gravity ranges from 10˚ to 19˚ API in the Beta Field, offshore Long Beach.

Initially, ESP manufacturers used the Hydraulic Institute method for the performance prediction based on Best Efficiency Point (BEP) flow rate. Later on in the 1970s, manufacturers developed their own correction factors from actual test data for viscous fluid performance of their pumps. Correction factors for viscous performance were limited to 800 cSt at 3500 rpm.

As world demand for oil and gas increases and as new fields are discovered with viscosities higher than 1000 cSt, correction factors for higher viscosity in two phase flow conditions at various speeds and across the flow range (not limited to BEP flow rate) are required [13–20]. Moreover, ESPs used for higher flow rates have specific speeds higher than 3000.

TESTING

Nineteen multi-stage ESP pumps having an outside diameter of 100mm to 260mm (4 inches to 10.25 inches) and a flow range up to 10000 m^3/day (1900 gpm) were tested in the BHI Centrilift viscosity test loop at various speeds (2625 to 4375 rpm) and up to 4000 cSt; performance data was collected. Eight stages were radial flow type and eleven stages were mixed flow type.

Test loop picture is shown in figure 4. Schematic diagram is shown in annex A for more details. The Viscosity test loop was designed and built in 2004. It utilizes pneumatic and motorized flow control, Coriolis flow metering, and variable speed drive control. Fluid is delivered to the test pump from one of four storage tanks containing fluid having a viscosity range from 2 to 2500 cP by means of a separate boost pump to maintain a constant intake pressure. A heat exchanger is located within the system to control the temperature / viscosity.

Flow meters used were Micro Motion Coriolis ELITE series, with +/-0.05% mass and volume flow accuracy; +/- .35% gas flow accuracy; +/-0.0002g/cc density accuracy.

- (1) 1 inch meter 1000 lbs/min Max. Flow
- (2) 2 inch meters 3200 lbs/min Max. Flow
- (3) 3 inch meter 10,000 lbs/min Max Flow

Pressure Transmitters used were Honeywell Sensotec Absolute (7 ea) Model 440 transmitters, 0.25% Accuracy, 0 to 50, 0 to 500 psig, & 0 to 2000 psig.

Horse power and RPM were measured using Lebow torque sensor having accuracy of 0.25%.

Fisher Flow Control valves were used for controlling flow.

Once a targeted single phase (liquid) flow rate, intake pressure, RPM, temperature etc. at the test pump is established and stable for a period of time, gas is injected into the liquid near the intake of the pump at a specified flow rate.

Fig.4, Aerial Picture of test facility

Numerous tests were preformed varying pressures and temperatures, flow, rpm, and torque. Data was recorded using automated data collection system. The data was then analyzed and corrected for speed.

RADIAL VS. MIXED FLOW

Higher viscosities aid a centrifugal pump ability to handle free gas. Free gas in any liquid is detrimental to the centrifugal pump performance. The gas tends to collect in pockets on the low pressures sides of the vane and in the impeller eye. This will interfere with and eventually block the flow. In radial designed pump stages, all of the fluid is required to flow in a radial direction. The gas flow will lag the liquid and attempt to remain in the impeller. In mixed flow or axial flow impellers radial component is greatly reduced and the gas can more easily be carried by the drag forces with the liquid. Higher viscosity fluid will show significantly less gas degradation in the axial and mixed flow pump designs than it would in the radial designs.

Effect of Speed - For simplification, it is assumed that speed affects flow correction factors only, based on water performance data. It is not very accurate in viscous flow applications. A correlation for an ESP pump, was developed for speed and flow data at various viscosities as shown in Fig. 5. Data is for radial flow pump, 400P3, specific speed 16 (US Specific speed. 807). Speed effect and flow correction factors are separate for each pump.

As speed increases, the Reynolds number increases and the friction factor moves from laminar to transition to turbulent region, resulting in lower friction and smaller flow correction factors.

Also efficiency of the pump increases with the increase in the operating speed as shown in figure 6. This is based on test data of WNE1600 pump, specific speed 80 (US Specific speed. 4122). Explanation is provided by Stepanoff [1] and Gullich [8] and is explained in earlier sections.

Effect of Viscosity - Figures 7 to 10 shows the effect of viscosity on two phase flow pump performance of WJE1200 32 stage pump, specific speed 55 (US Specific speed. 2829). Figures 7 & 8 shows head flow characteristics and figures 9 & 10 shows BHP flow characteristics. Pump was tested up to 69 Bar intake pressure, at 2917, 3500 & 4083 rpm, (50, 60 & 70 Hz respectively.) up to 2300 cP viscosity and up to 60% GVF. Fluid mixture consisted of synthetic oil and nitrogen.

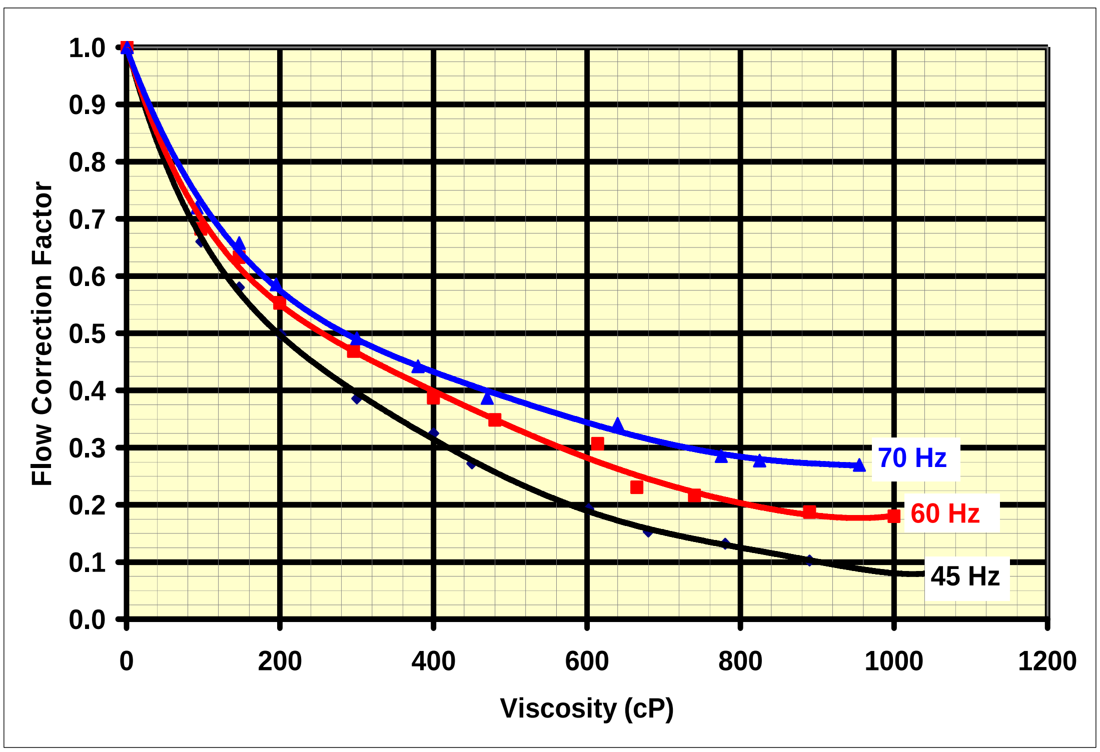

Fig. 5, Effect of speed on flow correction factors

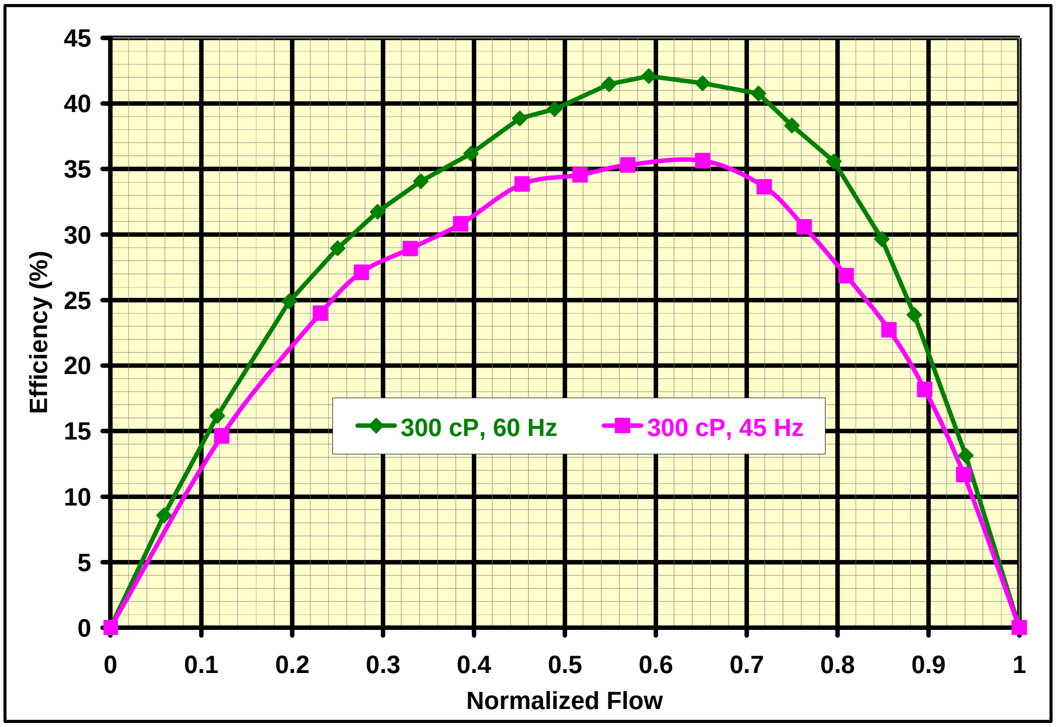

Fig. 6, Effect of speed on Efficiency

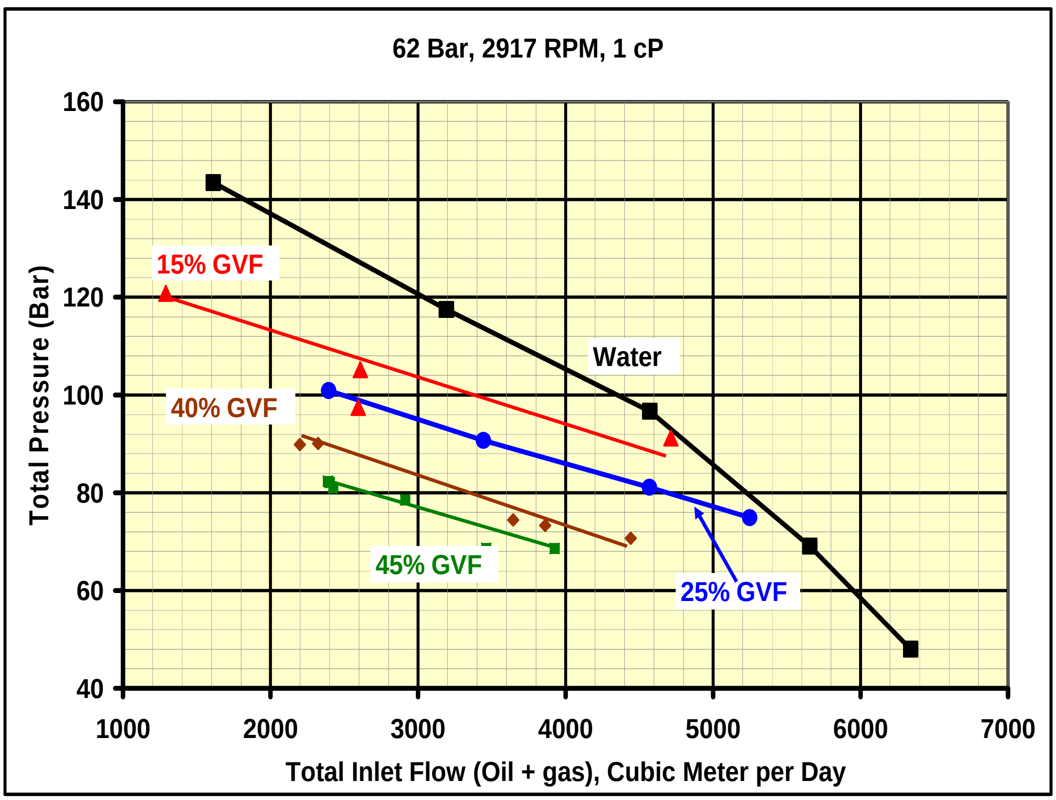

Fig. 7, WJE1200 32 stage pump performance at 62 Bar intake pressure, 1 cP viscosity and 2917 rpm.

Head flow performance at various GVF is shown in figures 7 and 8, for 62 Bar intake pressure at 1 cP and 265 cP viscosity of liquid and up to 45% GVF. Performance decreases as %GVF increases, showing effect of two phase losses.

Comparing 350 cP & 265 cP performances in figure 8 at constant GVF, there is very small (almost negligible) difference between the head flow characteristics. Again comparing 40% GVF comparison in figure 7 and 8 for 1 cP & 265 cP, shows the effect of viscosity and reduction of performance as viscosity increases. Based on the above comparisons, the effect of gas is more prominent than the effect of the viscosity.

There are two scenarios: (A) – Oil and gas are miscible. In this case, as pressure increases, part of the gas goes into oil and reduces the effective viscosity of the fluid and mixture. (B) Oil and gas creates emulsion. This situation will increase the effective viscosity of the mixture and performance will further deteriorate.

Knowing the oil and gas properties and laboratory testing of how they react with each other, is very important to better understand the performance of the two phase flow behavior.

<u>Effect of inlet pressure on two phase flow</u> – Next three figures, 9 to 11, show the effect of pressure on the two phase flow performance of the pump.

Effect of intake pressure is clearly visible as performance improves with the increase in the intake pressure. Solid lines representing higher intake pressure in figure 9 and 10 show improved head flow performance across the operating range. Additionally, deterioration of the performance at the lower flow rates decreases with the increase in the intake pressure. Higher intake pressure may also improve the stable operation at the lower flow rates.

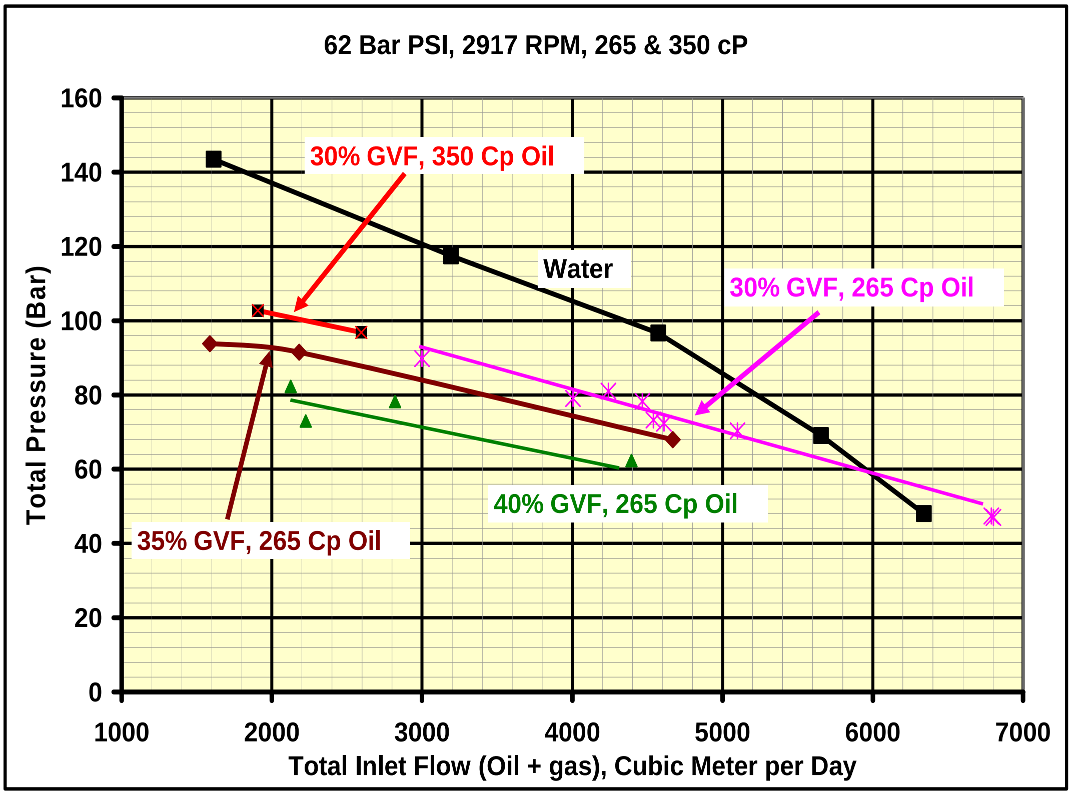

Fig. 8, WJE1200 32 stage pump performance at 62 Bar intake pressure, 265 & 350 cP viscosity and 2917 rpm.

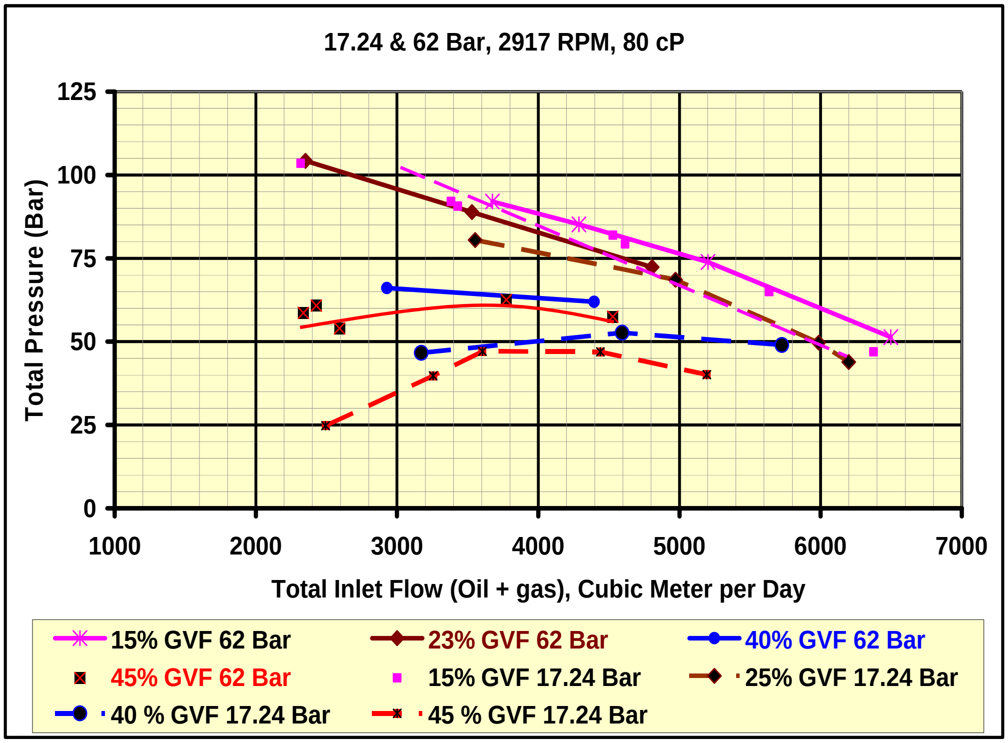

Fig. 9 Comparison of WJE1200 32 stage pump head flow performance at 17.24 & 62 Bar intake pressure,80 cP viscosity and 2917 rpm.

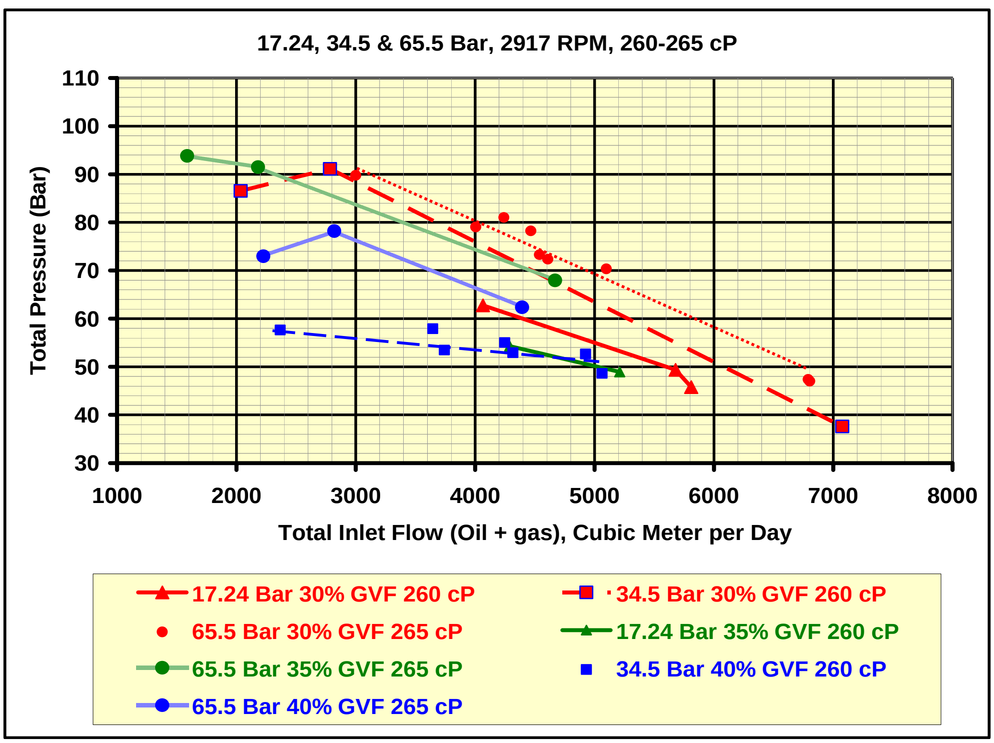

Fig. 10, Comparison of WJE1200 32 stage pump head flow performance at 17.24, 34.5 & 65.5 Bar intake pressure, 80 cP viscosity and 2917 rpm.

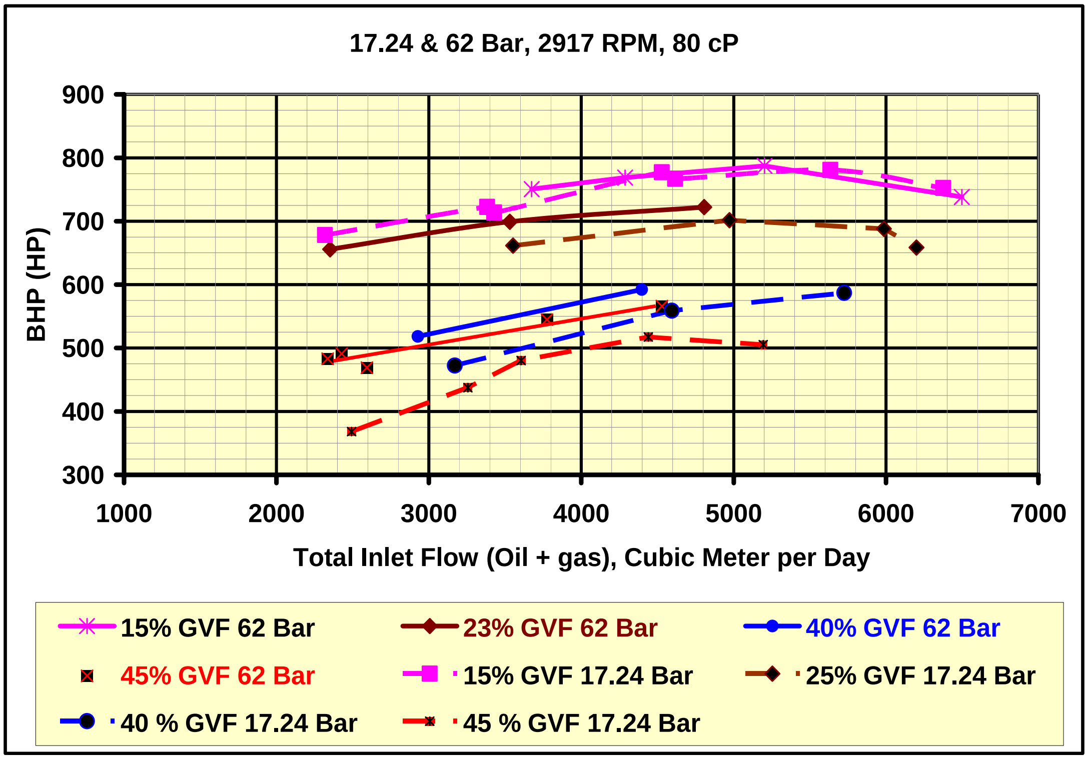

Fig. 11, Comparison of WJE1200 32 stage BHP flow performance at 17.24 & 62 Bar intake pressure, 80 cP viscosity and 2917 rpm.

BHP increases with the increase in the intake pressure due to increase in the density of gas. Pump performance becomes more stable from the BHP characteristics point of view as the intake pressure increases as shown in figure 11.

CONCLUSION

The Hydraulic Institute method for predicting viscous performance of a pump in a single phase fluid can be used as a preliminary estimate if no data is available.

BHI Centrilift test data does not match with the Hydraulic Institute method.

Pump manufacturer need to supply guidelines for predicting performance of multi stage pump in viscous and two phase flow conditions.

Stage by stage performance calculation of multi stage pump is recommended for viscous flow applications. Viscosity and gas volume should be calculated from first stage as a one stage pump, corrected by pressure and temperature rise, and then should be used for second stage and so on.

As intake pressure increases, gas behaves as more liquid than as a gas and chances of gas going back to solution increases, (a)lowering fluid viscosity if gas miscible in liquid or (b) if gas not miscible, then mixture viscosity will increase by creating emulsion.

An increase of the GVF will decrease the BHP requirement of the pump due to reduction in the mixture density. However, it may increase the BHP requirement if it creates emulsion.

Pumps should be run at the highest possible operating speed for viscous two phase flow conditions, considering thrust, NPSH, operating flow range, efficiency and erosion.

NOMENCLATURE

BEP = Best Efficiency Point
BHP = Brake Horse Power, HP
GVF = Gas Void Fraction (Gas volume / Total Fluid Volume)
cP = Centi Poise, Dynamic Viscosity
cSt = Centi Stokes, Kinematic Viscosity

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Annex A

Schematic Diagram of Test Loop