GAS-LIQUID FLOW THROUGH CENTRIFUGAL PUMPS— CORRELATION OF DATA

by

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ABSTRACT

The head capacity characteristics of three different submersible centrifugal pumps were correlated as a function of the gas-to-liquid ratio and pressure at the pump suction. The experimental data were from two independent laboratory investigations. One investigation used air and water as the working fluids, while the second study was conducted using diesel fuel and carbon dioxide with suction pressures up to 400 psig. The results indicated that at low pressures, gas volumes exceeding ten percent began to cause a serious degradation of pump performance.

Satisfactory performance was achieved at high gas ratios as the suction pressure was increased.

INTRODUCTION

It is known that the presence of free gas in a liquid affects the performance of a centrifugal pump. Test work as early as 1929 was concerned with the possibility of controlling pump output and, thus, conserving power by the admission of air to the pump suction ([1] as reviewed in [2]), although this idea has long since been abandoned [3]. Published studies of the effects of entrained air on the performance of a submersible centrifugal pump are presented [4], and effects of gas on staged submersible centrifugal pumps are presented [5].

Analysis of the charactristics of a pump under the loss of coolant accident for nuclear reactors is studied by Runstadler [6]. Patel and Runstadler present results of visual observations of air/water flow in a 1/20 scale model pump and show the similarity of a pump and rotating channel [7]. Sekoguchi presents experimental data using resistivity probe techniques and found as air increases the flow changes from bubble to shy flow [8]. Then a large air pocket forms near the inlet of each flow channel of the impeller and the pump fails to function effectively when the end of the slug reaches the impeller periphery. Slip measurements show gas slugs move slowly, compared to the liquid, effectively reducing the area for liquid flow.

A program to define the effects of free gas on the performance of electric submersible centrifugal pump stages is described. The effects of the free gas show up as a deterioration of the head-capacity curve, such as areas of unstable head production, and effects similar to cavitation at higher flowrates. Depending on the amount of free gas through the pump, these effects may vary from slight interference to gas locking.

Gas interference is indicated on the surface amperage chart by rapid variation of the motor loading. Gas locking occurs when the pump ingests too much gas and actually stops pumping because its head (or pressure) production is drastically decreased. This causes the motor to unload and to shut down because of low ampere surface control protection.

When designing an electric submersible pump for a gassy application, it is desirable to know the amount of free gas the pump can tolerate and to compare this to downhole gas conditions. Thus, the objectives of this project were (1) to generate experimental data relating pump performance (i.e., head-capacity) to the gas-liquid ratio at the pump suction and to the pump suction pressure, (2) to correlate these data, and (3) to develop a model which would predict head-capacity performance of a submersible pump as a function of gas-liquid ratio, suction pressure, pump type, and any other pertinent parameter.

Experimental data examined here were collected by two independent research efforts. Two test facilities and programs are described and results from each are presented and discussed [5]. Tests conducted by Amoco Production involved using water

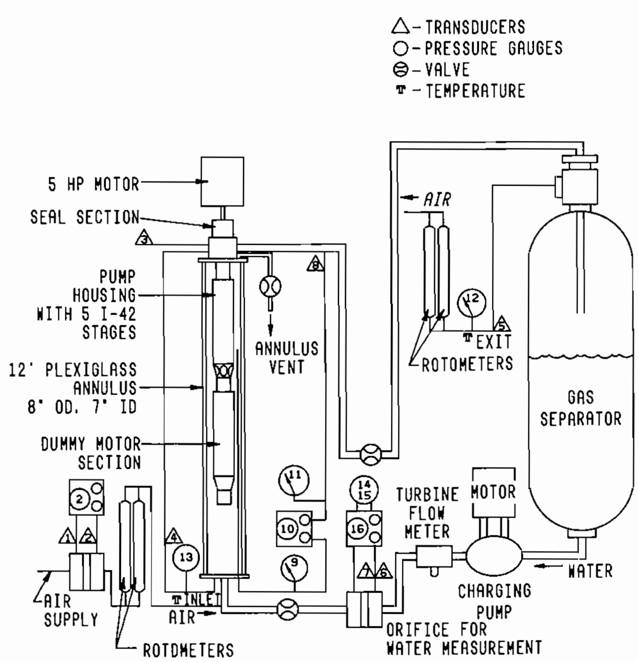

Figure 1. Schematic Diagram of Test Facility.

and air. A separate program was conducted for Centrilift Hughes, Incorporated at the R. C. Ingersoll Research Center, using diesel fuel and CO_2. In both programs, the purpose of the tests was to define the performance of typical submersible centrifugal pump stages, when free gas volumes were introduced at the pump suction under various flow and pressure conditions.

FACILITIES AND PROCEDURES FOR AIR/WATER TESTS

The test facility at Amoco was an above-ground installation in which the flow patterns and distribution of air volumes could be viewed. For simplicity, water and air were chosen as working fluids. A schematic diagram of the test loop is shown in Figure 1. The pump housing, containing five pump stages, was installed in an 8 in outer diameter (OD), 7 in inner diameter (ID) plexiglas tube. Because of the plastic container, the annulus backpressure was restricted to low pressures, with the majority of the tests being run at 25 psig to 30 psig.

The system was designed so that the water flowed through the system in a closed loop, but the air was directed to make a single pass through the pump or annulus before separation and venting. The injected air that did not enter the pump continued up the annulus and out through a backpressure regulator. The air entrained into the pump was separated from the fluid stream in a gas separator downstream from the pump and was vented through floating ball rotometers for measurement. See Lea and Bearden for additional details [5].

DATA REDUCTION AND RESULTS FOR AIR/WATER TESTS

The data collected were reduced to compare them with published performance curves, including the head, efficiency, and horsepower performance values. The motor horsepower delivered to the system was calculated. To obtain this value, the amperage through each of the three motor leads was measured at 440 V supply and used to calculate the motor horsepower published curves.

The average head, h, produced for one stage was calculated from the measured delta pressure across the pump and a calculated bulk density for the air/water mixture based on the following expression:

$$h = \frac{\Delta p(144)}{F_{wv\ aw}(n)} \tag{1}$$

where

Δp = delta pressure across the pump, psi

n = number of stages

$F_{wv\ aw}$ = average weight density of the air/water mixture through the pump, lbf/ft^3,

and

$$F_{wv\ aw} = \frac{F_{wv\ w} \times q_w + F_{wv\ a} \times q_a}{(q_w + q_a)} \tag{2}$$

where

$F_{wv\ w}$ = weight density of water, lbf/ft^3,

$F_{wv\ a}$ = weight density of air, lbf/ft^3, at the average pressure between stage inlet and outlet pressure, lbf/ft^3,

q_w = volumetric flow rate of water, gal/min, and

q_a = volumetric flow rate of air at the average pressure between stage inlet and outlet pressure, gal/min

The efficiency, E, of the pump for one stage was found from the ratio of the output hydraulic horsepower to the input horsepower as calculated from amperage and voltage data and the motor curves:

$$E = \frac{\Delta p(144)(q_w + q_a)}{(hp)(7.48)(60)(550)(n)} \quad (3)$$

where
hp is horsepower.

Results

The results (data points) for head only are shown versus published performance curves (smooth line) on Figure 2, repeated here from a previous study [5]. The data collected were focused on the recommended operating range. Each data set shows a different value of the percent by volume of gas at the pump intake. The results show the beginning of serious departure from the head curves at about seven percent free gas by total volume and intermittent gas locking at about eleven percent. The bands of calculated head, shown when surging begins, indicate the head oscillated from high to low values with a period of a second or two. Although there is still the periodic head produced for greater than eleven percent gas volume flows at the pump intake, the pump is not performing near published head values once the percent by volume of gas exceeds some point between seven percent and eleven percent by volume at the intake.

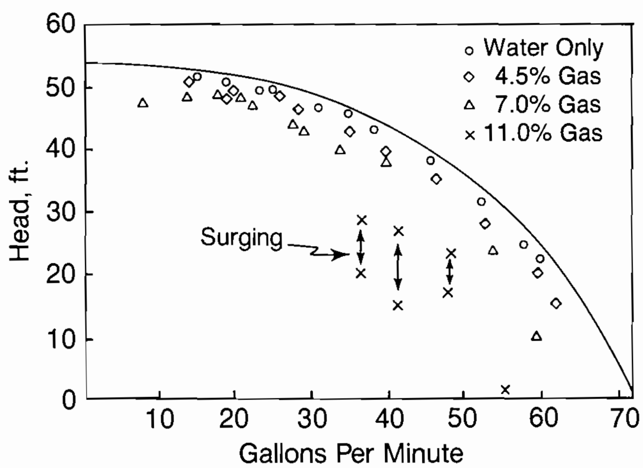

Figure 2. Deterioration of Water Head Curves with Gas [5].

FACILITIES AND PROCEDURES FOR CO_2/DIESEL TESTS

A simple closed-loop system was designed to test various submersible pumps. Simplicity was desirable because of the problems encountered with two-component flow.

A schematic diagram of the test facility is shown in Figure 3. The casing (1), which housed the submersible pump (2), is eight inch pipe, and the discharge flow line (3) is two inch pipe. The casing was designed so that various flow inlets (4 and 4a) could be used, and two windows (5) were mounted so that flow at the pump inlets could be observed. A sheet metal baffle (6) was placed between the casing and pump. The baffle has the same inside dimensions as a seven inch casing and forces the fluid to follow a path similar to an actual deep well setting. The pump was belt-driven by a 7½ hp vertical surface motor (7) mounted below the casing base flange. The nonelectrical instrumentation consisted of a positive displacement flowmeter (8), a differential pressure gauge (9) between the pump intake and discharge, a pressure gauge (10) for casing static pressure, and temperature thermocouples and gauges.

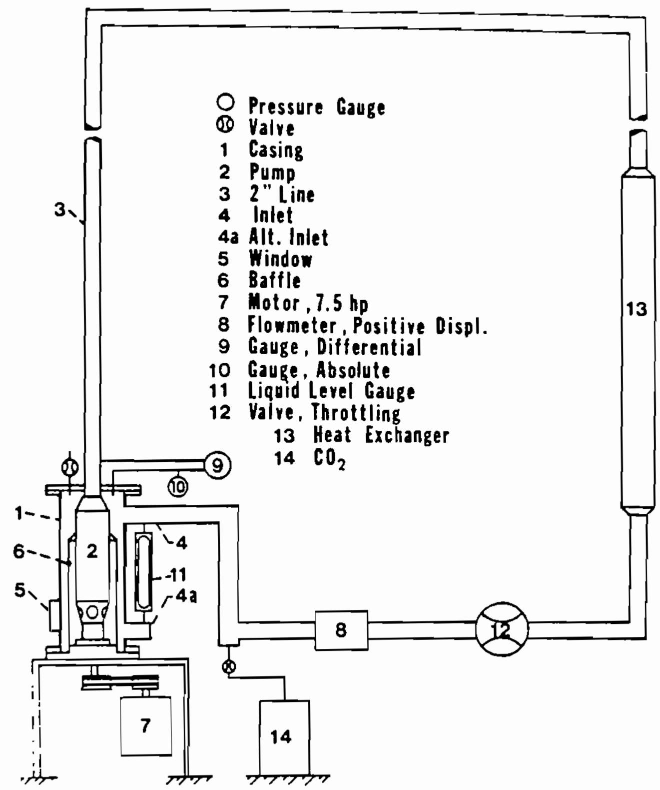

Figure 3. Research Pump Test Facility.

The test fluids were diesel fuel and CO_2. These fluids were chosen to test with a gas solubility effect present similar to actual oil well production fluids. Additional discussion may be found in the Lea and Bearden article [5].

Test Program

For each test, the loop was filled with diesel, and then a percentage of the total enclosed volume was displaced with CO_2. After the desired amount of disel was displaced for a particular test, the pump was started and the static system pressure was stabilized with an additional amount of CO_2. Once the system was stabilized at the desired test loop pressure and liquid volumetric percentage, the pump performance tests were started.

Numerous tests were conducted on several different types of pump stages: a radial pump stage designed for 42-gal/min optimal flow (I-42B), a radial pump stage with an optimal flow rate of 73 gal/min (C-72), and a mixed flow pump stage with an optimal flow rate of 80 gal/min (K-70) (Figure 4). The last two stages provided a comparison of two different design styles with the same approximate design flowrate.

DATA REDUCTION AND RESULTS FOR CO_2/DIESEL TESTS

The data collected for the various fluid mixture and pressure tests were plotted as pressure/flow rate curves and compared against the pump's pressure/flowrate performance with 100 percent diesel.

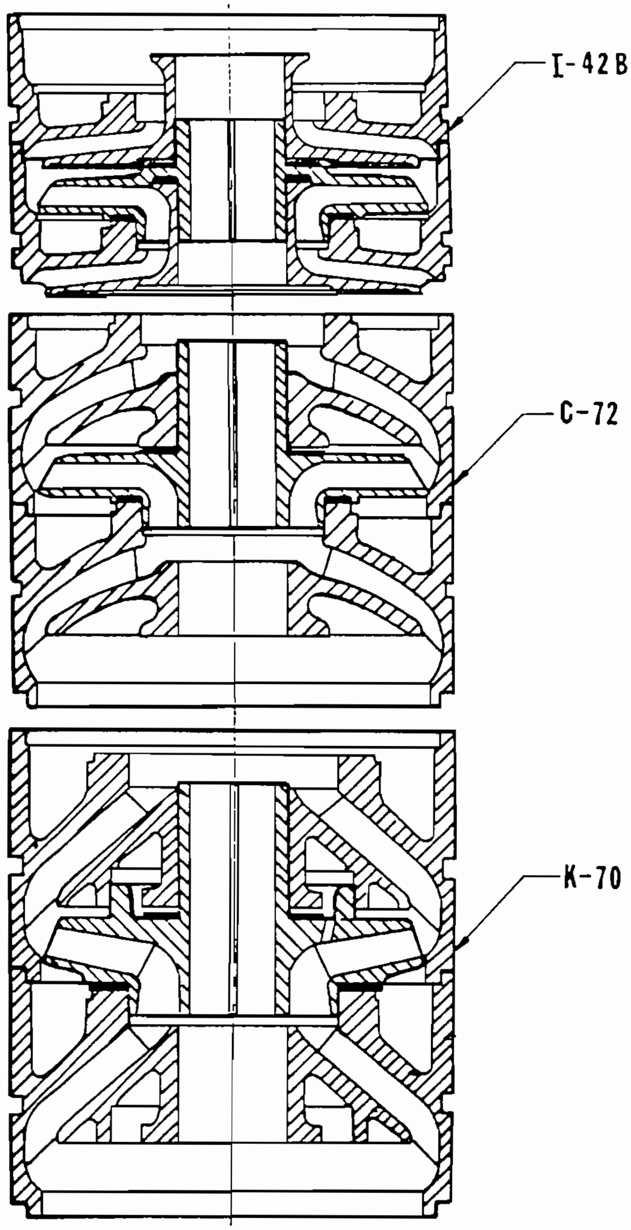

Figure 4. Pump Stages Tested.

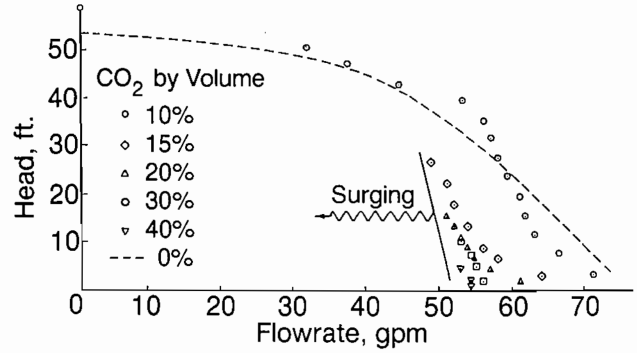

Figure 5. Diesel/CO_2, 50-psig Intake Pressure, I-42B Pump.

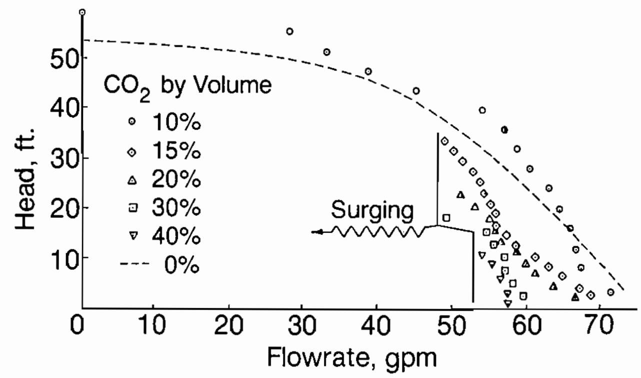

Figure 6. Diesel/CO_2, 100-psig Intake Pressure, I-42B Pump.

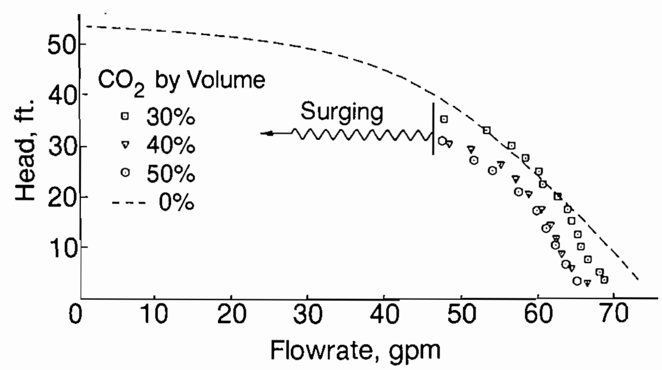

Figure 7. Diesel/CO_2, 400psig Intake Pressure, I-42B Pump.

The tests (Figures 5, 6, 7) run on the I-42B pump were reduced from pressure to feet of head so they could be compared with the air/water tests run on the same type stages. The pressure was reduced to head by using an equation similar to Equation 1. The average fluid gravity, Equation 2, was modified to account for the CO_2/diesel volume factor (two percent/100 psig).

Results

The results of the CO_2/diesel tests are shown in Figures 5, 6, 7, 8, 9, 10 and 11. The results of varying gas/liquid volumes at 50-,100-, and 400-psig pump intake pressures are shown in Figures 5, 6 and 7 for an eight-stage I-42B pump, the same type as used in the air/water tests.

Several conclusions can be reached from these three curves. First, pump performance is a function of both the percent of free gas by volume and the pump intake pressure. In general, the head measured to the right of the performance curves drops as the intake pressure decreases. The higher intake pressures require lower percent gas compression through the pump. Second, at values equal to or greater than 15 volume percent gas, the pump enters an area of severe head oscillation at flowrates less than the volume flow at the design point, which in this case was about 50 gal/min. Finally, at higher flowrates, the head takes a nearly vertical dip, which is similar to the effect of cavitation. It is noticed that the calculated head values for the lowest gas volume present (ten percent at 50 and 100 psig) are above the diesel-only base curves. This may be explained as caused by gas holdup in the system, where a small percentage of gas holdup is a large percentage of the total gas in the system. Therefore, fluid through the pump can be approaching a pure liquid and perhaps should not be corrected uniformly to account for gas presence as was done here. Although the gas presence decreases pressure output, the use of mixed density equations may simply not be adequate to describe head production over some ranges of performance.

The effects of gas on the performance of different types of stages (mixed and radial) are shown in Figures 8, 9, 10 and 11. The mixed-flow design, K-70, is represented in Figures 8 and 9, and the radial design, C-72, is shown in Figures 10 and 11. Both stages are designed for the same approximate design flowrate. Therefore, any difference in the ability to handle gas would be attributed to their design differences. From the test results, note that even though both stages have a surging region, the mixed-flow design has less deterioration in its pressure/flowrate curve. Therefore, as a pump stage tends toward a highly mixed or axial flow design, its gas-handling capability should increase. Even though the K-70 pump had less deterioration in head than the

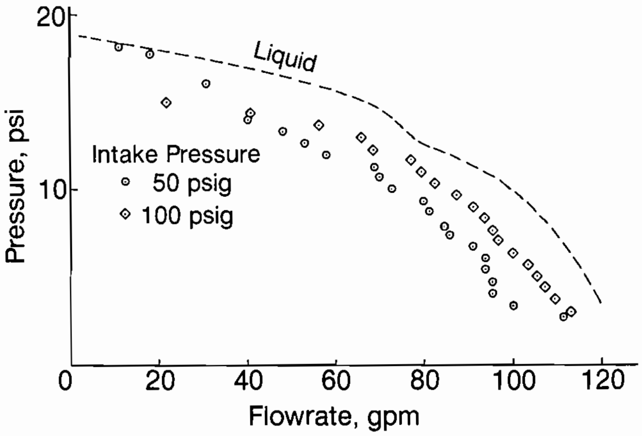

Figure 8. Diesel/CO_2, 10 vol% CO_2, K-70 Pump.

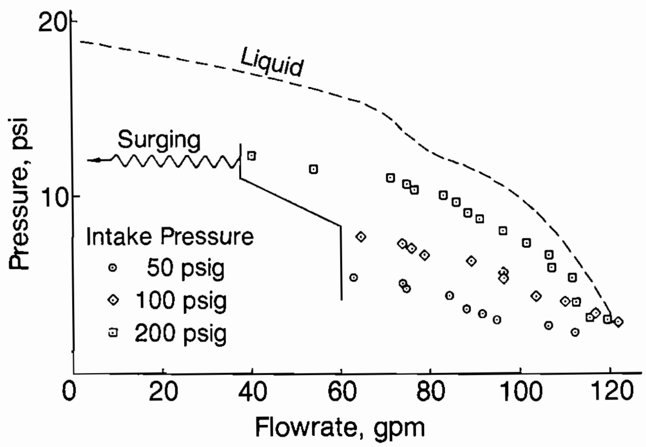

Figure 9. Diesel/CO_2, 15 vol% CO_2, K-70 Pump.

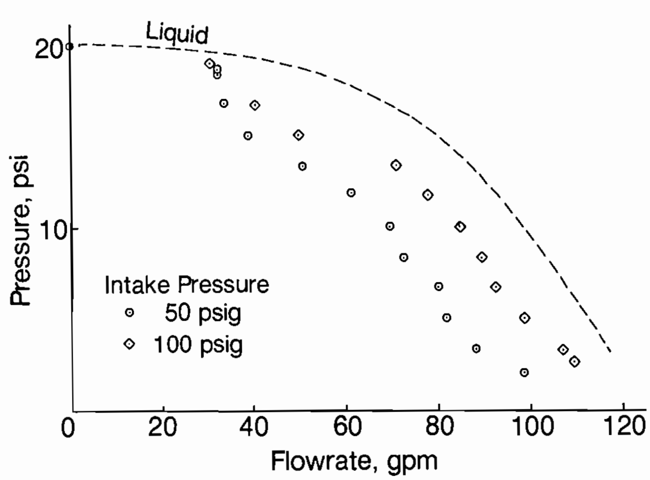

Figure 10. Diesel/CO_2, 10 vol% CO_2, C-72 Pump.

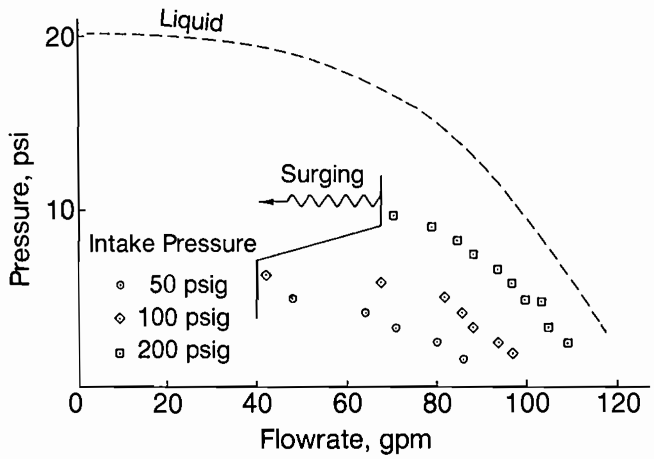

Figure 11. Diesel/CO_2, 15 vol% CO_2, C-72 Pump.

C-72 pump, both types entered a pressure oscillation or surging area at flowrates equal to or less than the optimal design flowrate.

CORRELATION OF THE DATA

Eighty-six experimental head-capacity curves were utilized to develop the correlations. There were twelve sets of data for the I-42 pump operating on air/water, 24 sets of data for the I-42 with CO_2/diesel, 24 sets of data for the C-72 with CO_2/diesel, and 26 sets of data for the K-70 pump with CO_2/diesel.

For a given pump, the controlling factors used to describe the deterioration of head were the gas/liquid ratio, the suction pressure, and the capacity. That is,

$$\frac{H}{H_{sp}} = \psi \left[\frac{q_s}{Q}, P_s, Q\right] \tag{4}$$

where

- H = head with gas-liquid flow
- H_{sp} = head with single-phase liquid flow
- ψ = indicates a function of following quantities
- q_s = volumetric flow rate of gas at pump suction,
- Q = volumetric flow rate of liquid at pump suction,
- P_s = pressure at pump suction

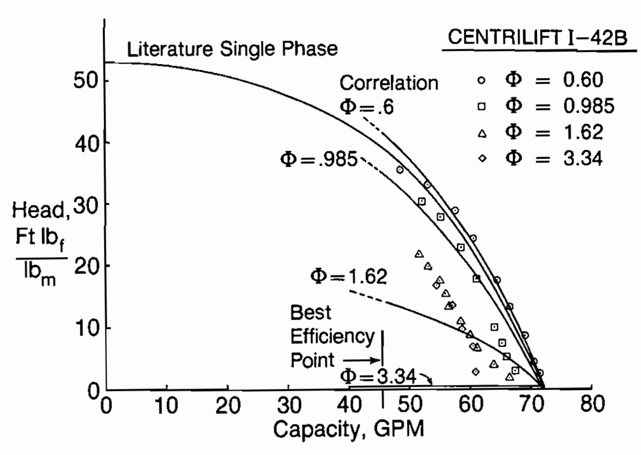

Figure 12. Pump Performance Deterioration for I-42.

Establishing the functional form of Equation 4 required considerable trial and error. Crossplots indicated a general exponential decay in performance with q_s/Q. The resulting correlations for each of the three pumps are included in the APPENDIX. The correlations predict the head-capacity curve fairly well for low gas volumes at low suction pressures and for higher gas volumes at higher suction pressures. The prediction falls off in the direction of higher gas and lower presssure conditions. However, the region of poor predictive capability of the correlations coincides with the region of unacceptable pump performance. The correlations only hold in general to the right of the best efficiency point and, as the data show, stable head production with gas only occurs in this region as the gas percentage increases.

A second relationship was developed to quantify the region of unacceptable pump performance. This was done by noting the pump performance at the various combinations of gas/liquid ratio and suction pressure employed. A quantity ϕ was defined as:

$$\phi = \frac{2000\ (q_s/Q)}{3\ P_s} \tag{5}$$

with P_s in units of psia. Plots of data with various ϕ values are shown in Figures 12, 13, and 14 against the correlations shown in the APPENDIX. Note that, in general, if $\phi \leq 1.0$, then the correlations are fairly accurate, and, of perhaps more importance, the head produced is still a substantial fraction of liquid only head. Values other than 1 could be chosen to be more or less conservative.

The following example illustrates how ϕ could be used to predict whether or not a gassy well could be pumped with a downhole centrifugal pump or not.

Example

Given the following data:
WOR (water to oil ratio) = 1.0
calculate the gas volume q_s (in bbl/d) at p_s:

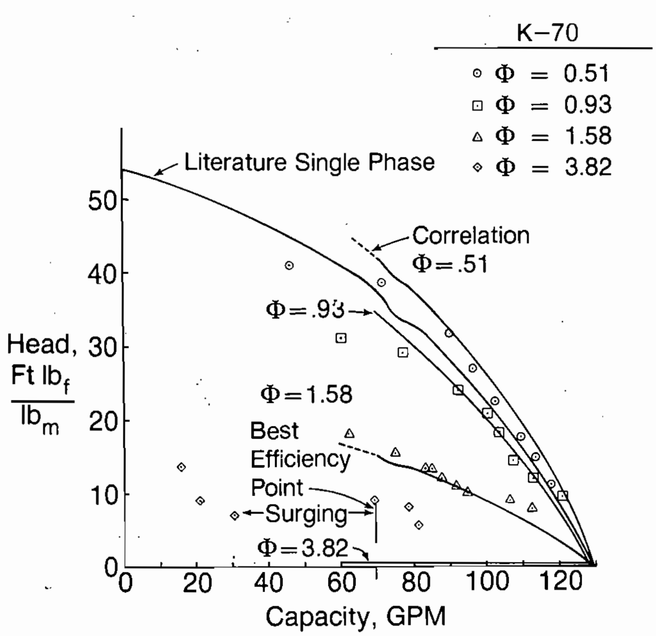

Figure 13. Pump Performance Deterioration for K-70.

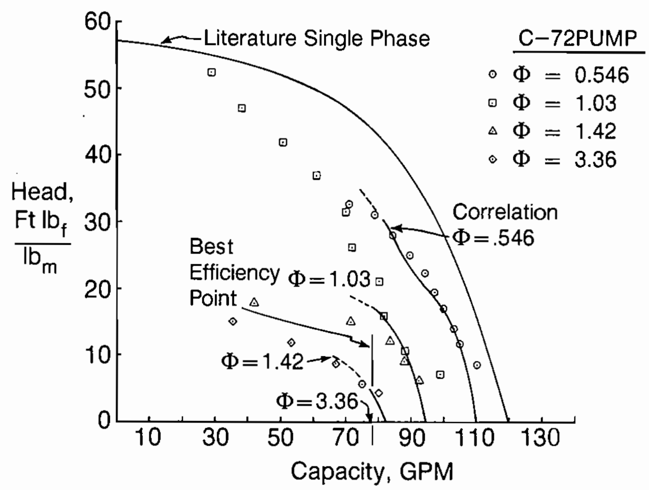

Figure 14. Pump Performance Deterioration for C-72.

$$q_s = Q_o(GOR - R_s)\left(\frac{14.7}{P_s}\right)\left(\frac{T+460}{520}\right) z \left(\frac{1}{5.61}\right) \tag{6}$$

where

Q_o = oil flow rate, bbl/d
R_s = solution GOR, SCF/bbl
T = pump intake temperature, °F
C = 5.61 ft^3/bbl

Calculate the liquid volume at pump suction in bbl/d:

$$Q = Q_o\ (WOR + B_o) \tag{7}$$

where

Q = total liquid volume rate at p_s, bbl/d
B_o = formation volume factor (oil swellage factor), bbl/bbl

Using Standing's correlations for R_s and B_o [9], the following procedure can be used to calculate p_s (minimum) from $\phi = (2000(q_s/Q))/3p_s$ where $\phi = 1.0$.

Procedure

1. Guess p_s.
2. Calculate q_s, Q, P_s given the ϕ relationship and previous expressions.
3. Compare p_s to guessed P_s
4. Return to Step 1 until old ≈ new P_s calculated.

Once p_s is found as a function of GOR, it can also be calculated as a function of the gas percentage at the suction:

$$\%\ \text{gas} = \frac{q_s}{q_s + Q} \times 100 \tag{8}$$

Intake pressure as a function of gas percentage at the intake and also versus various possible well GORs is shown in Figure 15.

In Figure 16, a family of hypothetical inflow performance relationships (IPR) curves for a well are plotted. A straight line relationship is calculated above a calculated bubble point. The IPRs are constructed with the shut-in well pressure of 3000 psi and a test point of 300 bbl/d at 1500 psi bottom hole pressure. Vogel's IPR curve is used to complete the curves below the

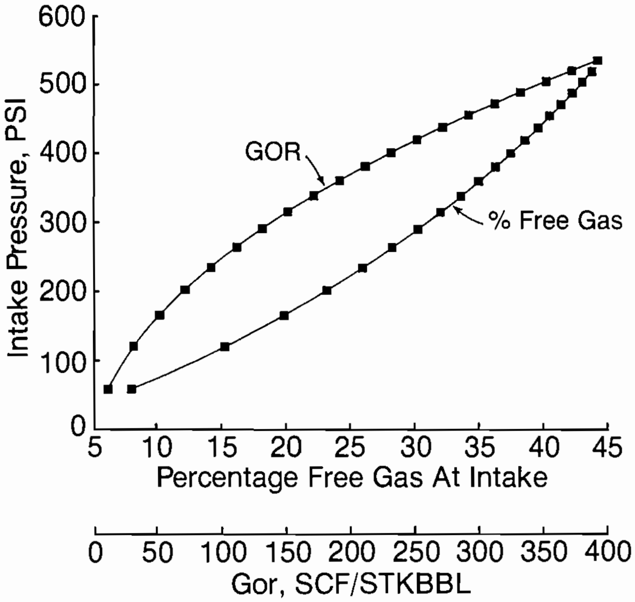

Figure 15. Acceptable Free Gas Correlation for ESPS.

bubble point [10]. Note that the five IPRs are shown with GORs ranging from 0 SCF/bbl to 400 SCF/bbl. Disregarding net positive suction head (NPSH) requirements, the 0 GOR well could be pumped off. However, as the GOR reaches 400, the well cannot be pumped off below P_s = 520 psi without the correlation indicating the pump would not perform satisfactorily with free gas present.

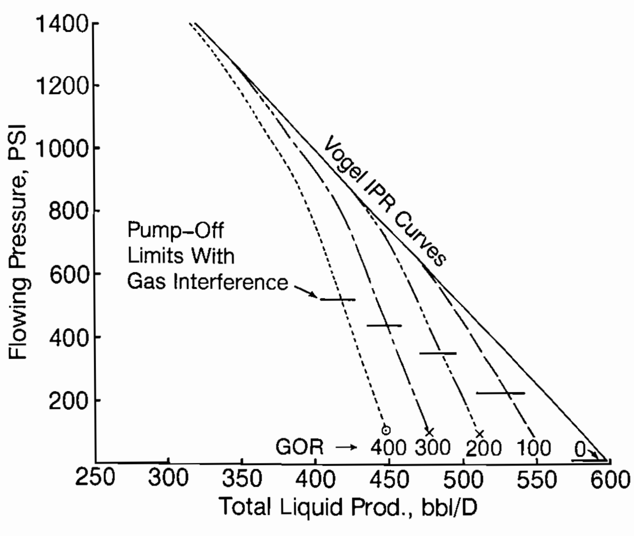

Figure 16. Example of Pump-Off Limits for Reservoirs with Various GORs.

This correlation allows a potential pump user to predict when gas becomes a problem. Then a gas separator can be used or less drawdown of the well can be expected. Some additional stages should be added to acount for head deterioration with gas. The additional stages required could be estimated from the correlations in the APPENDIX or from examination of test data presented. The pump should be designed to operate to the right of the head curve to avoid surging when gas is expected through the pump.

CONCLUSIONS

An empirical model was developed to predict the head-capacity curve for electric submersible centrifugal pumps as a function of the gas/liquid ratio and suction pressure. Approximate correlations for different pump designs are given by equations (A-1) to (A-5) of the APPENDIX. The correlations show the exponential decay of head with increasing (q_s/Q).

The regions of reasonably good pump performanced coincide with the regions of applicability of the correlations. These regions are identified by a parameter, ϕ

$$\phi = \frac{2000\ (q_s/Q)}{3\ P_s} \tag{9}$$

For $\phi < 1$, correlations are applicable and good head production with gas present is possible. For $\phi > 1$, correlations are not applicable, but pump performance is in general so poor that operation in these regions is to be avoided.

Representative values from Equation (9) indicate that each of the pump stages tested will tolerate about 13 percent gas (by volume) at 100 psia suction pressure and up to 37 percent gas at 400 psia suction pressure, with no serious deterioration of head performance. Test results show stable head production only at flowrates higher than the best efficienty point as the gas volume at suction increases until more gas finally completely gas locks the pump.

APPENDIX

Approximate correlation of the data for I-42 and the K-70 pumps is achieved by:

$$\frac{H}{H_{sp}} = e^{-a(q_s/Q)} \tag{A-1}$$

with "a" given by:

$$a = \frac{346430}{P_s^2}\left(\frac{q_s}{Q}\right) - \frac{410}{P_s} \tag{A-2}$$

Correlation of the data for the C-72 pump is by:

$$\frac{H}{H_{sp}} = e^{-a(q_s/Q)}\,[1 - 0.0258\ (Q - Q_D)$$

$$+ 0.00275\ (Q - Q_D)^2 - 0.0001\ (Q - Q_D)^3 \tag{A-3}$$

with

$$Q_D = 98.3 - 33.3\ \phi \tag{A-4}$$

and

$$a = \frac{285340}{P_a^2}\left(\frac{q_s}{Q}\right) \tag{A-5}$$

The correlations are designed for flowrates higher than the best efficiency point. Accuracy is rapidly lost as ϕ begins to exceed 1, but the exponential drop in head with increasing q_s/Q is a general characteristic.

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