DESIGN OF SUBMERSIBLE ELECTRIC PUMPING SYSTEMS

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ABSTRACT

A general method for designing submersible electric pumping systems is presented. The technique is based on a mathematical model of the pumping system which includes mechanical, fluid dynamical and electrical simulations. Basic parameters which affect the design of a submersible electric system are discussed. To illustrate the technique, applications are made to high water cut (non-gassy) wells, high GOR (gassy) wells, variable frequency drives and tapered pumps.

INTRODUCTION

The most common and the least complicated design procedure involves neglect of free gas. These design calculations can be easily made with a hand held calculator. Non-compressible designs should be applied to wells with minimum amounts of produced gas, usually with gas liquid ratios (GLR's) less than 50 standard cubic feet per barrel passing through the pump. Normally wells with high water cuts fall into this category. The design approach is to select the pump size based on anticipated producing rate with no consideration of crude shrinkage or free gas. To determine the number of pump stages, the Total Dynamic Head (TDH) concept is used. This concept assumes that the pressure developed by each pump stage is constant; thus, fluid density and volume are assumed to remain unchanged throughout the pump from intake to discharge. All pump stages are lumped together for calculating total pump discharge head and power required to drive the pump.

The TDH is made up of net lift, friction head and the head required to overcome wellhead tubing backpressure. This can be expressed by the following formula:

$$TDH = NL + F + BP$$

where:

NL is net lift, i.e. pump setting depth less the head supplied by pump submergence.

F is head loss from tubular friction.

BP is head required to overcome wellhead backpressure.

Adjustment for fluid specific gravity is made when computing power required by the pump. Since pump performance curves are based on a fluid specific gravity of 1.0 (fresh water), power required is the product of horsepower read from pump performance curves and actual fluid specific gravity.

The Total Dynamic Head design method does not consider the effects of free gas and can result in poor pumping system performance if applied to gaseous wells. Figure 1 shows how production and system efficiency are reduced from the optimum as the gas liquid ratio is increased. This applies to a pump setting depth of 5000 feet, a producing rate of 950 BPD and a pump intake pressure of 100 psi. However, results are similar regardless of pump depth and producing rate.

Start up and continuous operation are often difficult in gassy wells because instantaneous GLR's can be much higher than long term average GLR's. Gas is vented up the casing in a heading and cyclic manner which is apparent from a gas meter chart on an individual well producing gas up the casing. As gas migrates through the liquids up the casing by gravity segregation, the fluid gradient is reduced. As the amount of free gas increases, the gas velocity through the liquid increases and more liquid is carried (entrained) by the gas up the casing. This condition is often referred to as "heading". At some point the energy from the escaping gas is not adequate to carry the entrained liquids and "fluid fall back" occurs. When this occurs, no gas is vented and the pump is forced to handle more free gas. The cyclic venting conditions of "heading" and "fallback" occur regardless of how well free gas is separated at the pump. Therefore, produced gas often enters the pump intake in slugs which varies the density of the fluids entering the pump. As a result pump power requirement fluctuates with the amount of free gas handled which causes motor current to vary accordingly. If the total volume of free gas plus liquid should exceed the pump's head generating ability, then the pump will cavitate. Under this condition, pump power requirement approaches zero and motor current falls to near no load current. With properly set underload protection the system shuts down.

Lea and Bearden[1] have also demonstrated in the laboratory that pump head, capacity and efficiency are reduced as the volume of free gas is increased. This is especially apparent at pressures lower than 400 psi. Thus, the performance of the lower pump stages is adversely affected in gassy wells. In addition pump stage design affects performance. The mixed-flow impeller design handles gaseous fluids better than the radial flow design.

More laboratory testing of this type is needed to expand results. Performance data are needed on each type of pump stage at various gas volumes and pressures. With more accurate performance data under gaseous conditions, pump designs can be improved.

GENERAL METHOD OF DESIGN

A general method of design should treat the effects of free gas as well as reduce to incompressible fluid design as a special case. Following is a brief discussion of several important factors which affect system design. A logic diagram for the procedure is shown in Figure 2.

Well Characteristics

For any method to be successful an accurate knowledge of the well is mandatory. Basic data about the well should include physical information such as:

a) Casing size (minimum ID), producing interval, PBTD and anticipated flowline pressure.

b) P-V-T properties of crude, API gravity, viscosity, water specific gravity, producing GLR, and bottom hole temperature.

c) Power supply voltage and line capactiy.

d) Adverse operating conditions including abrasives, corrosion, scale, emulsions and paraffin.

An accurate inflow performance relationship (IPR) curve for the well is of prime importance. Misapplications and poor designs are caused primarily by not accurately defining the well's capability. Once the well's current behavior is understood, consideration should also be given to future performance. The following questions as to future performance should be considered.

a) What type of reservoir drive mechanism exists (solution gas, water drive, gravity or waterflooding)?

b) Will the well's productivity tend to increase or decrease with time?

c) Will the water cut and producing GOR change?

Answers to these questions affect equipment selection.

Selection of Producing Rate and Pressure

The desired producing rate can be selected from the well's IPR curve. However, physical limitations imposed by casing size, lifting depths and available pump sizes may dictate a compromise in producing rate.

Calculation of Volume of Fluids at Pump Intake

The rate selected from the IPR curve is based on barrels of liquids at surface conditions. To determine the amount of fluids entering the pump in BPD, the amount of free gas and crude shrinkage should be included. These calculations require a knowledge of crude P-V-T properties and measured GOR's. A hand method of calculating approximate pump intake volume can be made using the work table and non-dimensional P-V-T curves shown in Figures 3, 4 and 5. Also, a judgement must be made on the amount of free gas that is vented up the casing. The amount vented varies on a well to well basis and is dependent on fluid properties, pump intake pressure, casing size, producing rate and the effectiveness of gas separators (reverse flow, rotary and shrouds). It is generally assumed that 70-80% of the total gas produced passes through the pump. This is considered a fairly safe and reasonable assumption for design purposes.

Pump Selection

Once the total intake volume of liquids plus free gas is known a pump size can be selected. Normally the largest series (diameter) that will fit inside the casing is selected because efficiencies tend to be higher and equipment costs are lower.

In wells producing gassy (compressible) fluids, the density and volume change on a stage by stage basis as the pressure increases through the pump. The pressure generated by each pump stage and the power required to drive each stage is directly proportional to the fluid specific gravity. The exception to this rule is the lower intake stages of the pump where the pressure could be less than 400 psi. As discussed earlier the head, capacity and efficiency decrease as percent gas by volume increases and as pressure decreases. However, this effect can be simulated by derating the pump as a function of percent gas and pressure. To properly design for pumping gaseous fluids, the mathematical model should simulate the pump on a stage by stage basis starting at the bottom (intake) and progressing upward. Pressure generated by each stage and power required by each stage is accumulated until adequate pressure is developed to lift the well. The number of pump stages required is dependent on the anticipated pump discharge pressure which is determined by the outflow or tubing performance. has been developed in recent years concerning flowing gradients because this technology is used in optimizing tubing performance in flowing wells and in gas lift design. Contributors to this technology include Poettmann and Carpenter, Ros, Brown and others. In addition, programmable equations as described by J. Orkiszewski[2] can be used for modeling tubing performance. Flowing gradient considerations include not only the effects of free and dissolved gas in the oil, but also the effects of wellhead back pressure, tubular friction and densities of produced oil and water. Using this approach, the pressure at the foot of the tubing can be calculated which is also the discharge pressure of the pump. The number of pump stages required to pump the well is directly dependent on this discharge pressure.

Motor and Power Cable Selection

The motor selected is dependent on pump power requirement, casing size and power cable size and length. Normally the largest series (diameter) that will fit inside the casing is selected because of lower initial costs and slightly better efficiency. The horsepower rating must be at least equal to the pump's power requirement under the design conditions. In addition, allowances should be made for possible changes in the producing water cut over the life of the pump. An increase in water cut is often the case in strong water drive reservoirs or in waterfloods. Start up conditions should also be considered, especially if heavy kill fluids are used or if high producing water cuts exist at startup.

Several voltage ratings are usually available for each motor horsepower rating. The voltage selected is dependent on the power cable length and size. Power cable losses should be less than 10% of the total energy consumed by the pumping system. In deep wells, higher voltage motors are often selected to reduce power cable losses. Since the cable size is limited by clearance between the casing and tubing, the losses from tubing fluid friction should be weighed against power cable losses. The tubing and cable sizes selected should minimize combined energy losses and initial capital costs. The anticipated power cable life also influences the size selected.

Changing the frequency with a variable frequency controller affects motor performance. Motor speed, power output and voltage rating vary directly with frequency.

Once the motor, power cable and frequency have been selected, the performance of the electrical system can be evaluated. Using motor performance curves such as shown in Figure 6 and knowing the percent motor loading, the following information can be generated.

a) Motor current

b) Power factor

c) Power cable losses in volts, in KWH or in $/month if the cost per KWH is known.

d) Required surface supply voltage.

e) Power consumed in KWH/month. If a cost per KWH is known the power cost in $/month can be computed.

f) Overall system efficiency.

Higher overall pump efficiencies in very gassy wells can be attained by tapering the pump. This is accomplished by changing to lower volume pump stages when the lower portion of the pump becomes less efficient because of gas compression. Generally tapers are limited to two and sometimes three pump sizes.

Another method for improving pump performance under gassy conditions is to increase pump capacity by using a variable frequency controller. The following affinity laws apply when designing for variable frequency.

$$\frac{q_1}{q_2} = \frac{f_1}{f_2} , \quad \frac{h_1}{h_2} = \left(\frac{f_1}{f_2}\right)^2 \quad \text{and} \quad \frac{p_1}{p_2} = \left(\frac{f_1}{f_2}\right)^3$$

where:

f is frequency, Hertz
q is rate, BPD
h is head, feet
p is power, HP

For example, the pump capacity is increased 1½ times when the frequency is increased from 60 Hz to 90 Hz. In addition the head is increased 2.25 times and power requirement is increased 3.38 times. The increase in head capacity and power requirement is advantageous in very gassy conditions that tend to result in low motor power requirement and frequent under current shut downs. As the frequency is increased, there is less tendency for shut downs when gas interference is occurring.

Variable frequency controllers can be applied where the pump is proposely over-designed in anticipation of waterflood response or where production potential was over-estimated. To avoid overpumping and shut downs under low volume pumping, the frequency can be reduced to match pump capacity with the well's fluid inflow.

With frequencies above 60 Hertz, designs can be made to increase production when casing size limits pump size. For example, the maximum pumping rate inside 5½ inch casing with 60 Hz power is about 3000 BPD at pump conditions. However, with 90 Hz power pumping systems can be designed to increase pump capacity to about 4500 BPD at pump conditions.

Designing pumps for higher fluid viscosities (say in excess of 40 SSU), is handled by modifying the pump's performance curves. Viscosity can be accurately measured for oils, but is highly variable as water cuts increase and emulsions form. High viscosity and emulsion problems are normally not troublesome when water cuts exceed 80%.

A brief discussion of the computer method used in the general design method is given in Appendix B.

Selection of Tubing Size

Factors affecting selection of tubing size are initial cost, friction loss and clearance for the power cable. Generally speaking, the energy loss from fluid friction should be less than 10% of the total energy consumed by the pumping system. All submersible pump manufacturers furnish friction loss charts which are fairly accurate in evaluating friction losses. The effects of viscosity are often included. Most flowing friction information is based on past work by Hazen-Williams. Solution gas and especially free gas greatly reduce the tubing fluid gradient. Considerable information

Selection of Controller and Transformer

The surface controller and transformers are the last items to be selected in the pumping system. Two basic types of controllers are available, i.e., the standard switchboard and the variable frequency controller.

The standard switchboard serves to control and to protect downhole equipment. Protection is provided by either electro-mechanical or solid state systems. The size selected depends on the anticipated surface supply voltage and current.

The variable frequency controller provides all of the functions of a standard switchboard plus the capability of varying the frequency and voltage. These controllers are rated in kilovolt amperes (KVA) and should be sized using the maximum frequency that is anticipated. The following equation applies.

$$KVA = \frac{(Vm \times Hz/60 + Vc) \times I \times \sqrt{3}}{1000}$$

where:

Vm is voltage rating of the motor at 60 Hz
Vc is the voltage loss in the power cable
Hz is the maximum anticipated frequency
I is the current rating of the motor

Transformars are also sized in KVA and the above equation is applicable. If single phase transformers are used divide the total KVA by $\sqrt{3}$ to determine the minimum size of each transformer.

APPLICATIONS OF THE METHOD

To illustrate the method, several practical applications are made. An example well is chosen which has the following producing characteristics.

Productivity Index - 1.0 BPD/psi
Production Caability - 1000 BOPD and 200 BWPD with 200 psi pump intake pressure.
Pump Depth - 5000 feet
Gas Oil Ratio through Pump - 458 SCF/B
Bubble Point - 1801 psi
Formation Volume Factor - 1.312
Oil Gravity - 35^{0} API
Water Specific Gravity - 1.05
Casing Size - 5½ inches

Eight designs are described below which demonstrate the flexibility of the general method. Results are summarized in Figure 7.

Design 1 assumes that the well produces no free gas. This is comparable to the Total Dynamic Head method of design.

Design 2 uses the same unit as in Design 1 but considers that the well produces with a GOR of 458 SCF/B passing through the pump. If not considered in the design, the effect of free gas reduces pump performance and results in a loss in production.

Design 3 is the same as Design 2 except that the number of pump stages is increased from 227 to 327 stages. Adding extra stages (overstaging) does not improve performance.

Design 4 uses a variable frequency drive to increase frequency from 60 to 90 Hz. Production is increased but system efficiency is low.

Design 5 uses a higher volume pump to ingest free gas more efficiently. The same GOR is maintained and a frequency of 60 Hz is used. By optimizing the pump size to handle free gas, production and system efficiency are increased.

Design 6 uses a tapered pump design for handling free gas. The same GOR is maintained and 60 Hz power is used. Tapering the pump shows only a slight improvement over Design 5.

Design 7 uses a variable frequency controller with a low frequency of 50.8 Hz to follow a decline in well production from 1200 BFPD to 600 BFPD at 200 psi pump intake pressure. This design assumes that no free gas is produced so that comparisons can be made with Design 1. Pump efficiency and system efficiency are reduced but remain high.

Design 8 uses a variable frequency controller with a high frequency of 76 Hz to follow an increase in well production from 1200 BFPD to 1800 BFPD at 200 psi pump intake pressure. Again, this design assumes that no free gas is produced so that comparisons can be made with Designs 1 and 7. Pump efficiency and system efficiency are comparable with Design 7.

Appendix A shows an example design format and output of the predictive computer program.

CONCLUSIONS

This paper describes a general method of design. The entire pumping system can be mathematically simulated so that the effects of free gas can be considered. Variable frequency controllers and tapered pumps are shown to be useful in adapting system design to well production capability.

The method presented is intended to identify applications for submersible pumping and to improve overall pumping system performance.

REFERENCES

1. J. F. Lea and J. L. Bearden: "Effect of Gaseous Fluids on Submersible Pump Performance", J. Pet. Tech. December 1982, 2922-2930.

2. J. Orkiszewski: "Predicting Two-Phase Pressure Drops in Vertical Pipe" J. Pet. Tech., June 1967, 829-838.

SUBMERSIBLE PUMPING PREDICTIVE ANALYSIS

COMPANY, LEASE AND WELL NUMBER: EXAMPLE, DESIGN NO. 6

ANALYSIS NUMBER: OS3-1-14-1 DATE OF ANALYSIS: 1-14-83

***** DOWNHOLE MOTOR *****

MANUFACTURER: CENTRILIFT

RATED CURRENT (AMPS): 43 RATED VOLTAGE (VOLTS): 1130
RATED POWER (HP): 75 FREQUENCY (HERTZ): 60

PREDICTED CURRENT (AMPS): 27 SURFACE VOLTAGE (VOLTS): 1195

COMPUTED PERFORMANCE BASED ON MANUFACTURER DATA

VOLTAGE LOSS IN CABLE (VOLTS): 65
MOTOR OUTPUT (HP): 18.5

VOLTAGE AT MOTOR (VOLTS): 1130
POWER FACTOR: 0.41

***** DOWNHOLE PUMP *****

MANUFACTURER: CENTRILIFT

PUMP INTAKE PRESSURE (PSI): 224
PUMP DISCHARGE PRESSURE (PSI): 757
GOR IN PUMP (SCF/B): 457
GLR IN PUMP (SCF/B): 381
OIL SHRINKAGE FACTOR AT PUMP INTAKE PRESSURE: 1.141

	VOLUMES ENTERING PUMP AT INTAKE PRESSURE
LIQUIDS ONLY (BPD):	1296
LIQUIDS PLUS FREE GAS (BPD):	4417

TYPE	STAGES	AVG EFF %	INLET PRESSURE PSI	OUTLET PRESSURE PSI
Z-69	100	68.9	381	757
N-80	127	36.5	224	382

***** ENERGY SUMMARY *****

POWER COST BASIS ($/KWH)	MONTHLY POWER COST ($) TOTAL SYSTEM	POWER CABLE
.01	168	9
.02	336	18
.04	673	37
.06	1009	55
.08	1346	73
.10	1682	92
.12	2019	110

SURFACE POWER INPUT (HP): 31
SUBSURFACE POWER INPUT TO MOTOR (HP): 30
HYDRAULIC POWER OUTPUT OF PUMP (HP): 10
OVERALL SYSTEM EFFICIENCY (%): 32.3

NOTE: POWER COST FIGURES DO NOT INCLUDE TRANSFORMER AND SURFACE TRANSMISSION LINE LOSSES.

***** OTHER DOCUMENTARY DATA *****

PUMP DEPTH (FT): 5000
TUBING PRESSURE (PSI): 50
FRICTION LOSS IN TUBING (PSI): 45

WELLHEAD TEMP (DEG F): 100
CASING PRESSURE (PSI): 50
AVG TUBING GRAD (PSI/FT): .141

DESIRED PRODUCTION AND FLUID PROPERTIES

BFPD: 1160
BWPD: 193

BOPD: 967
GOR: 588

WATER CUT

16.6

OIL GRAVITY (API): 35
BUBBLE POINT (PSI): 1801
FORMATION VOLUME FACTOR (BBL/BBL): 1.312

WATER GRAVITY (SG): 1.05
SOLUTION GOR (SCF/B): 588

***** PUMP PROFILE *****

STAGE	GRADIENT PSI/FT	RATE BPD	PRESSURE PSI
1	.263	1840	752
8	.255	1899	714
15	.247	1960	677
22	.239	2025	643
29	.232	2093	611
36	.224	2165	580
43	.216	2240	552
50	.209	2317	525
57	.202	2398	499
64	.195	2480	476
71	.189	2564	454
78	.183	2649	434
85	.177	2734	416
92	.172	2818	399
99	.167	2901	384
106	.161	3005	366
113	.155	3119	348
120	.15	3233	332
127	.145	3347	318
134	.14	3460	304
141	.136	3572	292
148	.132	3681	281
155	.128	3783	272
162	.125	3879	263
169	.122	3969	256
176	.12	4051	249
183	.117	4126	244
190	.116	4192	239
197	.114	4249	235
204	.113	4299	232
211	.112	4342	229
218	.111	4378	227
227	.11	4417	224

APPENDIX B

When a design is being made for a gaseous fluid, the computations are too lengthy to be conveniently accomplished by hand. A digital computer solution is desirable. The principle problem is to determine the pressure difference in the fluid created by the downhole pump. This involves the head characteristics of the individual stages and the fluid rates within the pump. Also involved is a computation of the power required by the pump which in turn determines the size of electrical motor required.

The total pressure increase across the pump is given by

(B-1) $$P = \sum_{j=1}^{m} \sum_{i=1}^{n(j)} w_{ij} h_{ij}$$

where h_{ij} is the head contributed by the i^{th} pump stage in the j^{th} taper. Similarly w_{ij} is the specific weight of the fluid passing through the ij^{th} stage. The quantities m and n(j) are respectively the number of tapers and the number of stages in the j^{th} taper.

The specific weight of the fluid varies as the fluid passes through the pump. A reference amount of material (oil, water and gas) is defined as follows. For each stock tank volume of oil V_o, there is an associated volume of water given by

(B-2) $$V_w = V_o \left(\frac{1}{c_o} - 1\right)$$

in which c_o is the spatial volume fraction of oil. In a similar fashion there is an associated volume of gas which is highly dependent upon pressure and temperature. If the reference stock tank volume of oil is taken to be one barrel, there results the following total volume of oil, water and gas.

(B-3) $$V_t = f + \left(\frac{1}{c_o} - 1\right) + \frac{KGRT\ (FGOR)}{p}$$

In the above equation the following symbols are defined

V_t is the total volume of oil, water and gas associated with one barrel of oil passing the ij^{th} stage, at prevailing pressure and temperature.

f is the oil shrinkage factor at prevailing pressure and temperature.

K is a constant which depends upon the units being employed.

G is the specific gravity of the gas.

R is the constant for the hydrocarbon gas in consideration.

T is the temperature in the ij^{th} stage (deg. Rânkine).

FGOR is the free gas/oil ratio.

p is the pressure in the ij^{th} stage.

The specific weight of the reference volume is found by determining the corresponding weight and forming the quotient (weight per volume). This specific weight is then employed in equation B-1 to determine the manner in which pressure is added to the fluids as they pass through the pump stages.

Various assumptions can be made concerning the mass flow rate through the pump. The simplest and most useful assumption is to presume that fluid mass passes a given stage at a uniform rate such that oil, water and gas proportions are preserved. As pressure is increased, all or a portion of the free gas is assumed to be re-dissolved according to a liberation curve.

The above computation process is tedious and is best performed with a digital computer. A computer efficient scheme for determining head as a function of rate can be developed as follows. Since the reference volume of fluid passing through the pump is one barrel per day, the instantaneous rate passing through the ij^{th} stage is

(B-4) $$q = V_o V_t$$

V_o is the daily oil production rate in BPD. For a given type of downhole pump, there is a known relationship between head and rate. This relationship is available from the manufacturer. Imagine that head is known at equal increments of rate Δq starting at zero. Thus, head h_{ij}^{n} is known at discrete values of rate such that

(B-5) $$q_n = n\Delta q \quad ; \quad n = 0,1,2, \ldots \bar{n}.$$

Head can be found for an arbitrary value of rate q using the following simple procedure.

(B-6) $$n = \text{integer value of } \left(\frac{q}{\Delta q}\right).$$

Thus the head h_{ij} corresponding to a flow rate q through the ij^{th} stage can be found from simple interpolation as

(B-7) $$h_{ij} = h_{ij}^{n} + \frac{(q - n\Delta q)}{\Delta q} \{h_{ij}^{n+1} - h_{ij}^{n}\}$$

This involves only nine arithmetic operations which is time-efficient for the computer.

In designing a pump, enough stages and tapers are used to satisfy the following relation

(B-8) $$PIP + \Delta p \geqslant p_t$$

In this relation, p_t is the pressure at the bottom of the tubing just above the pump, and PIP is the pump intake pressure.

Owing to the discrete nature of the pressure added by each stage, the above relation should read "enough stages should be used to add enough pressure to equal or just exceed the pressure at the foot of the tubing".

Using a similar process, the power requirement of each stage is summed to determine the power requirement of the prime mover. When this is found, electrical data such as shown in Figure 6 are used to anticipate electrical performance of the system.

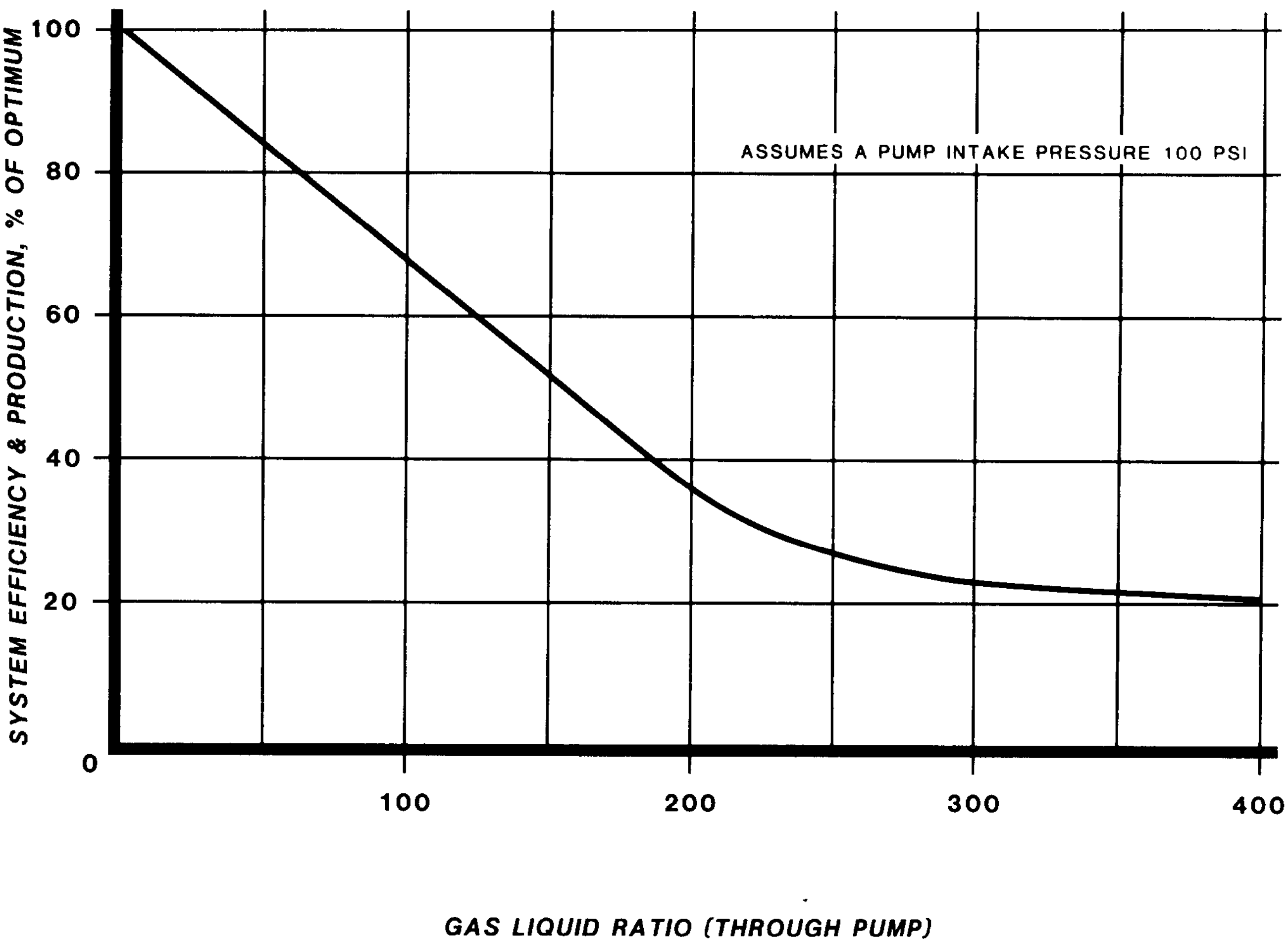

FIGURE 1 — EFFECTS OF FREE GAS ON PRODUCTION AND SYSTEM EFFICIENCY

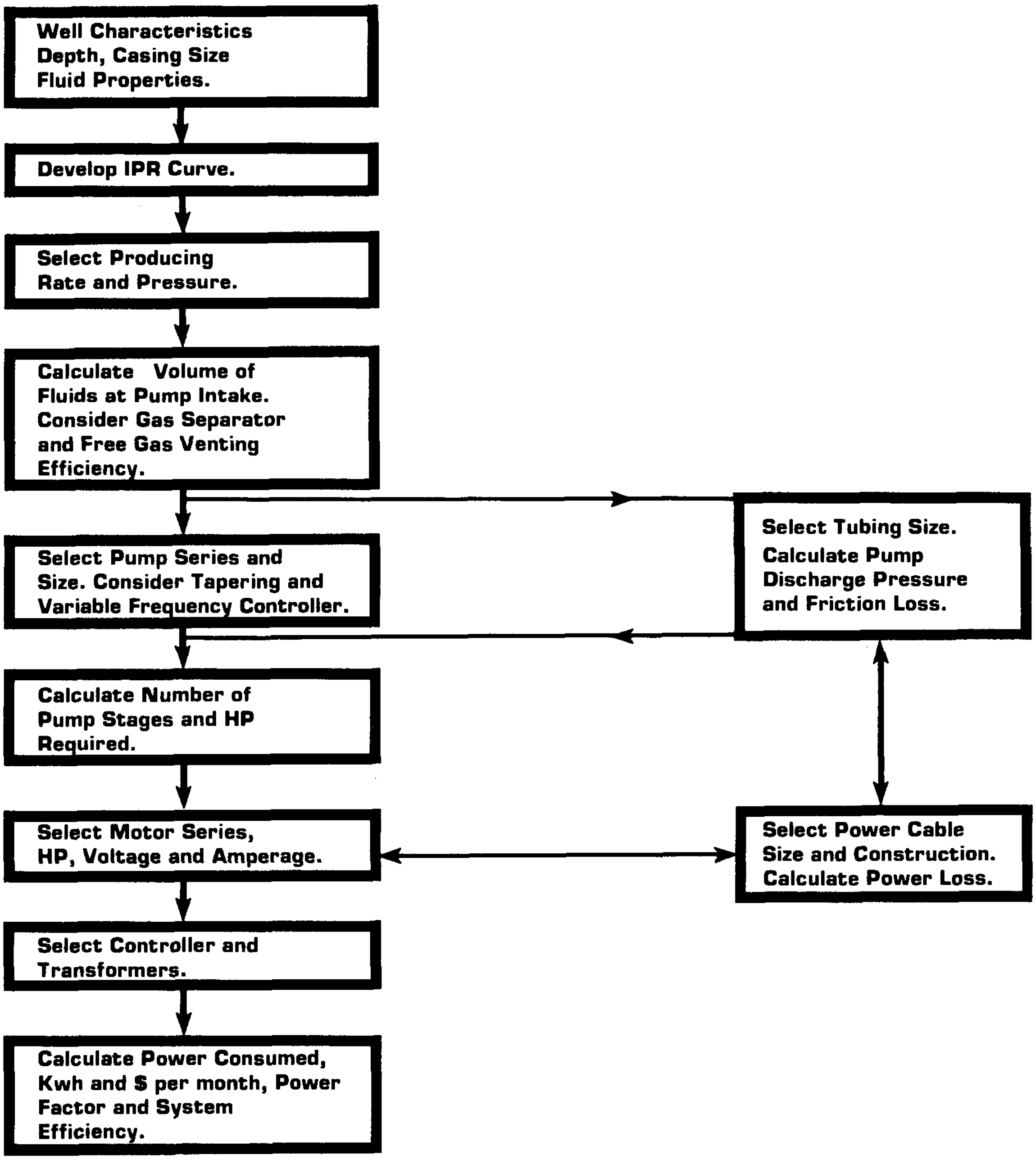

FIGURE 2 — LOGIC DIAGRAM FOR SUBMERSIBLE DESIGN

WORK TABLE FOR DETERMINING PUMP INTAKE CAPACITY [APPROXIMATE]

Pressure (psi)	BWPD	BOPD @ S. Tank	Shrinkage Factor	BOPD @ Pump	SCFGPD Total	SCFGPD Free *	BPD ** Free Gas @ Pump	Required Pump Intake Capacity (BPD)

* Do not include vented gas.

$$** \ BPD = \frac{3 \times SCFGPD}{\text{Gauge Pressure} + 14.7}$$

Assumptions:

Ideal gas i.e. $\frac{PV}{T} = C$

Temperature at pump intake = 140°F

FIGURE 3

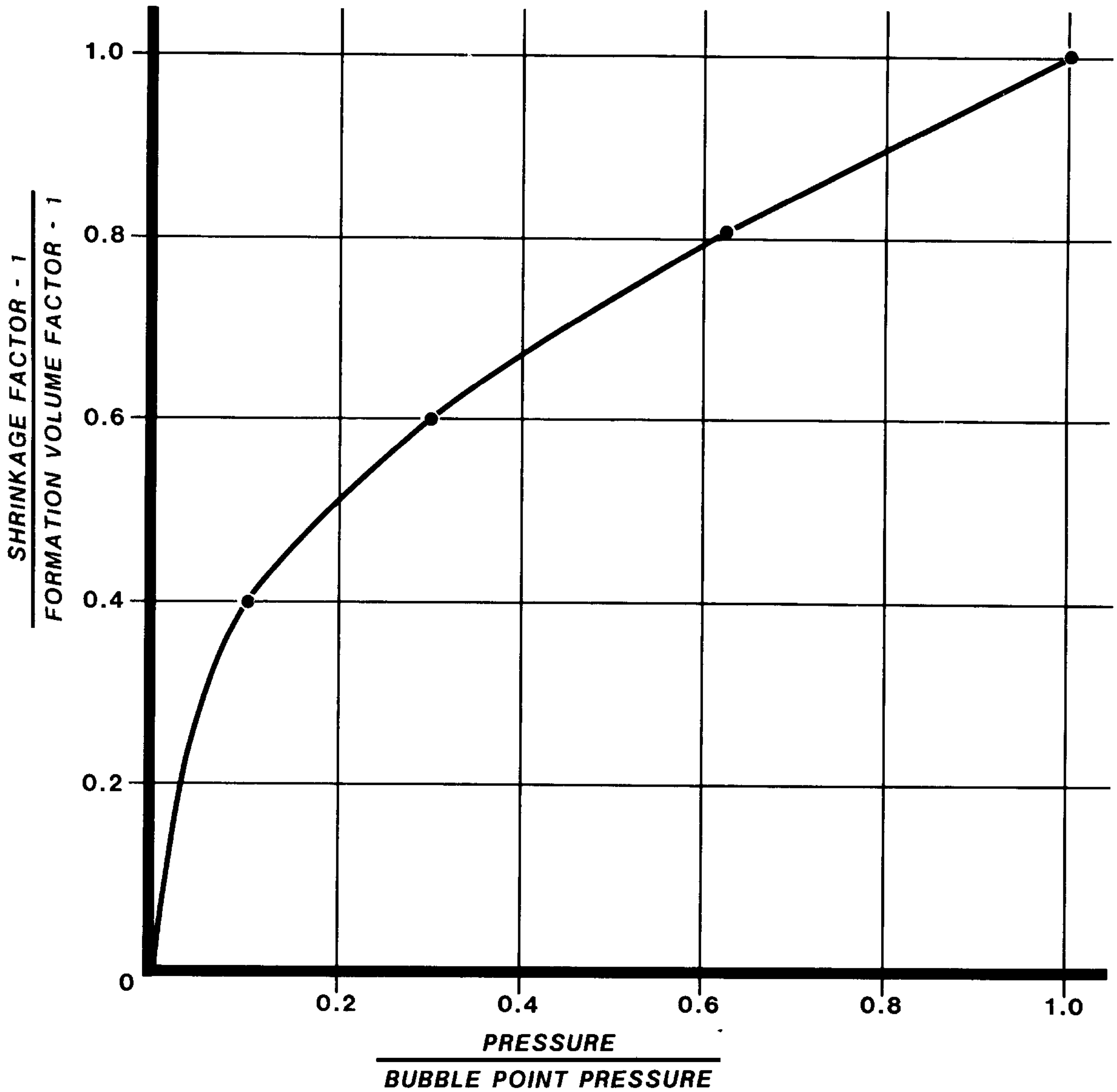

FIGURE 4 — NONDIMENSIONAL SHRINKAGE CURVE. (BASED ON PERMIAN BASIN CRUDES)

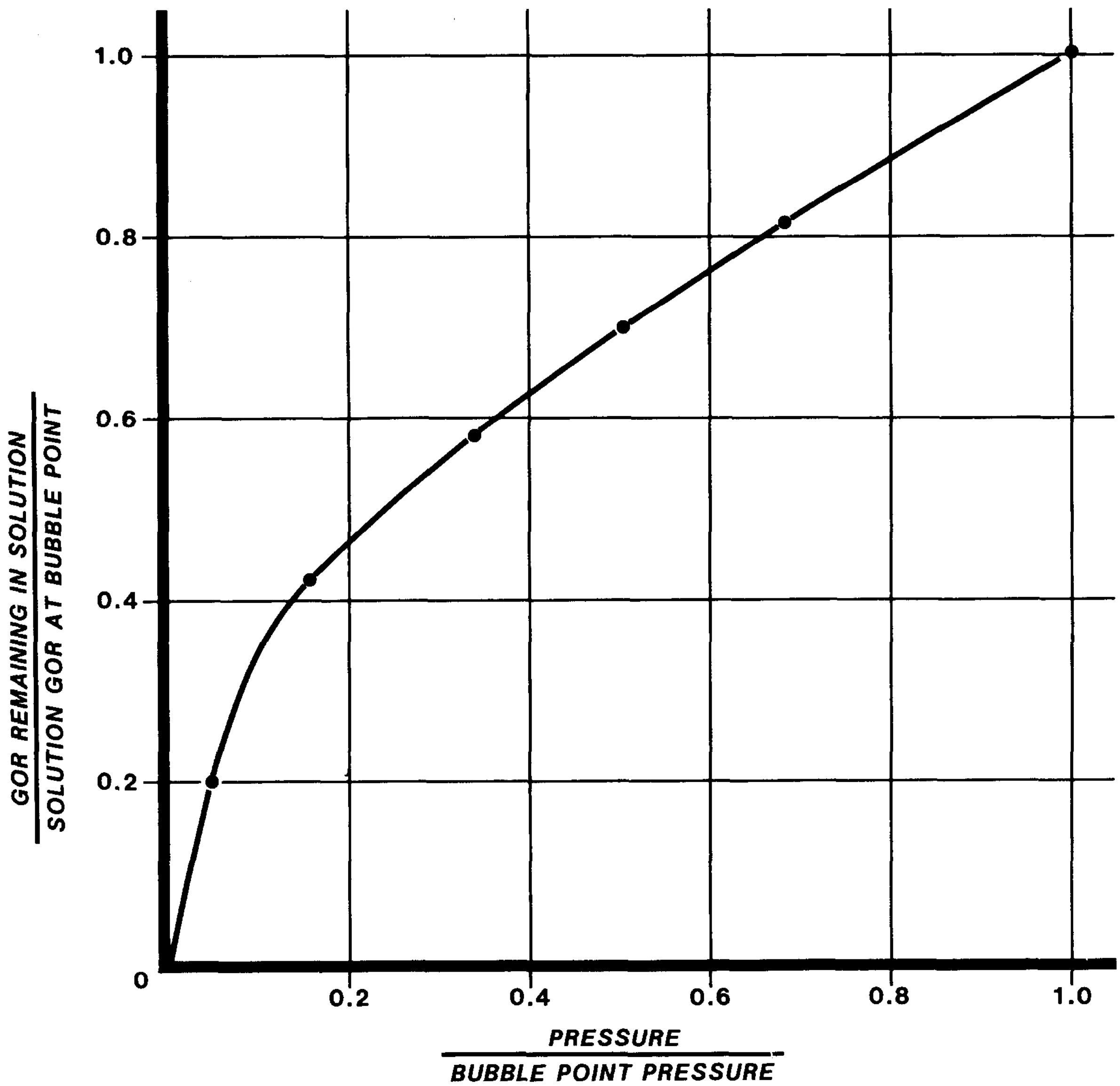

FIGURE 5 — NONDIMENSIONAL LIBERATION CURVE. (BASED ON PERMIAN BASIN CRUDES)

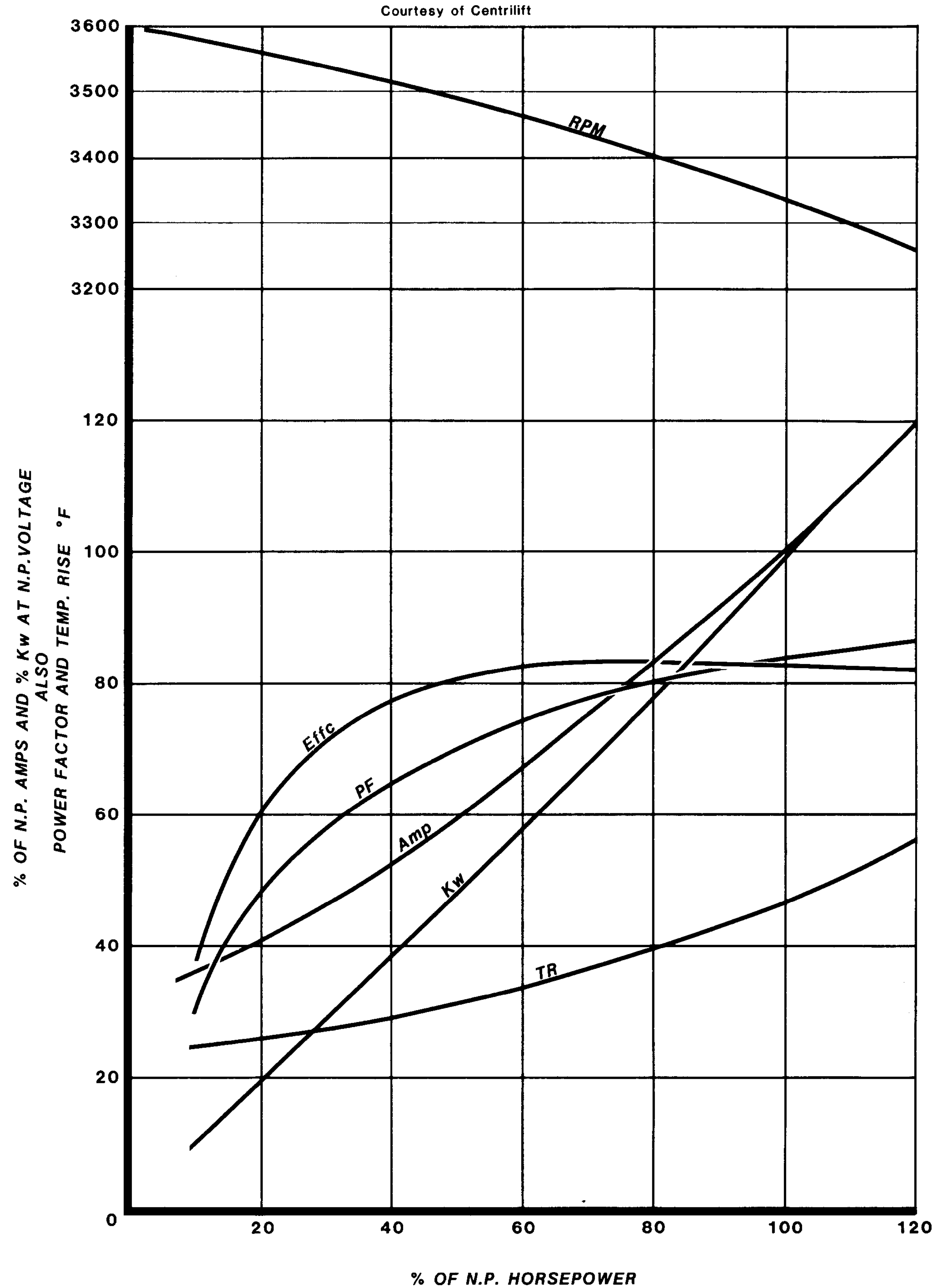

FIGURE 6 — CENTRILIFT MOTOR PERFORMANCE DATA

	Pump/stages	PIP psi	Prod. BFPD	GOR	Hertz	Motor Output HP	Pump Output HP	Pump Effec. %	System Effec. %	Comments
Design 1	M-34/227	200	1200	0	60	46.5	31	67	49.3	No free gas. Base case.
2	M-34/227	512	888	458	60	16.2	3	19	11.6	Gassy.
3	M-34/327	509	891	458	60	22.8	3	13	9.4	Gassy. Overstaging.
4	M-34/227	364	1036	458	90	34.7	7	20	12.4	Gassy. Increasing frequency.
5	Z-69/227	266	1134	458	60	16.2	9	55	32.0	Gassy. Higher capacity pump.
6	Z-69/100 N-80/127	224	1176	458	60	18.5	10	54	32.3	Gassy. Tapered pump.
7	M-34/227	200	600	0	50.8	27.1	15	55	39.1	No free gas. Low volume, low frequency.
8	M-34/227	200	1800	0	76	85.4	47	55	41.1	No free gas. High volume, high frequency. 75 HP motor rating increases to 95 HP at 76 Hz.

Note: A 75 HP motor (1130 volts, 43 amps) and a size 4 power cable are used in all designs.

FIGURE 7 — SUMMARY OF DESIGN RESULTS