Oiljetpump.com

Summary of Oil Jet Pump Principles & Applications mostly compiled from various (SPE papers #-ref)

[Contact SPE Dallas Texas for full reprint of reference papers]

Otis P. Armstrong P.E.

[2010]

#21117 Dollar F.O. Nat'l 1990

This not only improved downhole operations, but also made the fluid easier to handle on the surface. With the space confinements using this tubular arrangement, producing volumes are limited. Production rates in excess of 1,500 BPD have been reported.

Later configurations with cased holes allow for much higher volumes to he achieved (Figure 3). This configuration uses normal DST tool with a sliding sleeve in the drill string just above DST tools. A Jet Pump can be lowered into the open sleeve where it is sealed off at top and bottom and locked into the sleeve. Power fluid (normally water) is pumped down the casing through the jet nozzle and back up the drill string. This allows a downhole safety valve to be activated with power fluid while operating the jet pump. It also allows the produced fluids and gas to be confined by the drill string. The rig mud pumps, using water, are used as a surface power source for operating the jet pump. Production rates in excess of 16,000 BPD have been achieved.

#19713 Hatzlavramidis D.T. 1991

1. The governing equations for the liquid/liquid pump were derived and shown to be identical to those recently derived independently by other investigators. It is also shown that the equations used by some designers are limited to fluids of equal density.
2. A number of alternative methodologies, iterative and direct, were proposed for selecting the optimum size and flow-rate pump for a given production rate.
3. A critical review of design correlations was made and numerous modifications regarding the dependence of the pressure-loss coefficients on area ratios, flow rates, and fluid viscosities were suggested.
4. It was proposed that design of a pump that handles well fluid containing gas be considered an intermediate case between the liquid/gas and liquid/liquid pumps, while the design of a pump that handles power fluid containing gas should be considered an intermediate case between the gas/liquid and liquid/liquid pumps.
5. Governing equations for the gas/liquid pump were derived and an energy analysis was made for the same pump.
6. Results of calculations of compression ratios and efficiencies for all three pump types (liquid/gas, liquid/liquid, and gas/liquid), operating under conditions of negligible mass-flow-rate ratio and friction, were compared to determine the effect of gas and its location (as power or as well fluid) on performance.
7. In general, the presence of gas in either the power or well fluid reduces pump efficiency. When gas is present in the well fluid, the overall compression ratio of the pump is lower than that of a pump with gas in the power fluid operating under identical conditions.

#37427 Nornaha FAF et al 1997

A new model is proposed to predict performance of hydraulic jet pumps, HJP, when pumping two-phase gas liquid mixtures. The model performance is compared with Petrie et al. (World Oil, Nov. 83) and Jiao et al.'s (SPEPE, Nov. 90) model, using as input data the set of measurements taken by Jiao (1988) when experiencing an industrial HJP The present model results from the application of the one dimensional conservation law of mass, momentum and energy to the gas-liquid flow throughout the HJP. It differs from the previous published ones because it takes into account the flow of the two-phase compressible homogeneous mixture along the different parts of the device. It is not just an adaptation of a model originally developed for single phase flows, as the existent ones. Until now, most models representing the flow of a gas-liquid mixture in a HJP were adapted from models developed for incompressible single phase flows. The gas compressibility, for instance, is fully considered, as in Cunningham's paper (J. Fluids Eng., Sep. 74), when modeling a jet pump driven by a gas. The solution of the proposed model requires the knowledge of two constant energy dissipation factors, for the flow inside the nozzle and throat. Best values for these factors were obtained performing a regression analysis over Jiao's data[3]. The results showed a consistently better agreement with the experimental data than those delivered by the existing models, over the full range of the operational variables.

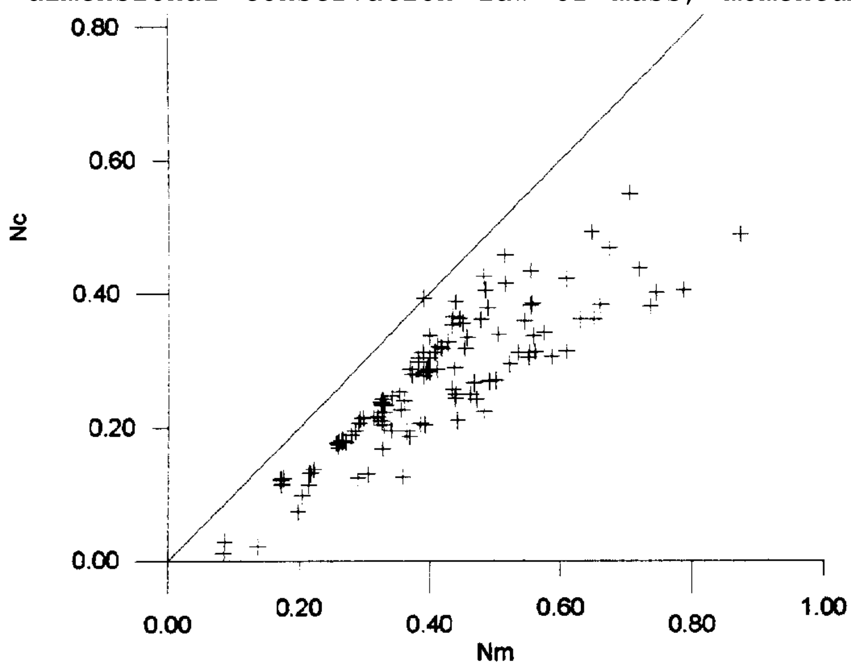


Fig. 10 - Petrie et al.´s model. Dimensionless pressure gain: measured x calculated values.

Comment

Authors fail to provide calculation examples. Moreover, 66 data points gave an $M > 0.8$, either at the suction section exit or inside the throat, before the shock region. Because greater values of k_s make things worse, these 141 points, under the strong suspicion of choked flow, were discarded and excluded from further analysis. Therefore, 123 data points were left for adjustment and verification of the model.

Additional self contradiction is: There are thousands of jet pumps installed in oil wells, most of them suctioning two-phase gas-liquid mixtures. However, some models developed to predict the pump performance when the produced fluid is a gas-liquid mixture'[2] are mere adaptations of single-phase flow models. : if these prior models so poor why so many pumps working without such an acclaimed improved model?

No new data is introduced. Rather data from prior industrial work is used, Jiao. Also data set was used on model stated to not be for high GOR, nor is it mentioned whether Petri calculation used recommended high GOR correction.

Claims of high correlation are only valid if selective data points are omitted as acknowledged in write up "These five experimental runs were also eliminated from further analysis." Authors claim for high model accuracy on one and only one data set, (using selective data points) while neglecting a comparison to many other available data sets nor was there development of independent data set.

Avoid this method if possible.

Summary #50490 Noronha, F.A.F. et-al 1998

A new model is proposed to predict the performance of hydraulic jet pumps (HJP's) when pumping two-phase gas-liquid mixtures. The model performance is compared with the models of Petrie et al.[1] and Jiao et al[2]. Input data used were the measurements taken by Jiao[3] when testing an industrial HJP.

The present model results from the application of the one dimensional conservation law of mass, momentum and energy to the gas-liquid flow throughout the HJP. It differs from the previous published ones because it takes into account the flow of the two-phase compressible homogeneous mixture along the different parts of the device. It is not just an adaptation of a model originally developed for single-phase flows. Until now, most models representing the flow of a gas-liquid mixture in an HJP were adapted from models developed for incompressible single-phase flows. The gas compressibility, for instance, is fully considered, as was done by Cunningham[4] when modeling a jet pump driven by a gas.

The solution of the proposed model requires knowledge of two constant energy dissipation factors, i.e. flow inside the nozzle and flow inside the throat. The best values for these factors were obtained by performing a regression analysis over the data of Jiao[3]. These results showed a consistently better agreement, over the full range of the operational variables, with the experimental data than those delivered by the existing models.

For Jiao's model[3] the mean square error of the dimensionless pressure gain was σ = 0.065. The mean square error of the dimensionless pressure gain, as calculated by the model of Petrie et al.,[1] was σ = 0.161. Analyzing the results, one finds that the proposed model gives a better performance than those of Petrie et al.[1] and Jiao et al.[2] Although the difference in σ is relatively small when one considers the model of Jiao et al.,[2] Fig. 10 shows the proposed model representing more consistently the data over the full range. The model of Jiao et al.[2] over predicts them for the lower N and under predicts at higher N. Best values for the energy dissipation factors in the nozzle and in the throat were calculated by regression analysis over the experimental data of Jiao.[3] The values of k_n = 0.100 and k_{td} = 0.192 were obtained.

Ref.	k_s	K_n	K_t	k_d	k_{td}
this	-	0.100	-	-	0.192
???	0	0.15	0.28	0.10	0.38
4	0	0.1	—	—	0.30
13	0.036	0.14	0.102	0.102	—
13	0.08	0.09	0.098	0.102	—
1	0	0.03	—	—	0.20

Comment: no example & lacks detail, Jiao & others more detailed. Results based on selective use of data set, other methods appear to be more transparent and more widely accepted. Lacks sound theoretical basis without substantial adjustments such as neglecting introduced variables: Ks, Kt, Kd. Avoid where possible, is review comment.

6). The energy dissipation coefficients that emerged from (selected) 123 data points were: ks = 0 and ktd = 0.216. ks does not reflect well the physics of the flow. The basic assumption of a homogeneous flow in the suction section leads to a miscalculation of the fluid properties, since the volumetric gas fraction in the mixture is overestimated.

Why such paper in same format published two times?

#102546 Hesham A.M.A. et al 2006

In spite of investigations on liquid-liquid flow (most water-water), none of them examined the case in which the secondary flow liquid

$$N = \text{Numerator} / \text{Denominator} \;\ldots\ldots.\; (1)$$

Where:

Numerator =

$$[1 - K_{diff} - (A_t/A_{diff})^2] +$$
$$2[(1+M)^{-1}\{R^{-1}(1+\Gamma M)^{-1} - 1\} +$$
$$M(1+M)^{-1}\{\Gamma M(1-R)^{-1}(1+\Gamma M)^{-1} - 1\}]$$
$$- K_t - \Gamma M^2\{(1+M)^{-1}(1-R)^{-2}(1+\Gamma M)^{-1}$$
$$Cd_s^{-2}\}$$

Denominator =

$$(1+\Gamma M)(1+M)^{-1}[R^{-2}\{Cd_n(1+M)\}^{-2}]$$
$$-[1 - K_d - A_t/A_{diff})^2] - 2[(1+M)^{-1}$$
$$\{R^{-1}(1+\Gamma M)^{-1} - 1\} + M(1+M)^{-1}$$
$$\{\Gamma M(1-R)^{-1}(1+\Gamma M)^{-1} - 1\}] + K_t$$

differs from the power flow liquid in density and viscosity, as done by this research. The subject was treated experimentally on a special test rig, with the primary jet water and the secondary different types of oils. Performance of the jet pump and static wall pressure inside the mixing chamber, were measured as a function of the mixture Reynolds number. A one-dimensional analysis was also carried out, taking into account the difference of the viscosity and density of the two liquids (each primary and secondary fluid).

Contributions of the present work are:

1. An equation for performance of a jet pump when the viscosity and density of the primary fluid is different from those of the secondary fluid.
2. Proposing pressure loss coefficient in the mixing chamber due to turbulent of mixing, as a function of Reynolds number.
3. Turbulent mixing for the axial and radial velocity for both carrier and carried phases. This was determined numerically using the assumption of Two-phase, Two- dimensional flow.

Where:

$Re = (\rho_m V_t D_t)/\mu_m$	$V_t = (Q_n + Q_s)/At$ $\Gamma = 1/\gamma$	$\rho_m = ((\rho Q)_n + (\rho Q)_s)/(Q_n + Q_s)$
$K_t = 377.5/Re^{0.63}$	γ=Density ratio = ρ_s/ρ_n	$\mu_m = \alpha\mu_Q + (1-\alpha)\mu_w$
A Area D= Diam	ŋ=Efficiency=100M x N,%	$M = m_s/m_n = (\rho Q)_s/(\rho Q)_n$
ρ=fluid Density	α=Void fract.=Qs/(Qs+Qn)	$N = (P_d - P_s)/(P_n - P_d)$
μ=Dynamic visc.	μT= Turbulent viscosity	R=Area Ratio = An/At
ε = Dissipation	rate of kinetic energy	$T = 2(P - P_5)/(\rho_m V_t^2)$
Q=Vol Flow rate	X-length mixing chamber	ν = Viscosity c-stoke
Cd=Dischg coeff	P = Pressure m=Mass rate	σ Empirical constant [12]
Subscripts:	K = Pres loss coeff	R_μ= Visc. ratio = μ_s/μ_n
d = delivery	t=mixing chamber	S = kinetic turbulence
ms = measured	n = nozzle or normal	o = oil; & s = suction
diff= diffuser	m = mixture; w= water	th = theoretical

#102546 Hesham A.M.A. et al 2006

The pump dimensions were calculated according to the best ratios developed by previous research [3] for R = 0.34 in a similar system (nozzle diameter = 7 mm & throat diameter = 12 mm). The power liquid was always water, while three types of liquids were used as suction fluid: (1) water. (2) light crude oil (SG=0.85 & v = 29.4 centistokes at 25 °C) and (3) viscous crude oil (SG=0.98 & v = 357 centistokes at 25 °C).

With heavy oil, full stabilization for the change of void fraction is obtained outside the mixing chamber. This means that mixing process does not happens completely inside the mixing chamber and its length have to be longer than that used in the present work to perform proper mixing between water as a power fluid and heavy oil as |a secondary fluid. Noting that, X/D_t = 10 for the mixing chamber used (X = 70 mm &D_t = 7 mm).

Calculation of M versus N & ŋ were carried out for the cases of water-water flow and water-oil flow, assuming Cd_n = 0.96, Cd_s = 0.9 and K_t =$378/Re^{0\ 63}$ to estimate the impact of the velocity and density on performance. Fig.(2) shows that at the highest efficiency, both M and ŋ are increased by increasing γ but the pressure ratio N does not change appreciably.

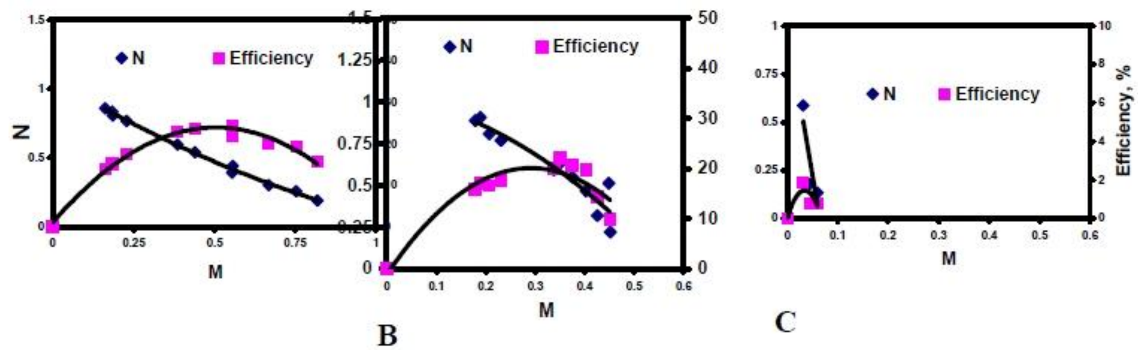


Fig.6 Experimental jet pump performance: (A) water/water, (B) water/light oil & (C) water/heavy oil

#4539 Bell A C et al 1973

This paper discusses hydraulic lift system installed by the ARMCO-Kobe and Continental Oil Company. This system combines a new venturi, hydraulic bottom-hole pump and a compact, centrifugal power fluid cleaning system into a single installation. This jet eductor pump, or jet pump, is relatively unaffected by high gas-liquid ratio production or dirty power fluid and requires very little maintenance. The fluid cleaning system provides power fluid which meets the quality requirements of the jet pump for considerably less initial investment and lower operating costs than a conventional power fluid system. This combination is particularly adaptable to single well leases or fields.

Currently, two of these compact systems are in operation producing sour crude from the deep Smackover formation in Mississippi. Excluding the problems corrected during an initial startup period, no system operating problems have been reported after a cumulative twenty-nine months of service. ROT: Twenty five percent submergence 12,000 ft pump needs 3000 ft static head equivalents to not cavitate.

#4539 Bell A C et al 1973, con'td

FIELD	East Barber Creek	West Nancy
WELL	C. G. Henderson #1	Unit 5-5 §1
FORMATION	Smackover Dolomite	Smackover Lime
PERFS.	14,820-830'	13,832-841
CASING	5 1/2"	7"
TUBING	2 3/8" DSS-HT	2 7/8" DSS-HT
TUBING @	14,750'	13,700'
CRUDE GR.	40.4API	40.3° API
ALLOWABLE	500 BOPD	400 BOPD
PRESSURE INFORMATION		
WELL	HENDERSON 11	UNIT 5-5 #1
Shut-In Bottom Hole	Pressure:	
Date:	7/71	2/72
Pressure (PSI):	5920	4081
Producing Bottom Hole Pressure:		
Date:	6/72	5/72
Pressure (PSI):	1197	3100
Prod. Rate:	75 BPD	900 BPD
RECENT WELL TESTS.		
WELL	HENDERSON #1	UNIT 5-5 #1
Date	11/72 6/73	11/72 6/73
Oil (BPD)	195 165	407 405
Water (BPD)	9 7	0 0
GOR (CFPSTB)	583 654	876 860
Power Oil Rate (BPOPD)	1700 1600	1050 1150
Power Oil Press.(PSI)	4300 4000	2400 2900

Date	Production	Solids >15 micron*	Salt
	Power Oil	(mg/1)	(#/1000 bbls)
7/72	Prod.	40	0.2
7/72	P.O.	39	0.1
8/72	P.O.	310	52.5
8/72	Prod.	360	94.0
8/72	P.O.	253	34.0
11/72	Prod.	120	1.0
11/72	P.O.	80	1.0
1/73	Prod.	898	288.2
1/73	P.O.	281	136.2
3/73	Prod.	252	48.0
3/73	P.O.	121	21.5
5/73	Prod.	316	68.0
5/73	P.O.	138	23.0

#21117 Dollar F.O. Nat'l 1990

DST is not new to the oil industry. One recent development is use of the Jet Pump to improve producing capabilities and make DST more meaningful and more accurate.

The Jet Pump is a form of artificial lift being used when the well does not have sufficient bottomhole pressure to flow adequate volumes to the surface to evaluate the reservoir being tested.

By running a sliding sleeve above normal DST tools, the Jet can be inserted into the sleeve where it locates, seals, and is locked in place by wireline.

The use of mud pumps with water provides the power source to operate the downhole Jet Pump.

The Jet Pump has the capability of producing volumes of less than 100 barrels per day (16 cubic meters) to several thousand barrels per day. Since the equipment required (in conjunction with normal DST tools) is minimal, the use of the Jet Pump becomes very attractive, both operational and cost wise. The Jet Pump can handle GLR up to 1,000:1, abrasive fluids, corrosive fluids, low and high volumes.

There are several configurations being used. Some of the first DST jets were used off the United States west Coast to lift heavy crude in offshore applications (Figure 2). In this configuration, the jet is made up on coil tubing and lowered inside the drill pipe to a predetermined depth where the lower end of the jet is packed off in the drill pipe. Power fluid is pumped down the annulus between drill pipe and coil tubing with produced fluids and spent power fluid being returned up the coil tubing. Several things were done with power fluid to reduce viscosity and friction of the low gravity crude, such as heated water, water diesel mixtures, anti-friction chemicals, and chemicals to emulsify the oil and water.

#63042 Lea JF 2000

Advantages	Disadvantages
Piston pump can pump a well down to low pressures. However, the jet pump does not pull the well down to low pressures.	Requires a second string to vent gas or for power fluid. Closed vented system may have three lines downhole.
Easy adjustable at surface to change in well productivity	Rate limited in 7 in. casing at approximately 1000 BFPD (as long as gas has to be vented).
Can be used with closely spaced wellheads.	Power oil systems are environmental and safety hazard. Will switch over to water after breakthrough.
Crooked holes present minimal problems.	Power fluid treating required to extend the life of surface and downhole pumps .
Good energy efficiency, but some of that lost when horizontal ESP is used for pore fluid pressurization.	Difficult to obtain good well tests in low volume wells.
Power source can be remotely located.	Cannot run production logs while pumping.
Flexible, can usually match well production as well declines.	Vented installations are more expensive because of more tubulars required.
Paraffin easily prevented by heating or chemically treating the power fluid.	Treating for scale below packer is difficult.
Downhole pumps can be circulated out or retrieved by wireline when worn.	High operating cost if Triplex pumps are used to pressurize the power fluid for individual wells.
Adjustable gear box offers more flexibility for Triplex power systems.	Gassy wells usually lower volumetric efficiency and shorter pump lift.
	Wellhead will freeze during shut-in (high WOR wells).

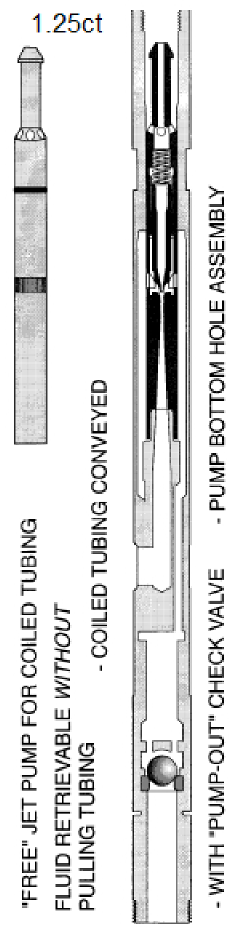
#63042 Lea JF 2000

Shown in Fig. 6 is a downhole schematic of a 1-1/4" Jet Pump. Case History Listed here is short summary of using jet pump to move fluids out of a well.

One (1 in.) coiled tubing is used as a power string. 1-1/4 in. coiled tubing is used at the return string and 2 7/8's tubing is used for gas flow. For 2500 ft. depth, the horsepower is 26.4 for 100 bpd liquid with 4000 psi needed as power fluid pressure.

Case (2): 1-1/4 in. coiled tubing is used for the power fluid string. 1 in. coiled tubing is used as the return siring and 2-7/8 in. tubing is for gas production. The horsepower required is 13.9 at 2600 psi power fluid for 100 BFPD total fluids production. The depth is 2500 ft. This is a "pump" installation.

Case (3) 1-1/4 in. coiled tubing is used for the power fluid. 2-7/8 in. is used for the returns of mixed flow, and a 1 in. vent string is used for gas venting. Twelve horsepower pump was needed to produce from 2500 ft. with 2250 psi power fluid.



Case (4): 1-1/4" coiled tubing inside 2-3/8's from 3800 ft. producing 250-300 bpd.

Case (5): 2-1/16 in. power fluid coiled tubing inside 3-1/2 in. tubing producing 1200 BFPD with 0.950 in. O.D. jet pump from 5200 ft.

Case (6): 1-1/2 in. concentric to 3-1/2 in. tubing. The gas is produced up the casing annulus. The production and power fluid returns up the tubing annulus. The production rate is 1800 BWPD. The pump is 12 in. long with a 1.12 in. O.D.

Approximate Costs of System. Case (I): depth 7.000 ft. This well was sidetracked horizontally 500 ft. This oil well is pressure depleted. Installation was a piston hydraulic pump system, which produced 300 BFPD with 48 hp and 2990 psi power fluid being pump at 860 BFPD. This application takes two strings: 2-3/8 in. for power fluid and 1-1/2 in. return string, which allowed for annular gas flow.

The jet pump will produce only 80% of above needing 600 BFPD of power fluid at 4000 psi from a 45 hp system with CT inside regular tubing.

Costs

- $37,700 for surface unit with triplex, transmission, motor (electric) and 2 vessels.
- $4,950 for wellhead with 4 way valve, ball valves, screen etc.
- For Jet: $8,500 for downhole equipment
- For reciprocating pump equipment:
- about $13,700 for downhole equipment
- Typical Rent- 10% of purchase/month with 60% of rental applying to purchase.

#97511 Chen, S et al.2007

Jet pumping driven by light oil is one of the preferred lift methods for producing heavy oil in a deep heavy-oil reservoir. Generally, the amount of light oil required is too large to be acceptable. One solution which reduces the amount of light oil required is to blend light oil with a portion of the produced fluid at a reasonable ratio. Then, the produced fluid/light-oil mixture is re-injected into the well as the power fluid. In this case, the viscosity of the blended power fluid keeps increasing and eventually reaches its equilibrium value, which has been found to be a function of reservoir-oil viscosity, light-oil viscosity, the ratio of light oil to blended power fluid (volumetric percentage), and the ratio of well rate to diluents rate (M ratio). Moreover, an optimal ratio of light oil to blended power fluid can be determined by using an iterative algorithm developed in this study. Variations in any of the previously mentioned parameters, especially the viscosity of light oil and the ratio of light oil to blended power fluid, result in a significant change in both the viscosity of the blended power fluid and the pressure loss in the production string. It has been shown in a field application that the amount of light oil used for driving the jet pumping operation can be reduced by optimization.

Introduction

It is difficult to produce heavy oil from wells deeper than 3000 m using conventional artificial lift methods (Christ and Petrie 1989). When pumping oil from a deep heavy-oil reservoir, the sucker rod-pump method undergoes rod stretch and breakage. The submersible-pump method suffers from high temperature and thrust bearing loads at high discharge pressures: furthermore, pump efficiency is greatly reduced at low production rates. The gas lift method requires a sufficient and sustainable gas source. In addition, it is expensive to compress the gas to high pressure and difficult to achieve low submergence. Therefore, more efficient methods must be sought for producing oil from deep heavy-oil reservoirs.

The jet pumping method has been proposed as an efficient artificial lifting technique for heavy-oil production (Cunningham 1957: Petrie et al. 1983a: Petrie et al. 1983b: Petrie et al. 1984: Tjondrodiputro et al. 1986: Tjondodiputro et al. 1987). In principle, a low-pressure fluid in the reservoir is boosted and produced by blending it with a high-pressure fluid pumped downhole from the surface (Fig. 1). Furthermore, the jet pumping method has advantages for producing oil in deep wells because of its simplicity, lack of moving parts, small size of pump, and ability to pump fluids with high viscosity and/or strong corrosives. In addition, light oil can be used as the power fluid in deep heavy-oil wells because it reduces the produced fluid viscosity and the pressure loss in the production string. The reduction of the pressure loss can be mainly ascribed to the instantaneous and thorough blending of the power fluid and the reservoir fluid in the jet pump throat, Ghetto and Giunla 1994).

#97511 Chen, S et al.2007

A jet pump is a dynamic pump with a performance curve similar to that of a centrifugal pump (Brown and O'Brien 1980: Zhang 2000). As shown in Fig. 2. When light oil is used as the power fluid, the amount of light oil should be enough, not only to reduce the viscosity of the reservoir fluid in the production string, but also to provide sufficient energy to lift the reservoir fluid to the surface. To maximize lifting flexibility and efficiency, it is advisable to operate the pump with a high R ratio (nozzle area/throat area). This will provide a high N ratio, $(P_D-P_S)/(P_N-P_D)$, in deep heavy-oil wells, where P_D, P_N, and P_S are pump discharge pressure, power fluid pressure, and pump suction pressure, respectively (all pressures measured at pump depth) (Zhang 2000). In general, high efficiency can be achieved for a lower M ratio together with a higher R ratio and a higher N ratio. This study found values in the range of 0.3 to 1.2 to be suitable for a reservoir depth >4500 m. If light oil alone is used as the power fluid, the amount of light oil required is 0.83 to 3.33 times the well rate. In addition, the amount of light oil required for viscosity reduction is only approximately 0.43 times the well rate, which is defined as the fluid flow rate from the reservoir into the wellbore (Qu et al. 2000a. b). Applicability of the jet pumping method is limited if a large amount of light oil is needed and the supply of light oil is insufficient.

In this research, a new technique is developed to reduce the amount of light oil used for driving the jet pump. Specifically, a portion of the produced fluid is blended with light oil at a reasonable ratio. Then, the produced fluid/light-oil mixture is re injected into oil wells as the power fluid. The key issue with this new technique is to determine the optimal ratio of light oil to blended power fluid (volumetric percentage) so that the amount of light oil required is minimized. It is also necessary to maintain low viscosity of the blended power fluid in the production string and reasonable wellhead pressure of the power fluid.

In this paper, the feasibility of this new technique is first discussed. A theoretical model is then presented for determining the equilibrium viscosity of the blended power fluid, the time required to reach the equilibrium state, and the optimal ratio of light oil in the produced fluid/light-oil mixture. Finally, a field application case is presented and discussed.

#97511 Chen, S et al.2007

a = variable defined in Eq. 5a
A_a = cross-sectional area of the annulus, m^3
A, = cross-sectional area of the tubing, m^3
b = variable defined in Eq. 5b
D_p = height of the jet pump, m
M ratio =$Q_{\text{well rate}}/Q_{\text{power}}$, m^3/m^3
n = number of circulations
n_c = number of circulations required to achieve constant viscosity and stable production
N = $(P_D-P_S)/(P_N-P_D)$
P_D = jet pump discharge pressure, MPa
P_N = power fluid pressure at pump intake (nozzle), MPa
P_S = well pressure at pump intake (throat), MPa
R = nozzle area to throat area ratio

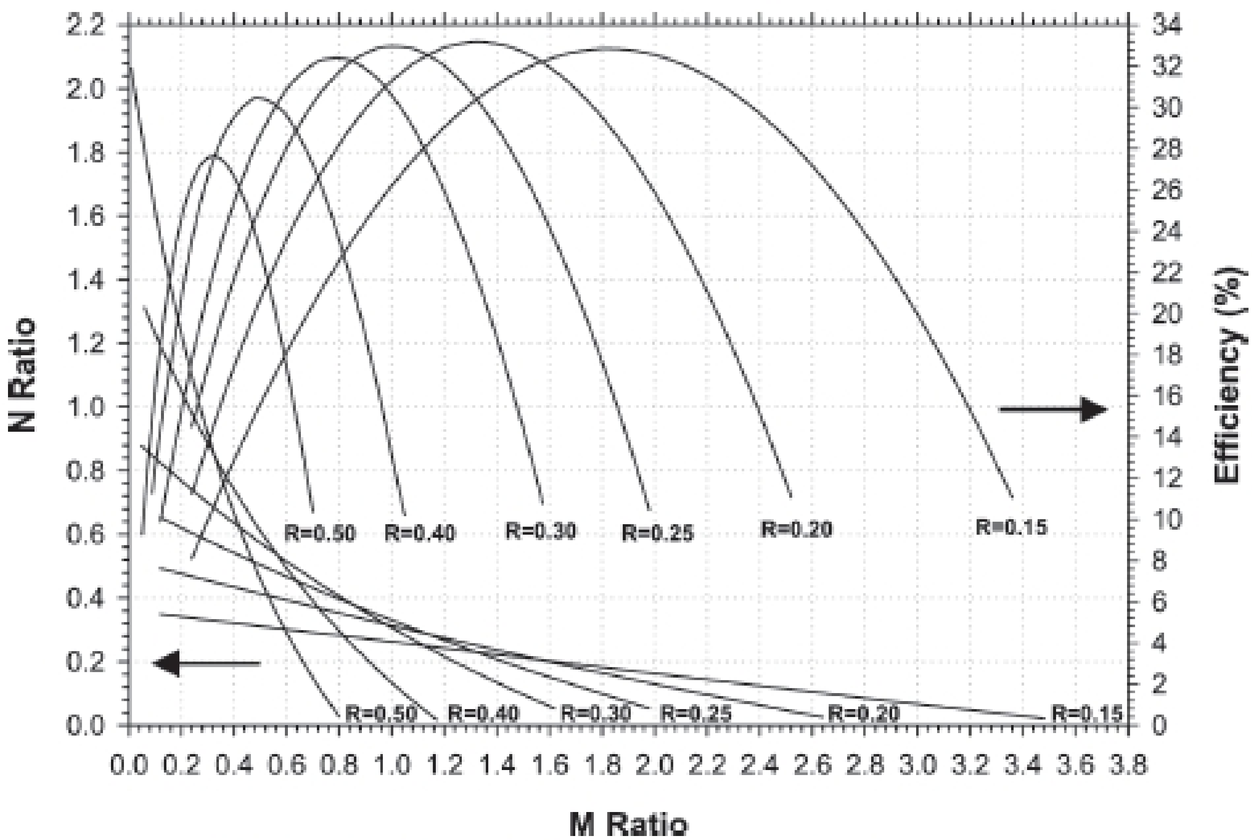


Fig. 2—Typical dimensionless performance curves for jet pumping.

#27595 DeGhetto G, et al 1994

The jet pump tests performed in Vega and Gela heavy oil wells allow us to draw the following conclusions:

- Jet pumps driven by liquid hydrocarbons were suitable to lift the tested heavy oils without operational problems.
- In packered wells equipped with a sliding side door valve, jet pumping demonstrated to be a cost-effective lifting method for replacing failed ESFs. Jet pumps can in fact be RIH / POOH by simple wire line operations avoiding the use of a costly rig.

In Vega field the cost differences were quite consistent: US$ 70000 for jet pump versus US$ 400000-600000 for ESP replacement.

- In Vega field, despite the low efficiency exhibited by the jet pumps, 20 - 30 lower than theory, the obtained lifting head was satisfactory (up to 200 psi) and sufficient for producing the level of oil rate achieved by the current lifting methods (ESP and diluents lift).
- The tests showed a very low system flexibility confirming what reported in literature. To maximize the lifting flexibility and the efficiencies it is advisable to operate the pump with M ratio (well rate / diluents rate) values in the range 1.0-2.0. The tests with M ratio values greater than 3.0 showed a non acceptable very low pump efficiency.
- To minimize operational problems (plugging, scaling, paraffin, etc.) it is advisable to select the biggest nozzle and throat area combinations from those suitable.
- Comparison between diluents lifting and jet pump tests clearly showed that the better fluid mixing obtained with jet pumps provides improved well performance because of the further reduction of friction losses (20 - 40 psi) in the production string.

	Units	VEGA FIELD	GELA FIELD
Reservoir Depth	ft	8364 - 8528	10880-11350
Oil Gravity	°API	15	7-11
Gas Gravity	air=l	1.5	1.5
Bubble Point	psia	427	611
GOR	scf/bbl	50	90
SBHT	°F	212	217
SBHP	psi	3698	4835
Oil Viscosity @ RC	cp	70	85
P.I.	BPD/psi	6-3000	2-66
WC water cut	%	20-40	5-50

#27595 DeGhetto G, et al 1994

Fig. 3: Vega Field. Well # 16 completion scheme

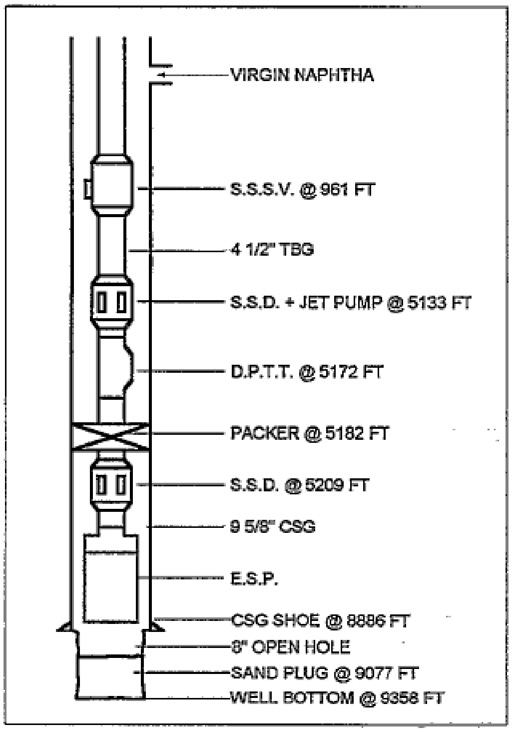
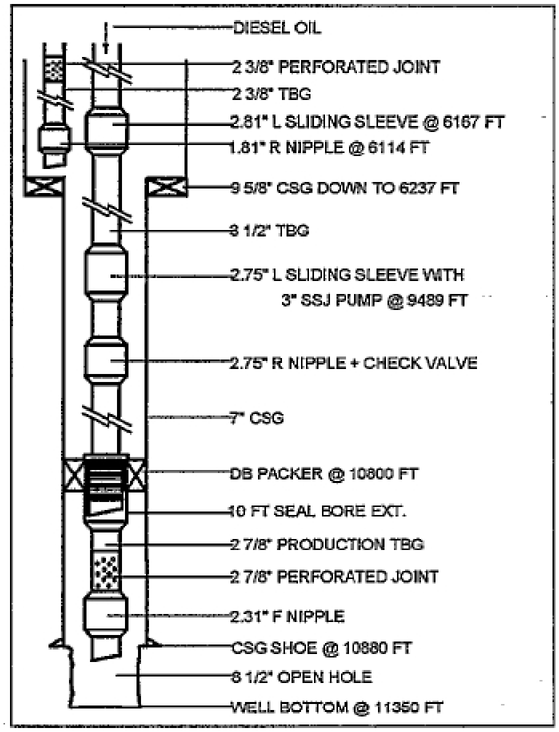


Fig. 4: Gela Field. Well # 12 completion scheme



Vega 16 well - Jet Pump 9D - Comparison between theoretical and actual performances

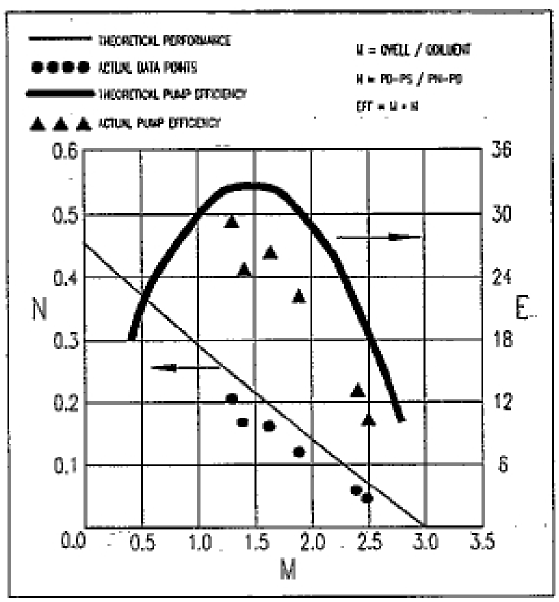
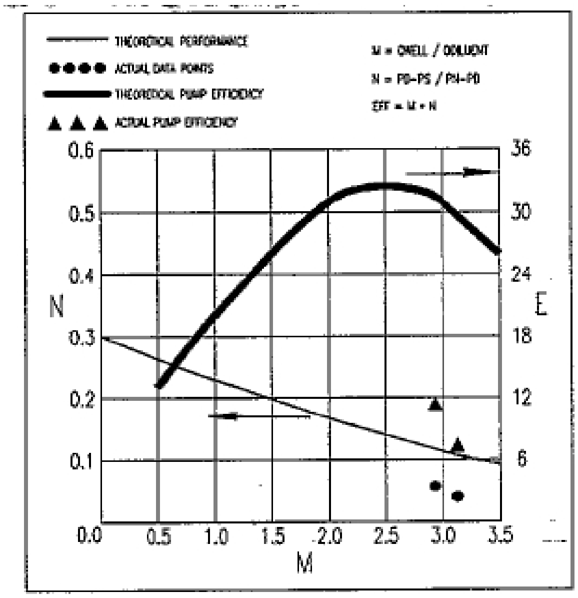


Fig. 9: Gela 12 Well - Jet Pump 2F -Comparison between theoretical and actual performances



15177 Christ FC et al 1989

Summary: Maximizing production in a well requires reducing the bottomhole pressure (BHP) to the lowest practical minimum by artificial lift. This task is particularly difficult in deep wells because of the need to transmit power a long distance to the bottom of the well and because of the high temperatures encountered at depth. Reciprocating hydraulic pumps have historically been used for this purpose in deep wells, especially those deeper than 10,000 ft. For more than a decade, the hydraulic jet pump has successfully produced deep wells. In a number of cases, jet pumps have produced deep wells to less than 10% submergence. This paper identifies the hydraulic jet pump as a reasonable method to produce deep wells to low BHP's with predictable performance. A number of examples and well tests illustrate the performance of jet-pump production systems.

Jet-Pump Performance

Nozzle area and pressure determine horsepower input. Nozzle /Throat area ratio determines the jet-pump performance. Within the throat, pressure must remain above liquid-vapor pressure to prevent throat cavitation damage. The cavitation rate is also a choked flow rate and is the limit on production obtainable at a given pump intake pressure. Fig. 1 shows the pressure history of produced fluid as it enters and travels through the working sections of a jet pump. Note that the pressure drops below pump-intake pressure as produced fluids accelerate into the throat mixing zone. If pressure drops to below liquid-vapor pressure, vapor bubbles will form. Cavitation damage is caused by the collapse (implosion) of these liquid vapor bubbles on the throat surface as pressure increases in the throat. Cavitation damage can eventually change a section of the throat, decreasing jet-pump performance.

Nozzle and throat diameters are sized so that the areas of different sizes are in a geometric progression. Table 1 shows an example. Each size is larger in flow area than the preceding size by a factor of 4/pi. Because of this sizing system, a given nozzle number with the same throat number always gives a nozzle-to-throat area ratio of 0.380. A throat number one larger than the nozzle number will always give an area ratio of 0.299. Other area ratios can also be obtained, as shown in Table 1. The characteristics of these area ratios are used to compute pump performance in a well. Area ratios are identified by X, A, B, C, D. and E. Pump size is designated by the nozzle size and the area ratio. For example, a No. 8 nozzle and No. 7 throat make up an 8X combination. A No. 8 throat with a No. 8 nozzle makes an 8A combination.

The X ratio is for high lift and low production rates compared with the power fluid rate; the C ratio is for low lift and high relative production rates. Fig 2 shows typical pump-performance curves. These curves are dimensionless, so they are valid for jet pumps of the same area ratio regardless of nozzle number.

15177 Christ FC et al 1989

"Submergence" as used in this paper is the ratio of pump-intake pressure to pump-discharge pressure. For example, 10% submergence in a 10,000-ft well producing 30°API [0.88-g/cm^3] oil is equivalent to a 380-psi pump-intake pressure and a 3,800-psi discharge pressure. This corresponds to 1,000 ft of fluid over the pump. An expression for cavitation - sets lower limit on percent submergence & can be derived following the format of Ref. 1. Fig.3 shows the working section of a jet pump and the nomenclature used. Assuming that power fluid and produced fluids are the same, the basic equations are given as

$$q_n = 832A_n\sqrt{[(P_n - P_{ps})/g_{pf}]}$$

$$R = F_{AD} = A_n/A_t$$

$$M = F_{mfD} = Q_s/Q_n$$

$$N = F_{pD} = (P_{pd} - P_{ps})/(P_n - P_{pd})$$

$$1/S = 1+[F_{pD}/(1+F_{pD})]\{[0.831(1-F_{AD})/F_{AD}]/F_{mfD}\}^2$$

$$1/S = 1+[N/(1+N)]\{[0.831(1-R)/R]/M\}^2$$

where

$A_{c\ min}$ = minimum throat annulus flow area, (A_t - A_n) to avoid cavitation, in.^{2}
A_n =flow area of nozzle, in.^{2}
A_t=flow area of throat, in.^{2}
E_p =jet-pump efficiency, fraction M times N
F_{AD} =ratio of nozzle area to throat area, dimensionless, R
F_{mfD} =mass flow ratio, dimensionless; M
F_{pD}=pressure recovery ratio, dimensionless; N
g_{pf}= gradient of power fluid, psi/ft
g_s=gradient of produced fluid (suction), psi/ft
K_n = nozzle loss coefficient, dimensionless
K_{td} = throat-diffuser loss coefficient, dimensionless
P_n= nozzle entrance pressure, psi
P_{pd} = pump-discharge pressure, psi
P_{ps} = pump-intake pressure (suction),psi
Q_n = nozzle flow rate, B/D =$832A_n\sqrt{((P_n-P_{ps})/g_{pf})}$
Q_s = pump-intake (suction), Cavitation B/D <$691A_{min}\sqrt{(P_{ps}/g_s)}$
S = submergence, fraction = P_{ps}/P_{pd}

15177 Christ FC et al 1989

TABLE 1—AVAILABLE JET-PUMP SIZES

Nozzle Number	Nozzle Area (in.^{2})	Throat Number	Throat Area (in.^{2})	Throat Annulus Area (in.^{2}) Ratio Designation X	A	B	C	D	E
1	0.0024	1	0.0064	—	0.0040	0.0057	0.0080	0.0108	0.0144
2	0.0031	2	0.0081	0.0033	0.0050	0.0073	0.0101	0.0137	0.0183
3	0.0039	3	0.0104	0.0042	0.0065	0.0093	0.0129	0.0175	0.0233
4	0.0050	4	0.0131	0.0054	0.0082	0.0118	0.0164	0.0222	0.0296
5	0.0064	5	0.0167	0.0068	0.0104	0.0150	0.0208	0.0282	0.0377
6	0.0081	6	0.0212	0.0087	0.0133	0.0191	0.0265	0.0360	0.0481
7	0.0103	7	0.0271	0.0111	0.0169	0.0243	0.0338	0.0459	0.0612
8	0.0131	8	0.0346	0.0141	0.0215	0.0310	0.0431	0.0584	0.0779
9	0.0167	9	0.0441	0.0179	0.0274	0.0395	0.0548	0.0743	0.0992
10	0.0212	10	0.0562	0.0229	0.0350	0.0503	0.0698	0.0947	0.1264
11	0.0271	11	0.0715	0.0291	0.0444	0.0639	0.0888	0.1205	0.1608
12	0.0346	12	0.0910	0.0369	0.0564	0.0813	0.1130	0.1533	0.2046
13	0.0441	13	0.1159	0.0469	0.0718	0.1035	0.1438	0.1951	0.2605
14	0.0562	14	0.1476	0.0597	0.0914	0.1317	0.1830	0.2484	0.3316
15	0.0715	15	0.1879	0.0761	0.1164	0.1677	0.2331	0.3163	0.4223
16	0.0910	16	0.2392	0.0969	0.1482	0.2136	0.2968	0.4028	0.5377
17	0.1159	17	0.3046	0.1234	0.1888	0.2720	0.3779	0.5128	
18	0.1476	18	0.3878	0.1571	0.2403	0.3463	0.4812		
19	0.1879	19	0.4938	0.2000	0.3060	0.4409			
20	0.2392	20	0.6287	0.2546	0.3896				
Combination Throat Number				Noz − 1	Noz	Noz + 1	Noz + 2	Noz + 3	Noz + 4
Nozzle/Throat Area Ratio				0.483	0.380	0.299	0.235	0.184	0.145

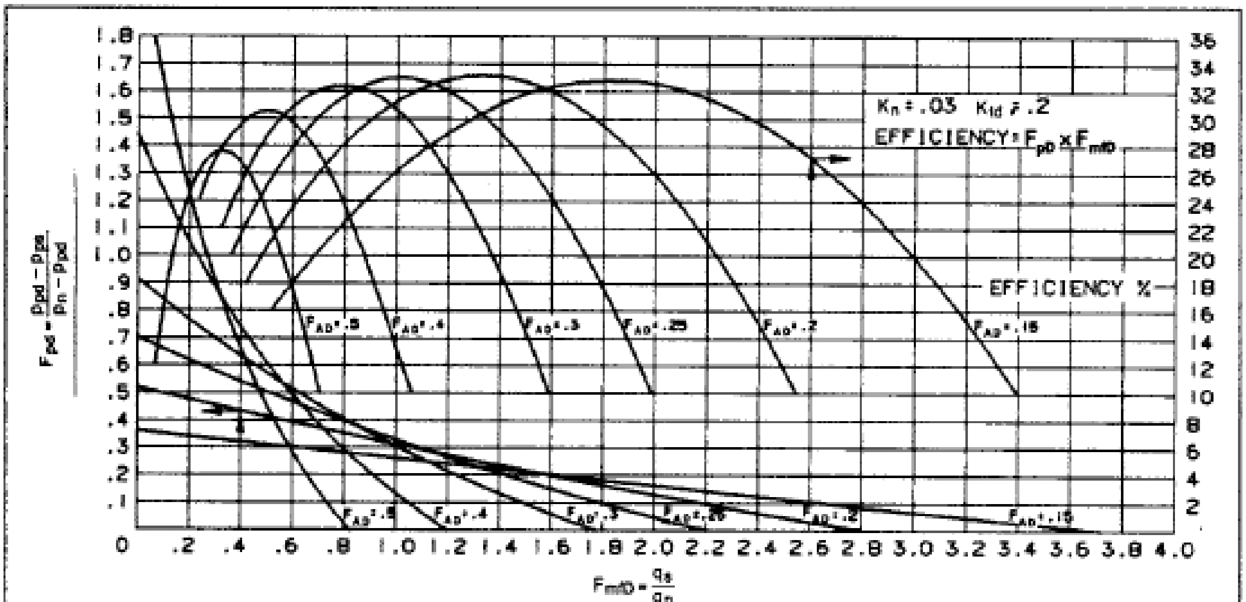
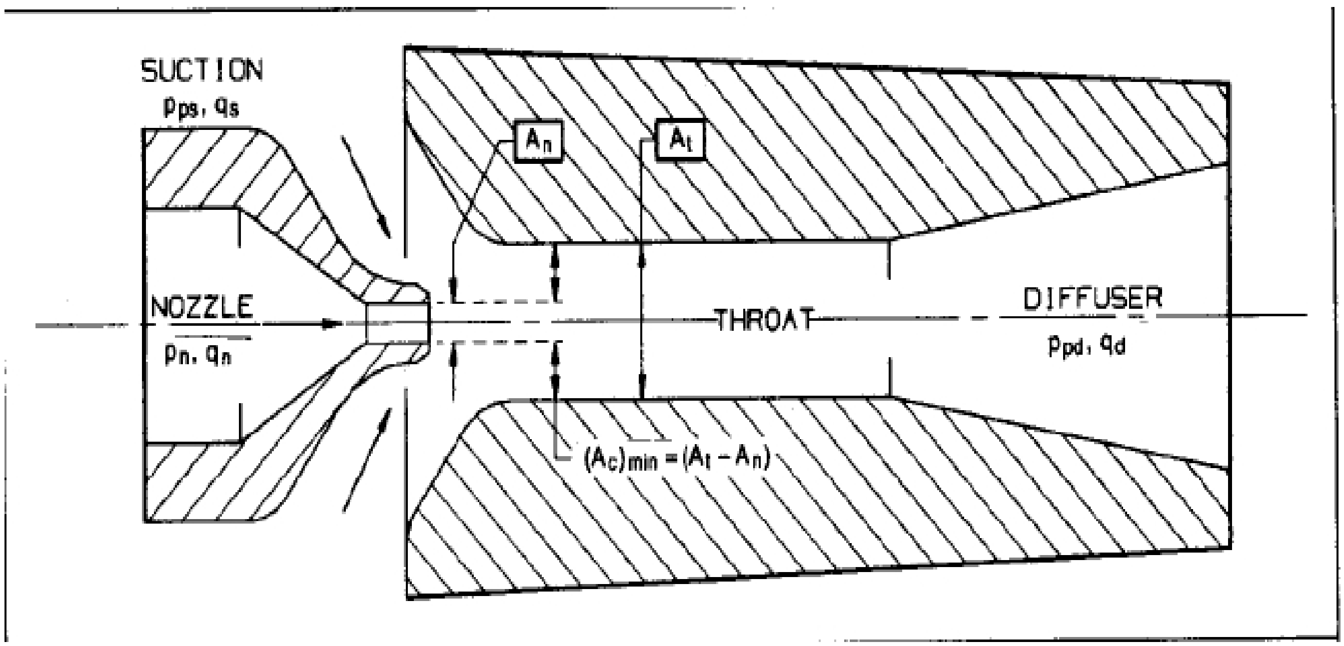
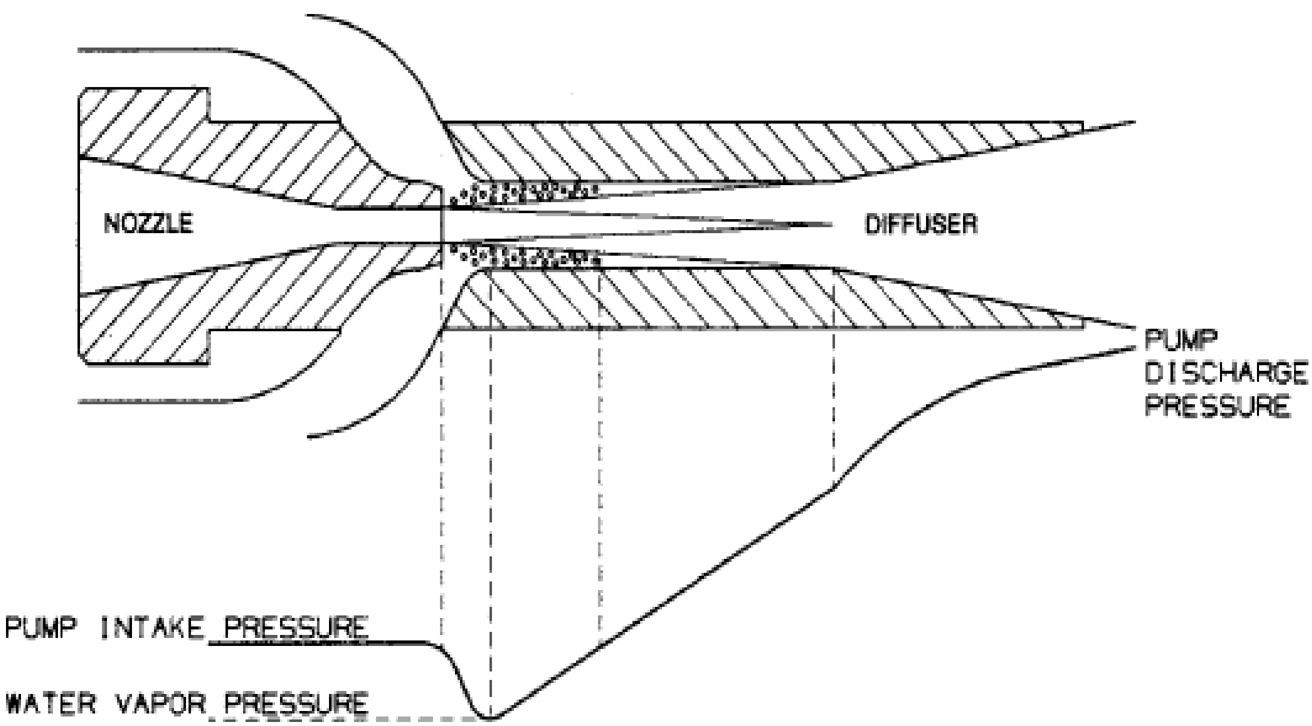


Fig. 2—Typical dimensionless performance curves.

15177 Christ FC et al 1989





15177 Christ FC et al 1989

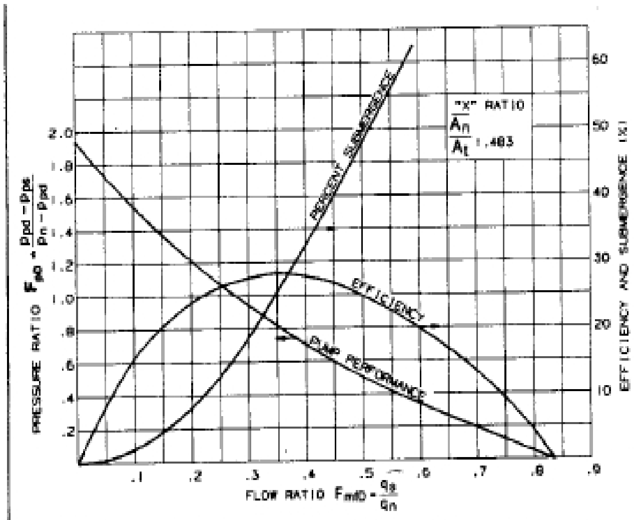


Fig. 4—Jet-pump performance and percent submergence.

$$F_{pD}=\{2F_{AD}+[(1-2F_{AD})(F_{mfD}{}^2F_{AD}{}^2)/(1-F_{AD})^2] -(1+K_{tD})F_{AD}{}^2(1+F_{mfD})^2\}/[(1+K_n)-\text{numerator}], \quad \ldots(5)$$

$$E_p=F_{pD}F_{mfD}=[(p_{pd}-p_{ps})/(p_n-p_{pd})](q_s/q_n), \quad \ldots\ldots\ldots(6)$$

and $(A_c)_{min}=(A_t-A_n)=q_s/691\sqrt{p_{ps}/g_s}$, ... (7)

where $(A_c)_{min}$ = minimum throat annulus flow area to avoid cavitation, in.^{2} [cm^2]; A_n = flow area of nozzle, in.^{2} [cm^2]; A_t = flow area of throat, in.^{2} [cm^2]; E_p = jet-pump efficiency, fraction; F_{AD} = ratio of nozzle area to throat area, dimensionless; F_{mfD} = mass flow ratio, dimensionless; F_{pD} = pressure recovery ratio, dimensionless; g_{pf} = gradient of power fluid, psi/ft [kPa/m]; g_s = gradient of produced fluid (suction), psi/ft [kPa/m]; K_n = nozzle loss coefficient, dimensionless; K_{tD} = throat-diffuser loss coefficient, dimensionless; p_n = nozzle entrance pressure, psi [kPa]; p_{pd} = pump-discharge pressure, psi [kPa]; p_{ps} = pump-intake pressure (suction), psi [kPa]; q_n = nozzle flow rate, B/D [m^3/d]; q_s = pump-intake flow rate (suction), B/D [m^3/d]; and S = submergence, fraction.

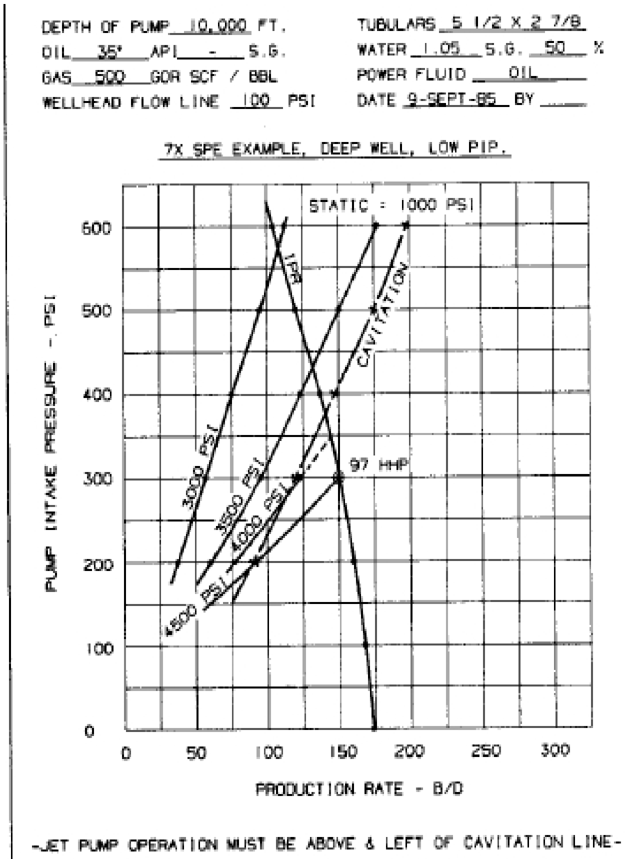


Fig. 5—7X jet-pump production unit performance.

Dividing Eq. 7 by Eq. 1 gives

$$q_s/q_n=F_{mfD}=(A_c)_{min}691\sqrt{p_{ps}/g_s}\,/832A_n\sqrt{(p_n-p_{ps})/g_{pf}}. \quad \ldots(13)$$

Because $g_s=g_{pf}$ and $(A_c)_{min}=(A_t-A_n)$,

$$F_{mfD}=0.831[(A_t/A_n)-1]\sqrt{p_{ps}/(p_n-p_{ps})}. \quad \ldots(14)$$

#15670 Grupping FW et al 1988 Peer review, rebuttal, & example

Grupping et al. in their paper "Fundamentals of Oil well Jet Pumping" (Feb. 1988 SPEPE, Pages 9-14) are to be commended for their derivation of Eq. 11 in such a short, concise manner. The previous derivation by Petrie et al.[1,2] was arrived at in a different manner, was more complex, and is valid only when densities of power fluid and production fluid are equal.

However, the conclusion that maximum efficiencies will be obtained when the ratio of throat-entrance pressure to pump intake pressure is in the range of 0.3 to 0.7 is in error and is not borne out by experience or experiment. Grupping et al. reached this conclusion by Fig. 5, which is a plot of efficiency vs. the aforementioned ratio for constant values of nozzle pressure/pump-intake pressure. But for a given problem for specified production rate and pump-intake pressure, the nozzle pressure will vary as throat-intake pressure varies because varying throat-intake pressure means varying nozzle and throat size.

Maximum efficiency will always occur when throat-intake pressure is 35 % of pump-intake pressure because cavitation or maximum flow for a given pump-intake pressure will occur when the pressure head is about 54% of the velocity head. This has been repeatedly borne out by experiment. This also assumes the vapor pressure of the fluids to be zero. Any suction area calculated for a given rate and a throat-intake pressure of less than 35% of pump intake pressure will not pass that rate yet will calculate a greater efficiency. Any suction area based on greater than this 35% will result in increasingly lesser efficiencies.

The example problem results in Table A-2 seem to refute the above statements. The reason is that Eq. 17 is used to calculate FVF, B. It is valid in only a very narrow range of GOR's and pressures, and one should use Standing[3] or Lasater.[4] For example, in the example where throat-intake pressure is 150 psi and GOR is 200, B is calculated to be 4.95! This is the reason the horsepower is 128.5 hp, compared with 61.5 and 67.6 hp for the other two cases. If a proper correlation for B is used, the 128.5 hp will calculate to be less than the other two cases, although 150 psi is not a valid case because the area will not be great enough for the specified volume.

Note that Petrie's K_n of 0.03 is for calculating nozzle flow rate, q_n, where $q_n = 857(1 - K_n)$... and is not the same K_n used for energy loss in the nozzle as used in Eq. 11, although it has been used interchangeably. Gosline and O'Brien's[5] K_{td} of 0.38 is very close to Petrie's 0.20 because they are used in entirely different equation forms. Gosline and O'Brien (actually Lorenz[2] in 1910) did not use Eq. 11, but a simpler solution using the Lorenz mixing loss equations. They arrive at the same results, but the experimental loss constants for the throat-diffuser are different. K_{td} varies not only with Reynolds number but with throat ID and surface finish, just as friction factor varies on a Moody or Fanning diagram.

#15670 Grupping FW et al 1988 Peer review, rebuttal, & example

The derivation of Eq. 11 is not new. We obtained it from work carried out by Cunningham[1] in 1955. Cunningham studied the application of jet pumps as a lubrication scavenge pump. With no entrained air, the power fluid and "production" fluid are the same. For which case Eq. 11 simplifies to the form shown by Petrie et al.[2]

Griffin states that our Fig. 5 must be in error because for a given pump-intake pressure, p_p (with corresponding production rate, q_f, the nozzle pressure, p_n, varies with the throat-entrance pressure, p_ϵ. What this says in effect is that the same value of p_n/p_p (the same curve) cannot be used when the value of p_e is changed. This is true, of course, but it does not invalidate Fig. 5. For other values of p_e, one must move to other curves with different values of p_n/p_p. Of all the possible curves, two have been presented in Fig. 5, for p_n/p_p values of 5 and 8.

The example problem with results in Table A-2 gives no indication about the p_e/p_p value at which maximum efficiency occurs. Efficiency can be calculated with Eq. 12b, using the density of the well fluids at throat entrance, ρ_{fg} it is not necessarily highest when horsepower requirements are lowest. We calculated the efficiencies for the three p_e/p_p values shown in Table A-2; all three were in the range of 29 to 32%, indicating that, also at low p_e, the pump operates efficiently (see also Fig. 6). The problem is, of course, that it is then (efficiently) pumping mostly gas instead of liquid. The reason for the higher horsepower requirement is therefore not its low mechanical efficiency but its low volumetric efficiency. Similar results would be obtained if another method were used to determine the FVF, B.

We nevertheless agree that Eq. 17 is valid only for a fairly narrow range of GOR's and pressures. We used it because its simplicity allows the fundamental aspects of jet pumping with free gas to be explained clearly and concisely.

Griffin repeatedly refers to experience and experiment to refute the conclusions of our analysis but does not provide experimental or field data in support. To the best of our knowledge, no experimental data exist on jet pumping gassy fluids under conditions of flow rate and pressure that occur at the bottom of oil wells. If accurate field data are available, it would be most helpful if these could be published.

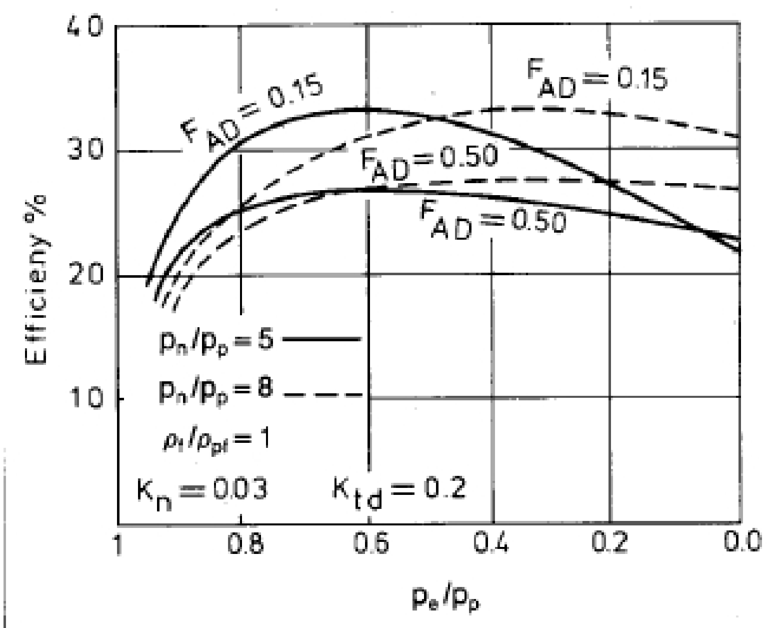


Fig. 5—Influence of p_e/p_p on pump efficiency for various combinations of p_n/p_n and F_{AD}.

#15670 Grupping FW et al 1988 Peer review, rebuttal, & gas example

In single-string jet pump installations, gas cannot be vented but must be produced through the pump. There are two sources of free gas at throat-entrance pressure, p_e: free gas at pump intake pressure, p_p, and dissolved gas that is liberated when the pressure drops from p_p to p_e. Cunningham[7] has investigated the effect of free gas on jet pump performance. His analysis ignores liberated gas and is valid only for small density differences in the gassy production at p_p and p_e. It was found that under these conditions the jet pump will pump free gas as if it were liquid.

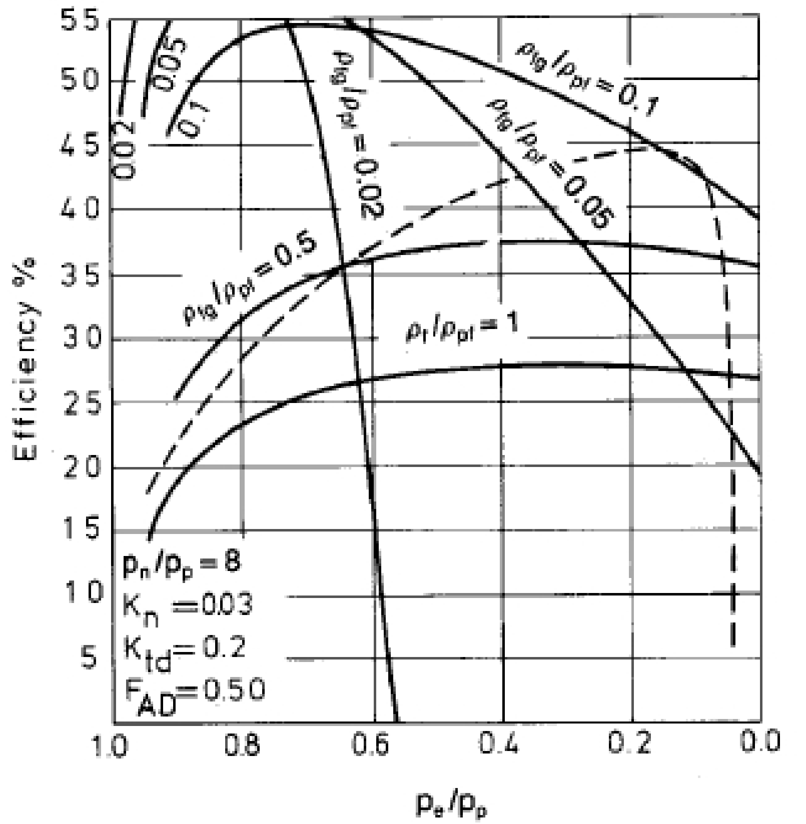


Fig. 6—Effect of gas on efficiency at various values of p_e/p_p.

To deal with free gas under oilfield conditions, its volume must be determined at the throat-entrance pressure, p_e. Because this is difficult and time-consuming, a simplified procedure has been proposed by Petrie et al.[6] They calculate an FVF, B, by $B = 1+2.8(R/P_e)^{1.2}$ (17)

This factor is stated to be reasonably accurate for medium-gravity crude with low to medium GOR. With this factor, the oil fraction of the produced fluids in Eq. 15 must be adjusted to calculate a new (larger) throat suction area:

$$A_e = \{(Bq_f)(1-F_{WM}) + (q_f F_{WM})\}/\{857\sqrt{((p_p-p_e)/\rho_{fg})} \quad (18)$$

The gradient of the production must also be adjusted to account for the decreased density. For the low GOR's in oil well jet pumping, the weight of the gas may be neglected:

$$(\rho_{fg}) = (\rho_f)/(B(1-F_{WM})+F_{wm}) \quad (19)$$

In that case, the equation for calculating the dimensionless mass flow ratio remains unchanged:

$$F_{wD} = q_f\rho_f / q_n\rho_{pf}$$

The equation for calculating the density ratio, ρ_{pf}/ρ_m, changes to

$$\rho_{pf}/\rho_m = ((F_{wD})(\rho_{pf}/\rho_{fg})+1)/(F_{wD}+1) \quad 21$$

#15670 Grupping FW et al 1988 Peer review, rebuttal, & gas example

This equation is based on the simplification that the free-gas volume in the throat is not dissolved or compressed when the pressure increases from p_e to p_t.

Pump Performance with Free Gas

In principle, for a jet pump in a free-gas environment, a similar relationship can be established between p_e, p_p, and p_n as for gas free production.

Fig. 6 shows the effect of gas on the pump efficiency for various values of p_e/p_p. The curve for $(\rho_{pf}/\rho_{fg})=1$, $F_{AD}=0.50$, and $p_n/p_p=8$ has been reproduced directly from Fig. 5. Additional standard curves have been constructed for (ρ_{fg}/ρ_{pf}) values of 0.5, 0.1, 0.05, and 0.02. The dotted curve shows a generalized performance curve of a jet pump handling gas.

At high values of p_e/p_p, most of the gas is dissolved, so the performance curve approaches the curve $(\rho_f/\rho_{pf}) = 1$. At lower pressure ratios, the performance curve meets standard curves of successively lower (ρ_{fg}/ρ_{pf}) values. Its efficiency gradually increases as p_e/p_p values become lower; at very low values, it dips steeply to zero efficiency. Optimum operating conditions i.e., those that require the least amount of horsepower for a given liquid production rate; occur in the range of p_e/p_p = 0.3 to 0.6. At higher values, the efficiency drops off rapidly at lower values; increasing amounts of free gas are pumped, resulting in low volumetric efficiencies.

TABLE A-1—WELL DATA AND JET-PUMP-LOSS COEFFICIENTS FOR EXAMPLE JET PUMP CALCULATION

Depth of reservoir, ft	5,000
Depth of jet pump, ft	5,000
Reservoir pressure, psi	1,250
Productivity index, bbl/(psi-D)	8
Water cut	0.50
GOR, ft^3/bbl	200
Wellhead pressure, psi	150
Oil gradient, psi/ft	0.380
Water gradient, psi/ft	0.446
Power-fluid gradient, psi/ft	0.380
Nozzle-loss coefficient	0.03
Throat/diffuser loss coefficient	0.20

TABLE A-2—RESULTS OF EXAMPLE JET PUMP CALCULATION

p_e, psi	1,050	875	150
p_e/p_p	0.913	0.761	0.130
A_e, in.^{2}	0.0655	0.0402	0.0327
Surface injection pressure, psi	1,314	1,770	3,495
Horsepower	67.6	61.5	128.5

#15670 Grupping FW et al 1988 Peer review, rebuttal, & gas example

Appendix—Example Jet Pump Calculation

See Table A-1 for well data and jet-pump-loss coefficients. For simplicity, friction looses in injection and production conduits have been neglected, as has been the effect of free gas on the gradient in the production conduit.

Calculations.

1. q_f=800 B/D [127 m^3/d] (selected).
2. p_p=1,250−(800/8)=1,150 psi [7929 kPa].
3. p_e=875 psi [6033 kPa] (selected).
4. B=1.476.
   ρ_f=[(0.5)(0.38)]+[(0.5)(0.446)]=0.413 psi/ft [9.3 kPa/m].
   ρ_{fg}=0.334 psi/ft [7.6 kPa/m].
   A_e=0.0402 in.^{2} [0.26 cm^2].
5. F_{AD}=0.40 (chosen).
6. p_n=3,900 psi [26 890 kPa] (chosen).
7. q_n=1,925 B/D [306 m^3/d].

8. ρ_d={[(800)(0.413)]+[(1,925)(0.38)]}/(800+1,925)=0.390 psi/ft [8.8 kPa/m].
   p_d=150+[(0.390)(5,000)]=2,100 psi [14 480 kPa/m].
9. F_{wD}=0.4517.
10. ρ_{wf}/ρ_{fg}=1.138.
    ρ_{wf}/ρ_M=1.043.
11. F_{pD}=0.6313.
12. p_n=3,605 psi [24 856 kPa] (calculated).
13. Because the calculated value of p_n (3,605 psi [24 856 kPa]) is unequal to the chosen value of p_n (3,900 psi [26 890 kPa]), another value for p_n must be chosen. Repeated iteration results in p_n=3,670 psi [25 305 kPa], q_n=1,839 B/D [292 m^3/d], f_{wD}=0.4728, and F_{pD}=0.6039.
14. Surface injection pressure=3,670−(0.38)(5,000)=1,770 psi [12 200 kPa]. Required horsepower=[(1,839)(1,770)]/52,910=61.5 hp [45.9 kW].

15. Repeating the procedure for F_{AD}=0.3 results in a surface injection pressure of 2,335 psi [16 100 kPa] and a horsepower requirement of 57.8 hp [43.1 kW].
16. Repeating the procedure (with F_{AD}=0.4) for p_e values of 1,050 and 150 psi [7240 and 1035 kPa] gives the results shown in Table A-2.

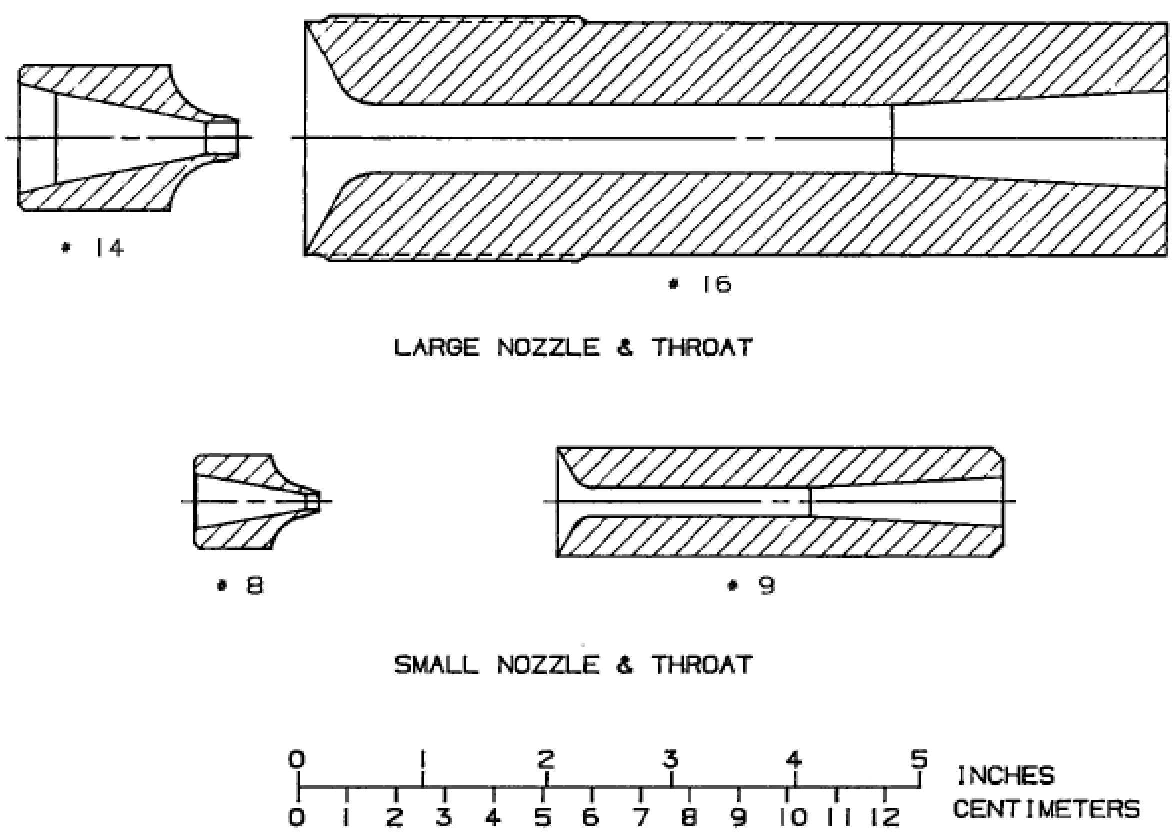
Remarks.

1. A_e values are theoretical values; available sizes from pump manufacturers will be slightly different.
2. The possible occurrence of choked flow or cavitation at p_e=150 psi [1035 kPa] has been disregarded.

#114332 Telkov VP 2007

There are technologies of WGM producing and pumping by means of jet pumps. They differ in the placing of jet pump: in the first case jet pump is located on the surface (fig.4), in the second case it is placed in a well (fig. 5). In both situations jet pump make fine dispersed WGM. Nevertheless these technologies have not gained ground. The former ejector cannot produce required pressure to inject WGM in a layer; the latter one does not have an opportunity to control the process of WGM injecting as it is placed in the well. There is an effective jet and electrical centrifugal pumping technology of fine dispersed mixtures production and injection (fig. 6). This technology uses simple and available equipment such as jet pumps, electrical centrifugal pumps (ECP). The cost of this equipment is comparatively low. Locating all system units on the surface allows regulating the process of injection WGM. This technology solves the problems which could not be tackled using above-listed technologies. According to this technology injected water (inducing stream) and gas (induced stream) are mixed in an ejector. Gas is taken from an oil-gas separator, flare, or gas well. It is important that this system does not require high gas pressure. An ejector produces fine dispersed WGM the pressure of which can be increased by means of booster pump till the necessary level is reached. The obtained WGM can be injected in a single pumping well and in all the wells in the field. Using surfactants not only stabilizes WGM but also reduces harmful effect on booster ECP operating caused by gas. Moreover using surfactants in a mixture increases oil reservoir recovery.

#11748 Christ FC et al 1983



#11748

Fig. 1—NPS jet pump nozzle and throat patterns.

#11748 Christ FC et al 1983

Development of the high volume Jet pump has occurred over a lengthy time period. The basic concept of an oil well Jet pump was first proposed in the early 1930's. Theoretical basis for current performance calculations were developed at this time. Little was done with the idea until the 1950's when a development program confirmed that a jet pump could indeed be used to produce a significant number of existing oil wells. It was not until the 1970's that the Jet pump was commercially used to produce oil wells, there are now over 1000 in use. The jet expanded the producing capabilities of hydraulic oil well pumping systems, offering a number of advantageous performance characteristics. The basic jet pump is actually a working unit averaging just over two feet long. However, the overall subsurface production system consists of the pump, bottomhole assembly, standing valve, and swab nose. Briefly, the bottom-hole assembly (BHA) is the receptacle which holds the pump and provides the barrier between high and low pressure fluids. A standing valve rests in the BHA, beneath the pump, to act as a check valve during pumping out operations. The swab nose is the locomotive attached to the top of the pump to provide a larger surface area while pumping out. The heart of the unit, though, is an accurately contoured nozzle and throat (Figure 1), whose diameters determine production performance. The pump is a dynamic pump, having no moving parts, which allows tolerance to produced gas and abrasives. These characteristics, combined with creativity, offer a production pump capable of diverse and flexible applications.

Consistent with the Jet pump’s flexible characteristics, units have been designed to operate primarily in four different tubing sizes: 1.9, 2.375, 2.875, and 3.50 inches O.D. Additionally, adaption to the existing bottom-hole assembly is usually possible by altering pitch lengths. Generally, this can all be accomplished while maintaining its "free" pump capabilities, light weight, and large selection of nozzle/throat combinations for changing well conditions and/or production rates.

Production volumes of the jet pump are limited primarily by two factors; cavitation and friction losses. Cavitation can usually be avoided by proper sizing of nozzle/throat ratios. Friction losses are inherent in any system but can be minimized in the jet pump by prudent design of flow passages and fluid crossovers. At least one fluid crossover is needed in each pump. Two equipment problems occurred which required modification. During the equipment running phase of the first well, WS #2, the standing valve refused to seat in the bottom hole assembly and was, in fact, catching in the BHA. Examination of well program charts showed a highly deviated hole. A 10 in. guide was added to the standing valve bottom allowing it to properly fall and seat. An additional problem appeared during retrieval when the pump refused to enter the lubricator. It was determined the 1/2 in. D. flow area of the fluid outlet tube at the lubricator top was insufficient to exhaust sufficient water and thus carry the pump inside. The tube was enlarged to one inch, resolving the problem.

#16923 Corteville JC et al 1987

Jet pumps are known as being able to pump any sort of fluid. They are used for downhole pumping because of their simplicity, ruggedness, flexibility of use and ease of maintenance. Several thousand wells are equipped with jet pumps, mainly in the United States, but their implementation remains limited to cases of difficult production (deep well hot or corrosive crudes, production of sand or formation of deposits, etc.). There are two major related reasons why their use is not spreading faster:

- Their energy efficiency is reputed to be low. For oil jet pumps, the typical efficiencies announced range from 26 to 33 %
- Sizing methods are often considered as inaccurate, incomplete and somewhat mysterious. Models developed by experimenters in a restricted parametric range of geometric shapes, flow rates, pressures and fluid natures, cannot be generalized to all jet pumps and all operating conditions. Formulations may diverge, depending on the simplifying hypotheses retained, thus giving discordant results. Lastly, there is no tried-and-true design method in two-phase pumping conditions.[3]

CHARACTERIZING JETPUMPS

Jet pumps operate on the principle of the venturi tube. A high-pressure driving fluid is ejected at high speed through a nozzle, thus creating a depression enabling the fluid to be sucked up and entrained in a cylindrical mixing tube (throat) where the velocities are homogenized. A conical diffuser then converts the kinetic energy of the mixture into pressure (Fig. 1).

NOMENCLATURE A = area (mm) D = Diameter (mm); F1, F2, F3, F4 = Pressure functions GOR = volumetric gas oil ratio k = Density number (ρ_S/ρ_P) K = Friction loss coefficient L = Length (mm)
M = Volumetric flow ratio (Q_S/Q_P)
N = Pressure ratio $(P_7 - P_2)/(P_1 - P_7)$ P = Pressure (bar)
Q = Volumetric flow rate (m^3/h)
R = Nozzle/throat entrance area ratio (A_4/A_5)
U = Velocity (m/s) WOR = Water Oil Ratio
λ = Throat entrance/throat exit area ratio A_6/A_5
ŋ =Energy efficiency, N*M; ρ = Fluid density (Kg/m^3).

Subscripts:
G = gas L = liquid P = primary flow (driving fluid)
S = secondary flow (sucked fluid)
1 to 7 = sections as indicated Figure 1.

#16923 Corteville JC et al 1987

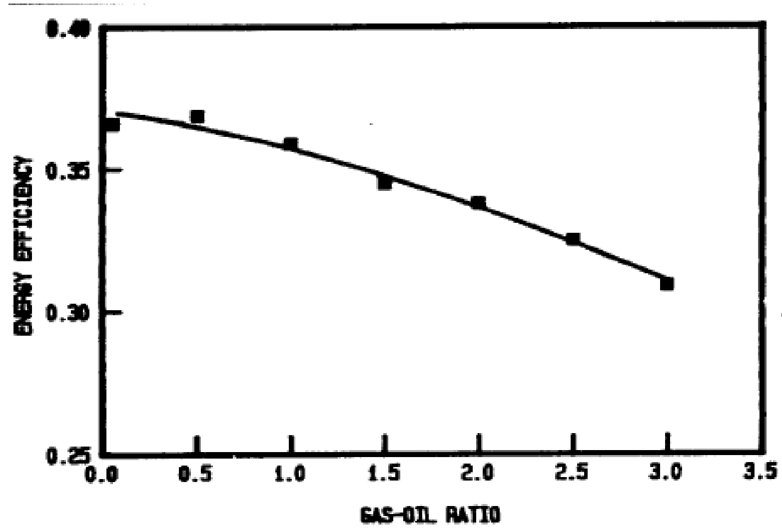


Fig. 8—Measured energy efficiency vs. suction GOR (prototype jet pump).

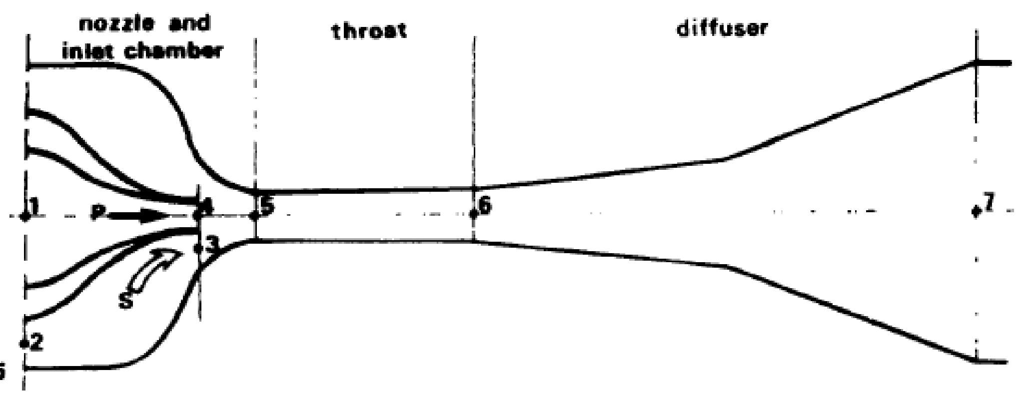


Fig. 1—Principle of jet pump and nomenclature.

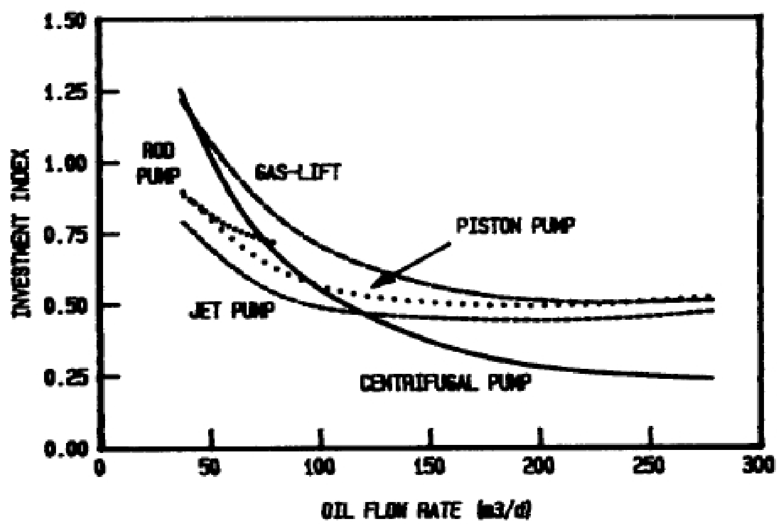


Fig. 10—Investment index of activation systems vs. production rate.

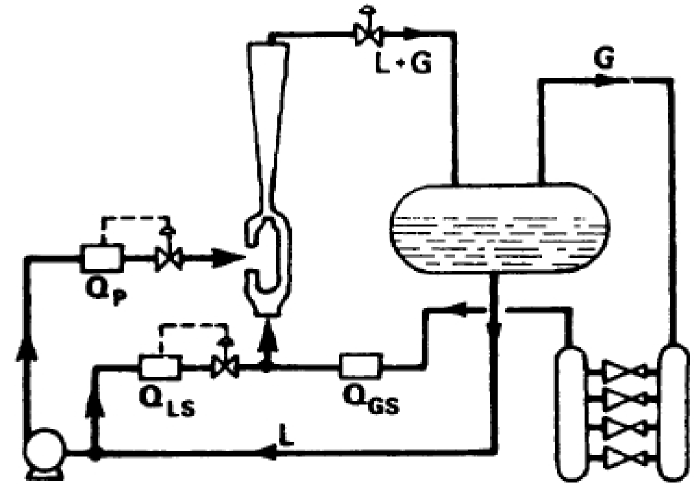


Fig. 2—Jet pump two-phase flow test bench.

#16923 Corteville JC et al 1987

Fluids tested: a light gas oil (density 835 kg/m3, viscosity: 0.005 Pa.s at 15°C) and nitrogen for two-phase tests. The suction, discharge and driving pressures are determined by the same differential pressure sensor to improve precision. Other differential pressure sensors are used to plot pressure profiles recorded at the Inlet of the throat, the inlet of the diffuser and at three Intermediate points on the diffuser.

Flow rates are measured by two different methods: diaphragms and vortex-effect flow meter. Data were recorded by a computer and processed statistically, with 40 data samples. The conditions examined are typical of the conditions encountered in medium-production wells: suction flow rate up to 16 m3/h. power flow rate 8 to 14 m3/h. suction pressure 10 to 50 bar. Driving pressure ranged from 30 to 80 bar.

Industrial Pump Tests: The industrial 2 1/2" jet pump examined is capable of operating at medium flow rates (10 m3/h). Internal fluid circulation was adapted (elimination of two fluid crossings), to be able to test the specific part of the jet pump under the same conditions as the prototype pump. The principal characteristics were: Throat diameter D = 11.03 mm. Nozzle diameter = 6.76 mm (0.38D) length of cylindrical throat = 125 mm (11.3D), constant nozzle throat spacing = 8.4 mm, angle of diffuser opening = 6°.

For the nozzle-throat set used, the manufacturer Indicates a maximum efficiency of 36% for a flow rate ratio of M = 0.75. Liquid single-phase suction: performance curves N = f(M) and ŋ = f(H) were plotted for different driving flow rates (8 to 14 m3/h) and different suction pressures (30, 40 and 50 bar). The efficiencies are not very sensitive to these two parameters, and the best efficiency obtained was 37.5% for M = 0.75 (F1g. 3 & 4). Hence the values indicated by the manufacturer were confirmed, considering that two fluid crossings were not taken into account in these tests.

. Two-phase suction: a sensitivity study of the pump's efficiency to suction GOR was made (Table 1). For low GORs (GOR < 1), the pump's behavior is not affected by the presence of gas, and the efficiency holds at around 38%. For higher GORs, the efficiency decreases. Single-phase gas suction: the main parameter investigated was the suction pressure. Efficiency improves when suction pressure increases (Fig. 5).Indeed, a higher suction pressure reduces the contrast between densities, thus enhancing entrapment of the gas by the liquid.

Prototype Jet Pump Tests

The first phase of testing consisted in successively studying the different constituent parts of the pump (nozzle, Inlet chamber, threat and diffuser) to obtain an optimized prototype pump. The aim of the second phase was to create a test database in a wide parametric range to act as basis for modeling.

#16923 Corteville JC et al 1987

The first pump was made on basic rules-of-the-art taken from the literature (cylindrical throat 5D to 8D long, tapered diffuser nozzle with opening angle of 7°) and by computing the ratio of cross-section R as a function of desired characteristics.

Shapes of nozzle and Inlet chamber: 5 nozzle shapes were tested with two cross-section ratios and three curve radii (Table 2). Nozzle 3, with a cross-section ratio R of 0.4 and Intermediate curve radii (inside radius of 50 mm and outside radius of 35 mm) gave best performance.

Likewise, two inlet chamber opening curve rad11 were tested (10 and 16mm). The efficiencies obtained by these two shapes were identical.

Nozzle-throat spacing: this sensitive parameter (Fig. 6), was systematically studied for all geometries tested. A range of optimal spacing's corresponding to the geometry, depending on mixing zone length.

Throat Design: a slightly tapering cone (opening angle of 1.3°) was adopted. Indeed, a cylindrical throat forms a zone in which the mixture of sucked and driving fluids is already accompanied by pressure recovery. This cylindrical shape, which had previously been extensively tested in prior laboratory experiments, is thus not technically needed for separating the mixing and pressure-recovery functions. However, a diverging shape, enabling pressure losses to be reduced in the nixing zone, can improve overall efficiency.

Different throat lengths were tested (3D, 5D, 8.5D, 12D) with single and two-phase fluids. A correlation between throat length and optimal nozzle-throat spacing was revealed. Long throats (8.5D and 12D are more efficient (table 3) and optical spacing is all the less as the throat Is long, while the throat intake chamber can ensure same fluid mixing and a reduction in the velocity contrast.

Design of the diffuser: the important parameter for the diffuser is the angle of opening. Two diffusers with varying angles (l0°/5°/7^0 and 3.5°/7^0) were tested comparatively to a classical diffuser with an angle of 7° (Table 4). The efficiency of the (3.5°/7^0) progressive-angle diffuser is slightly greater than that of the 7^0 diffuser. The l0°/5°/7^0 diffuser shoved poor performances during its first tests and was eliminated. The optimal profile for the throat-diffuser unit thus approaches a diffuser with a continuous Curve having a regular progression of the angle of the cone.

An approach to optimal geometries was to plot pressure profiles (Fig. 7) to work out a sizing model for the pump. Measurements were made in single and two-phase flows, by varying three parameters: driving throughput (8 to 13 m3/h), suction pressure (15 to 50 bar) and GOR (Fig. 8). Performances diminish slightly with an increase in GOR, but for single-phase gas suction, efficiency is greater than with single-phase liquid suction. The results obtained reveal the following main points: The geometric shapes, especially those of the nozzle and throat, have a clear-cut influence on efficiency. Correct efficiency (40%) can be obtained with single-phase liquid flow.

Efficiency can be kept at a high level for two-phase suction and single-phase gas suction and may even rise above 50[1] in the latter case.

#16923 Corteville JC et al 1987

In November 1983, Petrie, Wilson and Smart, representing the three leading U.S. companies manufacturing petroleum jet pumps, published a unique and simple formulation. With the notations used here, it can be written in the form:

$$1+1/N = (1+K_n)/[2R+(1-2R)\{(kMR)/(1-R)\}^2-(1+k_{TD})(1+kM)R)^2]$$

The formulation has two friction loss coefficients whose values may be K_N = 0.03 for the nozzle and k_{TD} = 0.2 for the throat-diffuser. The geometric shapes intervene by only one parameter R, which is the cross-section ratio of the nozzle and throat. Likewise, the fluid properties intervene only via the single parameter, k, which is the density ratio of the secondary and primary fluids. This formula thus cannot determine the optimum nozzle-throat spacing in relation to the throat inlet, which is an important parameter for which manufacturers have determined the optimum as a function of operating conditions. Furthermore, it is insensitive to throughputs and fluid viscosities, which exert an influence in the form of Reynolds numbers.

This formulation predicts an optimal efficiency of 31% at M = 0.54 for the industrial jet pump (measured value 37.5% at M = 0.75), and of 30.5% at M = 0.50 for the prototype jet pump (measured value 40% at M = 0.75).

The recompression of two-phase mixtures has only exceptionally been considered to date. As an initial approximation, Petrie et al. propose to use the same formulation with a throughput M determined by considering that intake flow rate Qs is the sum of the volume flow rates of both the gas and liquid phases to be recompressed, determined under suction conditions.

Therefore, as it now stands, a proposed four-equation model does not seem to have any physical meaning and thus brings no advantages over formulation by Petrie et al. However, these conclusions remain provisional until a more thorough analysis can be made of the experimental data acquired and until a more in-depth theoretical analysis can be made. Modeling work is continuing with a view to simplifying hypotheses.

Table 3 to Right

EFFECT OF THROAT GEOMETRY ON MAXIMAL EFFICIENCY OF PROTOTYPE JET PUMP (GOR = 0, Diffuser : 7°)

THROAT LENGTH (mm)	NOZZLE - THROAT SPACING (mm)	EFFICIENCY η
30	13	0.330
	25	0.360
	33	0.320
53	13	0.360
	18	0.376
	25	0.340
90	8	0.389
	13	0.409
	18	0.373
	43	0.249
130	8	0.381
	13	0.393
	18	0.380

ENERGY EFFICIENCY OF THE INDUSTRIAL JET PUMP VERSUS SUCTION GOR — TABLE 1 (P_s = 30 bar, Q_p = 13 m^3/h)

(m^3/m^3) GOR	0	0.5	1	2	3	∞
Efficiency η	0.380	0.380	0.380	0.359	0.312	0.384

#otc6754 Bryant R et al 1991

This paper reviewed the system selection process and implementation of an artificial lift system (Jet pump) for drill-stem testing a heavy oil exploration prospect in a mobile offshore drilling unit (MODU) environment. The application of artificial lift technology was used on a floating semi-submersible in exploration drilling. It is the first of its kind in Canadian waters. It is unique by virtue of the system's simple and self-contained configuration. The system was comprised of a reverse circulated jet pump set into a conventional 3.5 inch drill pipe test string. Power fluid for pump operation was routed down the annulus via the marine riser kill line using the drilling unit cement pumps. Reservoir fluids were entrained in the power fluid stream at the jet pump and produced up the test string. Surface handling and storage requirements of fluids were minimized with a simplified separator configuration and by re-circulating power fluid for re-use. Actual field operation of this system proved it to be a safe and effective means of employing artificial lift to an offshore exploration testing program.

INTRODUCTION

With increasing frequency, heavy oil reservoirs In the offshore environment are being recognized for their massive development potential.[1] Technology Improvements have made evaluation of these reservoir types at an exploration stage more common, although testing programs often have either extensive equipment requirements or are of limited scope.'-[5]

CONCLUSIONS

As demonstrated by this test, jet pumping is a viable artificial lift system for testing low flow-capacity reservoirs in a MODU environment Running the jet pump on wireline is the preferred method of tool deployment. Jet pumps typically have very small nozzle diameters that are prone to plugging by wellbore debris. The wireline configuration allows the pump to be recovered and redressed without pulling the entire test string.

Because of the relatively high power fluid ratios required, sampling equipment for jet pump testing should be designed for accurately determining oil cuts at very low values i.e., <10%). Standard BS&W determination centrifuge tubes are designed for relatively low water-cuts (i.e., <25%) and are difficult to use with high water-cuts.

REFERENCES

Jennings, E.E. et al 1985. 'Offshore California Heavy Crude: Potential and Production*. In The Third International Conference on Heavy Crude and Tar Sands. Long Beach. California. July 22-31. 1985. (Unitar/UNDP Information Centre for Heavy Crude and Tar Sands: New York, 1985) pp. 1450-1469.

Crossley, E.G. 1986. 'Experience with Electric Submersible Pumps for Testing Heavy Oil Reservoirs From Floating Drilling Vessels'. Paper in OTC 5318 In The Eighteenth Annual Offshore Technology Conference. Houston. Texas. May 5-8. 1986.

Home, J.S. 'Engineering Evaluation for Development of a Heavy Oil Reservoir in an Offshore Environment*. In the Sixth Annual SPE Southeast Asia Offshore Conference. Singapore, Jan. 28- 31, 1986.

Buramjee, R.J. 'Heavy Crude Oil Categorized', Oil and Gas Journal (July 4. 1983) pp. 78-82.

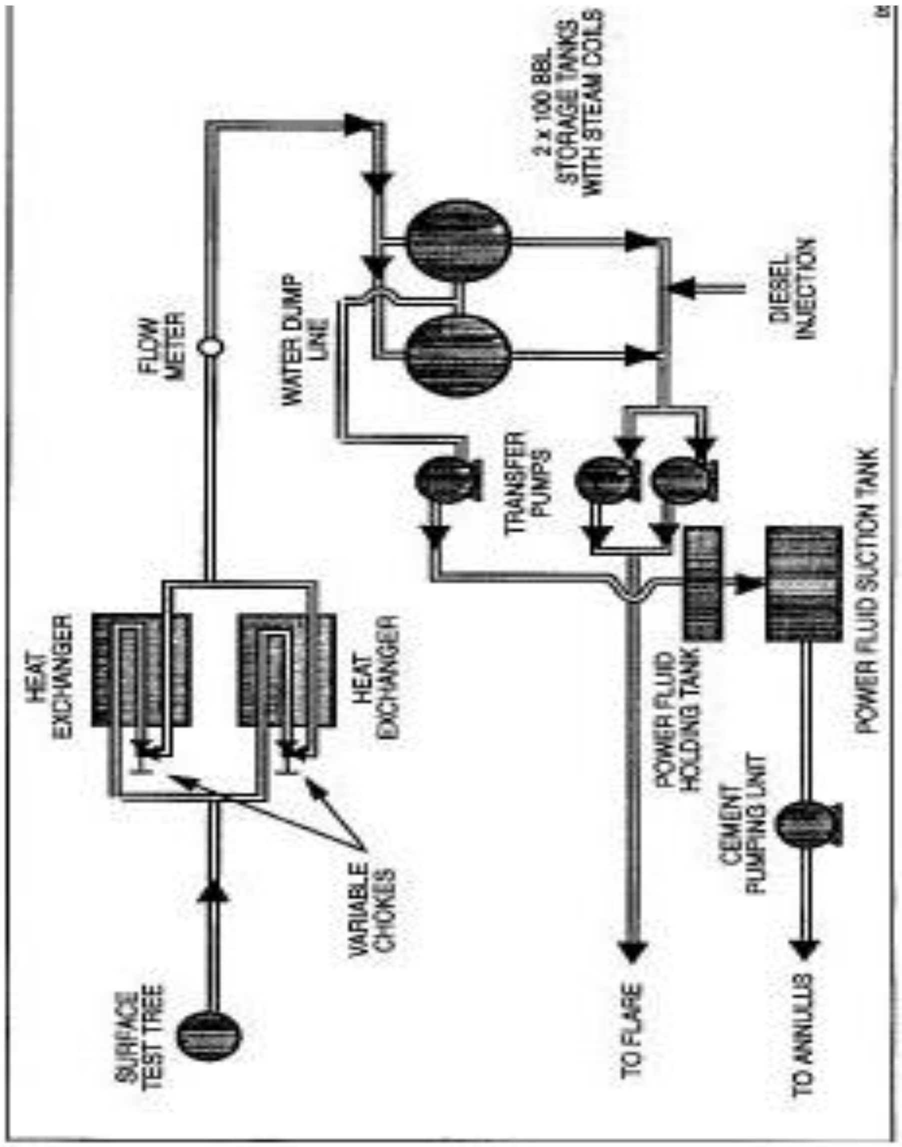
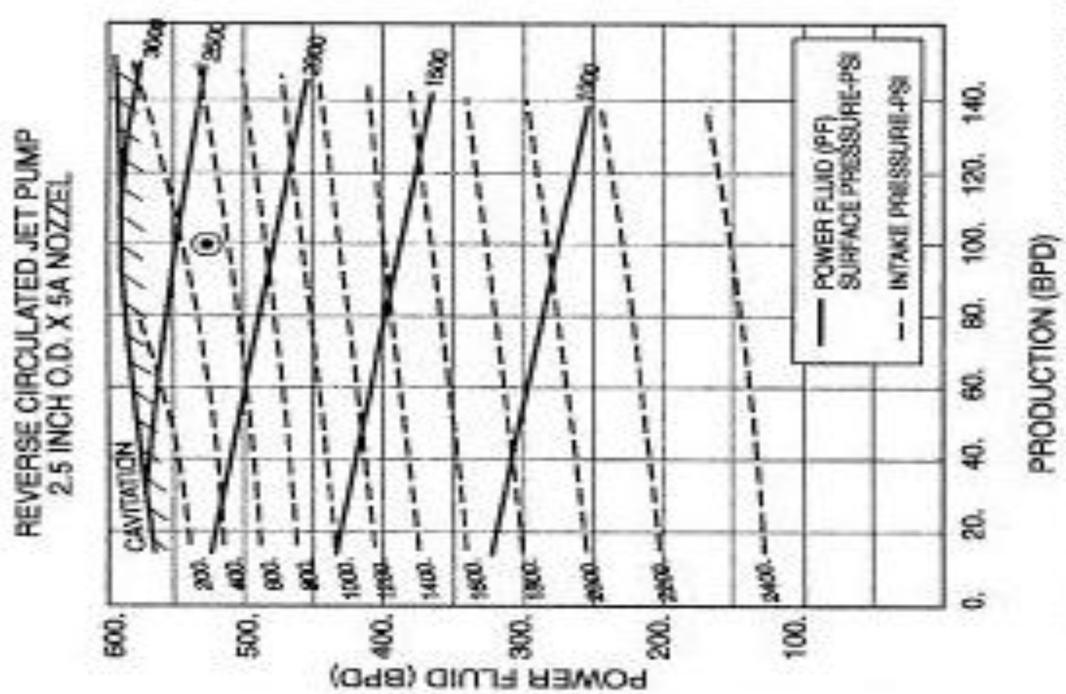


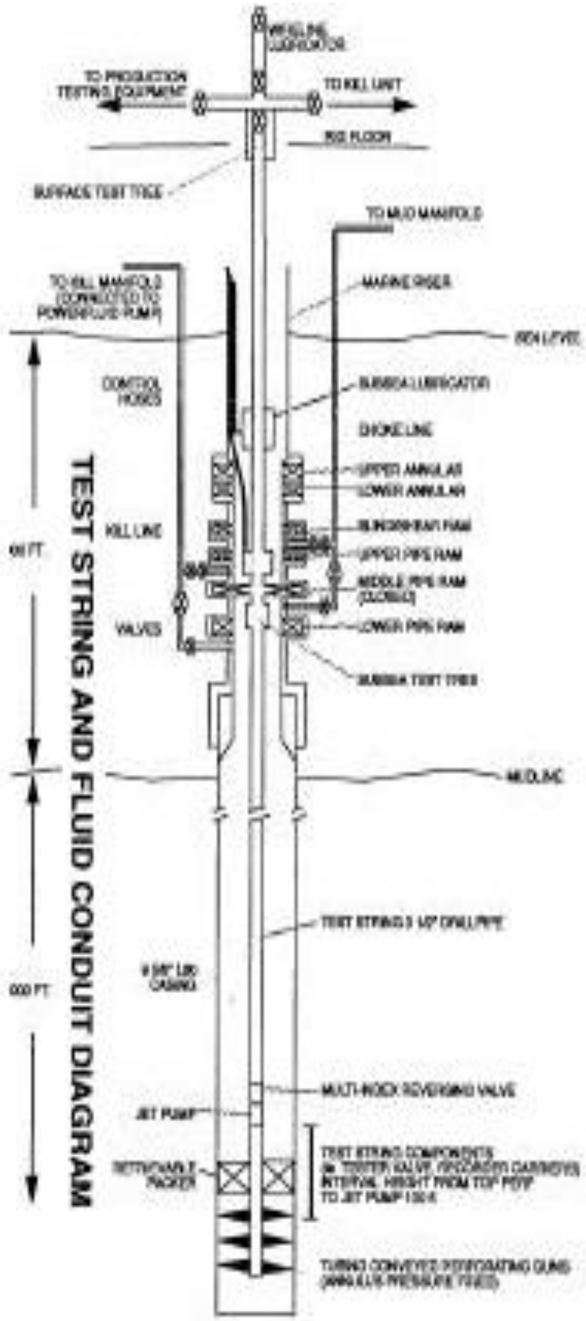
Figure 4 SURFACE TESTING FACILITY



JET PUMP PERFORMANCE CHART

EXAMPLE
- 500 PSI INTAKE PRESSURE
- 100 BPD PRODUCTION
- 2350 PSI PF SURFACE PRESSURE
- 530 BPD PF

#otc6754 Bryant R et al 1991



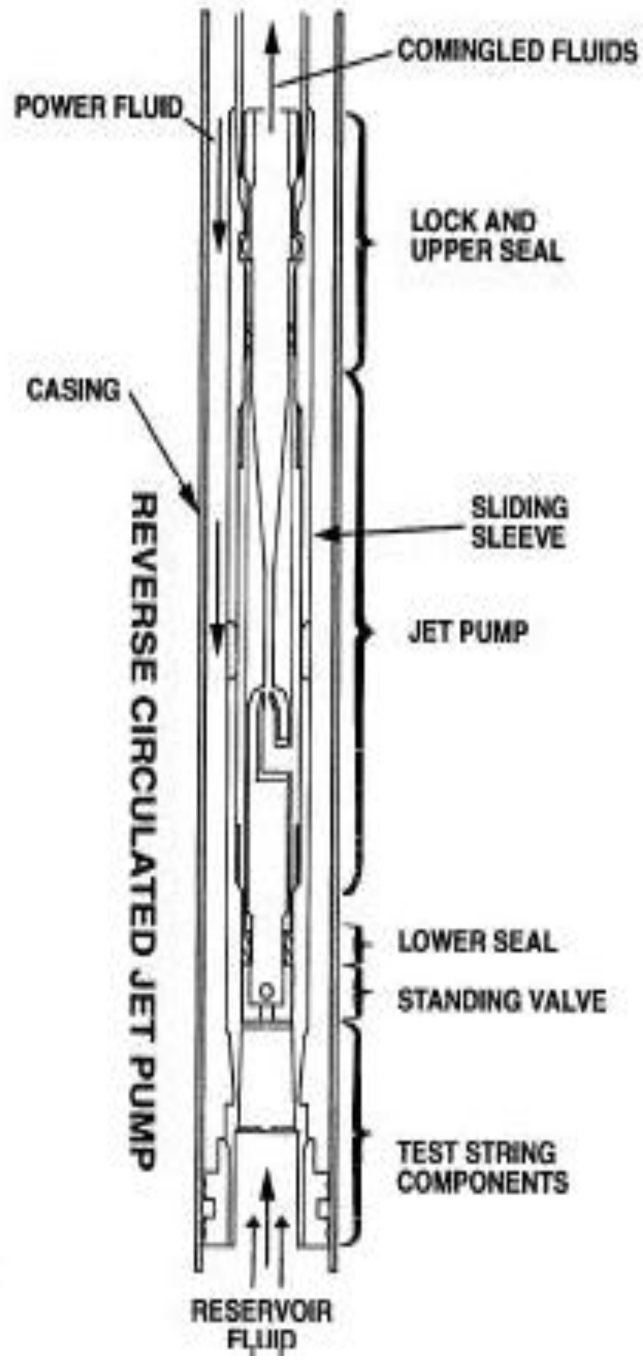


Diagram of Reverse Circulated Pump

#18190 Jiao B. et al 1990

The objective is to estimate production efficiency for a selected nozzle/throat combination. A variety of geometries are evaluated and the optimum design is selected. For field applicability, we use the air/water ratio at standard conditions. The steps involved are as follows.

1. Compute the pump-intake fluid gradient.
2. Compute the minimum throat annulus to avoid cavitation.
3. Select the nozzle and throat sizes from manufacturers' information and calculate F_A.
4. Select a surface power-fluid pressure and estimate the resulting nozzle pressure and flow rate. First, compute the Reynolds number in the power tubing, determine whether the flow is laminar or turbulent, and calculate the frictional loss.
5. Characterize the returning fluid, which consists of the suction and the power fluids. The gradient, water cut, and gas Liquid ratio are needed.
6. Determine the frictional loss of the return flow through the casing/tubing annulus.
7. Estimate a reasonable pump-discharge pressure.
8. Set f_n =0.04, and compute f_{td} with Eqs. 11 through 15.
9. Compute F_M with Eq. 7.
10. Compute F_P with Eq. 4.
11. Recompute the nozzle and intake pressures from F_P and p_d.
12 Recompute the surface pump operating pressure and maximum non-Cavitation power-fluid flow rate.
13. Calculate efficiency, E, with $F_M F_P$ in Eq. 10.
14. Determine the required horsepower of the surface pump from its operating pressure and power-fluid flow rate.
15. Compare the efficiency and horsepower requirement on other nozzle/throat combinations.

As an example, let us select a pump at 6,000-ft depth with 5 in. casing and 2⅜-in. tubing, a bottomhole pressure of 1,000 psi, a temperature of 145°F, a GOR of 1,000 scf/STB. a water gradient of 0.455 psi/ft, and a water cut of 0.25. The suction oil and power fluid are both 35"API oil of 5-cp absolute viscosity. Set the wellhead backpressure of 100 psi and the desired production rate of 600 B/D. Choosing an NPS No. 9 nozzle and No. 10 throat (F_A =0.0167/0.0562 =0.2972), we find=0.4523, F_M =0.5018, Fp =0.4514, and £=23%. These values for pressure recovery and efficiency appear reasonable.

$$B \equiv (1-2F_A)F_A^2/(1-F_A)^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

Then,

$$F_p = \frac{2F_A + BF_m^2 - (1+f_{td})F_A^2(1+F_m)^2}{(1+f_n) - 2F_A - BF_m^2 + (1+f_{td})F_A^2(1+F_m)^2} \quad \ldots\ldots\ldots (4)$$

Petrie *et al.* give values of f_n=0.03 and f_{td}=0.2 for a pump operating on pure liquid. Note that the model is relatively insensi-

#18190 Jiao B. et al 1990

$$q_{iaw} = 0.2178 q_{ia}. \quad (6)$$

Thus, $F_m = (q_i + 0.2178 q_{ia})/q_p$. (7)

We must still consider pump efficiency, defined as

$$E \equiv P_t / P_i \quad (8)$$

$$= [(p_d - p_i)(q_i + q_{iaw})]/[(p_p - p_d) q_p] \quad (9)$$

$$= F_p F_m. \quad (10)$$

Thus, pump efficiency is merely the product of the pressure-recovery and mass-flow-rate ratios.

$$f_{td} = 0.1 + a(F_{pdp})^b (F_{aws})^c (F_A)^d. \quad (11)$$

By taking logarithms of each variable and using regression programs in the SAS Statistical Package,™ Version 6.03, values for a, b, c, and d were found:

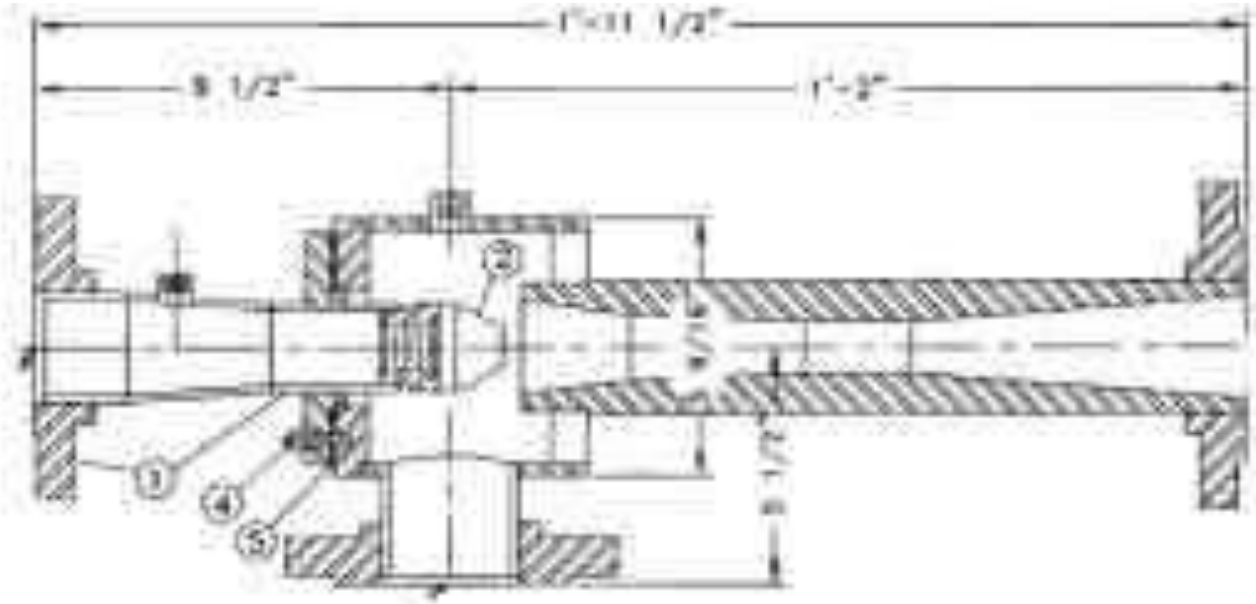
$$a = 4.1 \times 10^{-3}, \quad (12)$$

$$b = -2.6, \quad (13)$$

$$c = 0.62, \quad (14)$$

and $d = 0.53$. (15)

Simpson, David, **Use of Eductor for Lifting Water Eductor Theory**



From the group of thermo-compressors that includes Air Ejectors, Evacuators, Sand Blasters, Jet Pumps, and Eductors • High pressure gas entrains and compresses suction gas and the combined stream is left at an intermediate pressure • Compression rations 1.3-2.0 are possible, limited by: – Power gas pressure – Suction pressure – Discharge pressure

Eductor Theory
• From the family of thermo-compressors that include: Air Ejectors, Evacuators, Sand Blasters, Jet Pumps, and Eductors
• High pressure gas entrains and compresses suction gas and the combined stream is left at an intermediate pressure
• Compression ratio's between: 1.3 & 2.0 are possible, limited by:
- Power gas pressure
- Suction pressure
- Discharge pressure

Author claims: At 10 psig, 60°F, and 168 MCF/d you can Move 0.75 bbl/day up 3,000 ft of 2-3/8 tubing

Eductor Rules of Thumb

• For an eductor with gas as the power fluid:
– Exhaust pressure should be less than ½ power-gas pressure (in absolute terms)
– Exhaust pressure should be less than twice suction pressure
– Mass flow rate of power gas will be about twice suction mass flow rate.
• With liquid as the power fluid:
– You don't get to critical flow so the exhaust pressure can be higher
– More "compression ratios" are possible (i.e., the ratio of the exhaust over the suction can be more than 2)
• If the power gas is a mixture of gas and liquid, calculate the density and the mass flow rate carefully

Editor note: max lift of liquid water commonly taken as atmospheric pressure less vapor pressure at 32F converted to feet by water density or 14.73*144/62.4= 34feet, salty water have less lift. Evaporative processes may work to dewater by evaporation to greater depths.

Simpson, David, **Use of Eductor for Lifting Water Eductor Theory**

Gas is Methane	
Patm=	12 Psia
m° power=	600 lb/hr
m°suct=	300 lb/hr
Ppwr=	100psig
Tpwr=	80°F
Pdisc=	21 psig
R=	1.5

$$\rho_{power} = \frac{MW * P}{10.73 * T * Z} = \frac{(16) * 112\,psia}{10.73 * (460 + 80) * .97} = 0.319 \frac{lbm}{ft^3}$$

$$q_{pwr} = \frac{\dot{m}_{pwr}}{\rho_{pwr}} = \frac{600 \, {}^{lbm}/_{hr}}{0.319 \, {}^{lbm}/_{ft^3}} \left(\frac{112 * 520 * 0.98}{14.73 * 540 * 0.97} \right) \frac{24}{1000} = 333 \, {}^{MCF}/_{d}$$

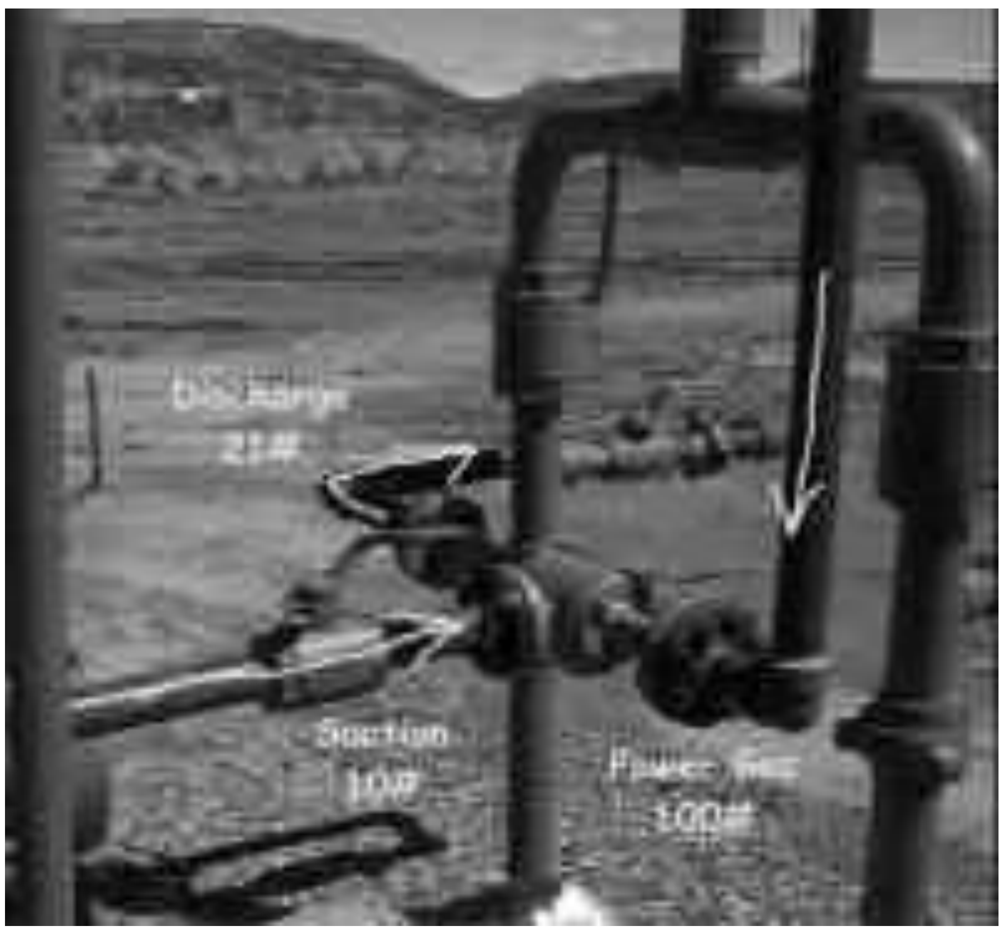
$$P_{suction} = \frac{P_{disch}}{R} = \frac{(21\,psig + 12\,psi)}{1.5} = 22\,psia$$

$$\rho_{suct} = \frac{16 * 22}{10.73 * (460 + 60) * .98} = 0.064 \frac{lbm}{ft^3}$$

$$q_{suct} = \frac{300 \, {}^{lbm}/_{hr}}{0.064 \, {}^{lbm}/_{ft^3}} \left(\frac{22 * 520 * .98 * 24}{14.73 * 520 * .98 * 1000} \right) = 168 \, {}^{MCF}/_{day}$$

Eductor Rules of Thumb

- For an eductor with gas as the power fluid:
  - Exhaust pressure should be less than ½ power-gas pressure (in absolute terms)
  - Exhaust pressure should be less than twice suction pressure
  - Mass flow rate of power gas will be about twice suction mass flow rate.
- With liquid as the power fluid:
  - You don't get to critical flow so the exhaust pressure can be higher
  - More "compression ratios" are possible (i.e., the ratio of the exhaust over the suction can be more than 2)
- If the power gas is a mixture of gas and liquid, calculate the density and the mass flow rate carefully



References various source per above

1. Rankin, J.M.: Proc., Royal Soc. of London (1870) No. 123, 80-83.
2. Blais, R.N. et al.: "Test Facilities for Gas-Lift Valves and Hydraulic Jet Pumps," Advances in Test Measurements, Instrument Soc. of America, Research Triangle Park, NC (1985) 22, 525-30.
3. Bonnington S.T., King A.L. and Hennings J.A."Jet Pumps and Ejectors, Second Revised Ed., a State of the Art Review and Bibliography" BHRA Fluid Eng. Ser. (1976)
4. Brown, K.E. "The Technology of Artificial Lift Methods" Vol., 2b Pet. Pub. Co., Tulsa (1980)
5. Corteville. J. C., Ferschneider, Hoffmann, F. C., and Valentin, E.P., Sep.(27-30), 1987; "Research on Jet Pumps for Single and Multiphase Pumping of Crudes", published in the 62nd Annual Technical & Exhibition held: Dallas TX, Sept. 27-30 1987. pp.437-448. paper SPE 16923
6. Cunningham, R.G.: "The Jet Pump as a Lubrication Oil Scavenge Pump for Aircraft Engines," Technical Report 55-143, Wright Air Development Center, Dayton, OH (1955).
7. Cunningham, R.G.: "Jet Pump Theory and Performance with Fluids of High Viscosity," Trans., ASME (1957) 79, 1807-20.
8. Cunningham, R.G., Hansen. A.G. and Na, T.Y. 1970, "Jet-Pump Cavitation," ASME Journal of Basic Engineering, pp.483-494
9. Cunningham, R.G.: "Gas Compression with the Liquid Jet Pump," J. Fluid Eng. (1974) 203-15; Trans., ASME, Series I, 196.
10. Cunningham, R.G. and Dopkin, R.J.: "Jet Breakup and Mixing Throat Lengths for the Liquid Jet Gas Pump," J. Fluid Eng. (1974) 216-26; Trans., ASME, Series I, 96.
11. Cunningham, R.G., Hansen, A.G., and Na, T.Y.: "Jet Pump Cavitation," J. Basic Eng. (1970) 11-20; Trans., ASME, Series D, 92, 11-20.
12. De Ghetto, G., Riva, M., and Giunta, P. 1994. Jet Pump Testing in Italian Heavy Oils. Paper SPE 27595 presented at the SPE European Production Operations Conference and Exhibition, Aberdeen, 15-17 March. DOI: http://www.spe.org/elibrary/servlet/spepreview7id = 27595-MS.
13. Gosline, J. E., and O'Brien, M. P. 1934, "The Water Jet Pump", Publications in Engineering, University of California, 3, pp. 167-190. (1942) 3, No. 3, 167-90. *Conflicting pub. Dates both given'34 & '42*
14. Gosline, J.E. and O'Brien, M.P.: The Water Jet Pump, U. of California Publications in Engineering, U. of California Press, Berkeley, CA (1942) 3, No. 3, 167-70. *Conflicting pub. Dates both given'34 & '42*
15. Grupping, A.W., Coppes, J.L.R., and Groot, J.G.: "Fundamentals of Oilwell Jet Pumping," SPEPE (Feb. 1988) 9-14.
16. Haberman, W.L. and Morton, R.K., 1955, David Taylor Model Basin Report 802, Naval Research Center. Bethesda MD.
17. Jiao, B.: "Behavior of Hydraulic Jet Pumps When Handling a Gas-Liquid Mixture," MS thesis, U. of Tulsa, Tulsa, OK (1985).
18. Jiao, B.: "Performance Model for Hydraulic Jet Pumping of Two-Phase Fluids." PhD dissertation, U. of Tulsa, Tulsa, OK (1988).
19. Jiao, B., Blais, R.N., and Schmidt, Z., 1990, "Efficiency and Pressure Recovery in Hydraulic Jet Pumping of Two-phase Gas/Liquid Mixtures", SPE Production Engineering, Vol.5, No.4, pp.361-364.
20. Jiao, B., Schmidt, Z., and Blais, R.N.: "Efficiency and Pressure Recovery in Hydraulic Jet Pumping of Two-Phase Gas/Liquid Mixtures, "SPEPE (Nov. 1990) 361-65.

21. Liu, T. and Xu, C. 1992. Calculation Methods and Amelioration for the Viscosity of Diluted Heavy Oil. Storage and Transportation of Gas & Oil 11 (2): 56-62.
22. Marco Villa. De Ghetto, G., Paone. F., Giacchetta G. and Bevilacqua, M., 1999, "Ejectors for Boosting Low-Pressure Oil Wells", SPE Prod. & Facilities 14(4).
23. Mueller. N. H. G., May, 1964, "Water Jet Pump" Proceedings of the ASCE, 90, pp.83-112.
24. www.nrc.gov/reactors/bwrs.html.
25. Noronlia, F.A.F, Franca, F.A., And Alhanati. FJ.S. 1988. "Improved Two-phase Model for Hydraulic Jet Pumps". SPE Journal. September pp.227-231.
26. Petrie, H.L., Wilson, P.M., and Smart, E.E. Jet Pumping Oil Wells: Part 1. Design Theory, Hardware Options and Application Considerations. World Oil 197 (6): pp.51-56 (Nov. 1983)
27. Petrie, H.L., Wilson, P.M., and Smart, E.E. Jet Pumping Oil Wells: Part 2. Hand-Held Computer Programs for Installation Design. World Oil 197 (7): pp 109-114. (Dec. 1983)
28. Petrie, H.L., Wilson, P.M., and Smart, E.E.; Jet Pumping Oil Wells: Part 3. How Design Calculations Compare with Actual Field Performance. World Oil 198 (1): pp 101-108. (Jan. 1984)
29. Petrie, H. and Brown, K.E.: The Technology of Artificial Lift Methods, Penn Well Publishing Co., Tulsa, OK (1980) 2b, Chap. 6.
30. Qu, Z., Xing, J., Zhang, H., and Liu, H. 2000a. Study of Mixing Light Hydrocarbon in Deep Heavy Oil. Oil and Gas Recovery Technology 1 (3): 26-28.
31. Quan, Z. 1985. Study on the Calculation Method for the Viscosity of Diluted Heavy Oil. Storage and Transportation of Gas & Oil 4 (5): 33-36.
32. Reddy, Y. R. and Kar, S., 1968, "Theory and Performance of Water Jet Pump", ASCE Journal of Hydraulics. 94, 5, pp.1261-1281.
33. Sanger N.L., "Non-Cavitating and Cavitating Performance of Two Low-Area-Ratio Water Jet Pumps with Throat Lengths of 5.66 Diameters" NASA Technical Note 0-4759 (Aug. 1968)
34. Smart, E.E.: "Jet Pump Geometry Selection," Southwestern Petroleum Short Course, Lubbock, TX (1985).
35. Tjondrodiputro, B., Gaul, R.B. and Gower, G.H. *1986*. Hydraulic Jet Pumping in a Remote Location: Part 1. World Oil *203* (6): 35-37. (Dec. 1986)
36. Tjondrodiputro, B., Gaul, R.B. and Gower, G.H. *1987*. Hydraulic Jet Pumping in a Remote Location. World Oil *204* (1): 80-83.
37. Zandi, I., and Govatos, 1970, "Jet Pump Slurry Transport". Hydro-transport 1, First International Conference on the Hydraulic Transport of solids in Pipes, (BHRA), University of Warwick. U.K. pp. L2.17-L2.32.

References to basic engineering principles & texts omitted as such information available from various handbooks. Most handbooks have one or more sections devoted to Jet or Hydraulic or Hydro or Thermo pumping - compression and Jet Mixing. Industrial examples are: Mud Pans, Dry Chemical Shear Mixers, Tank Mixers, Air-Fuel Pilot Mixers, Air aspirated Smokeless Flares, Thermo-Compressors used for MED Evaporators, Steam Ejectors for NC gas removal in Steam Condensers, Powering Jet Skies and Jet Boats for low draught hulls, use as water well lifts both surface (shallow) & submerged (deep well lift), load/unload solids. All work by momentum transfer and in most cases converting velocity head to pressure head. There are just too many general examples to cover. So these references limit to oil or gas jet pumping.

Vendor Data Schlumberger and Weatherford

Nozzle Number	3	4	5	6	7	8	9	10	11
Nozzle Area (in.^{2})	0.0040	0.0052	0.0067	0.0086	0.0110	0.0144	0.0186	0.0240	0.3100
Throat Number	3	4	5	6	7	8	9	10	11
Throat Area (in.^{2})	0.0100	0.0129	0.0167	0.0215	0.0278	0.0359	0.0599	0.0774	0.7740
				7A-	7A	7B	7C		

Applications

- Deviated and dogleg wells
- Deep wells
- Offshore platforms
- Remote and urban locations
- Multiple zones
- Complex well completions
- Production fluids with sand and solids
- Paraffinic fluid and heavy oil production
- Interval testing
- Retrofitting wells that previously used other lift methods
- Corrosive well fluids
- Gas well de watering
- Well cleanup/swab services
- Testing services
- Drillstem testing services

Benefits

- Economical
- Environmentally friendly
- Low maintenance
- Interchangeable components

Features

- Flexible production capacity
- Transportable modular surface equipment
- Field repairable
- No moving parts
- Hydraulic coiled tubing or slickline deployment and Retrieval
- Can use oil or water as power fluid
- Interchangeable into existing pump cavities

Nozzle Throat Combination Annular Area, Sq. inch						
Nozzle	A-	A	B	C	D	E
1		0.0036	0.0053	0.0076	0.0105	0.0143
2	0.0029	0.0046	0.0069	0.0098	0.0136	0.0184
3	0.0037	0.0060	0.0089	0.0127	0.0175	0.0231
4	0.0048	0.0077	0.0115	0.0164	0.0227	0.0308
5	0.0062	0.0100	0.0149	0.0211	0.0293	0.0397
6	0.0080	0.0129	0.0192	0.0273	0.0378	0.0513
7	0.0104	0.0167	0.0248	0.0353	0.0488	0.0663
8	0.0134	0.0216	0.0320	0.0456	0.0631	0.0856
9	0.0174	0.0278	0.0414	0.0589	0.0814	0.1106
10	0.0224	0.0360	0.0534	0.0760	0.1051	0.1428
11	0.0289	0.0464	0.0690	0.0981	0.1358	0.1840
12	0.0374	0.0599	0.0891	0.1268	0.1749	0.2382
13	0.0483	0.0774	0.1151	0.1633	0.2265	0.3076
14	0.0624	0.1001	0.1482	0.2115	0.2926	0.3974
15	0.0806	0.1287	0.1920	0.2731	0.3780	0.5133
16	0.1036	0.1668	0.2479	0.3528	0.4881	0.6629
17	0.1344	0.2155	0.3203	0.4557	0.6304	0.8562
18	0.1735	0.2784	0.4137	0.5885	0.8142	1.1058
19	0.2242	0.3595	0.5343	0.7600	1.0516	1.4282
20	0.2896	0.4643	0.6901	0.9817	1.3583	1.8444
21	0.3743	0.6000	0.8916	1.2681	1.7544	2.3830

Typical Jet Pump Size, Flow, & Lift Capacity			
Tubing OD (in.)	Pump OD (in.)	Max Production (B/D)	Max Lift Capacity (ft)
1.660	1.25	1,000	10,000
2.375	2	2,000	15,000
2.875	2.5	6,000	15,000
3.500	3	10,000	15,000
4.500	4	30,000	15,000

Application guidelines

- Sufficient reservoir pressure is required to ensure reasonable jet pump efficiency (100 psi per 1000 ft of well depth is a general rule).
- Heavy oil lifting systems typically require the power fluid to act as a diluents.
- The typical power fluid to well fluid lifted ratio is 0.7/1-2/1.
- Jet pumps are more tolerant than piston pumps of harsh downhole conditions, including abrasives, gas, and paraffins.

Selection process

- The primary objective is to achieve maximum production from the well using the minimum amount of hydraulic horsepower by matching of jet pumps to well inflow

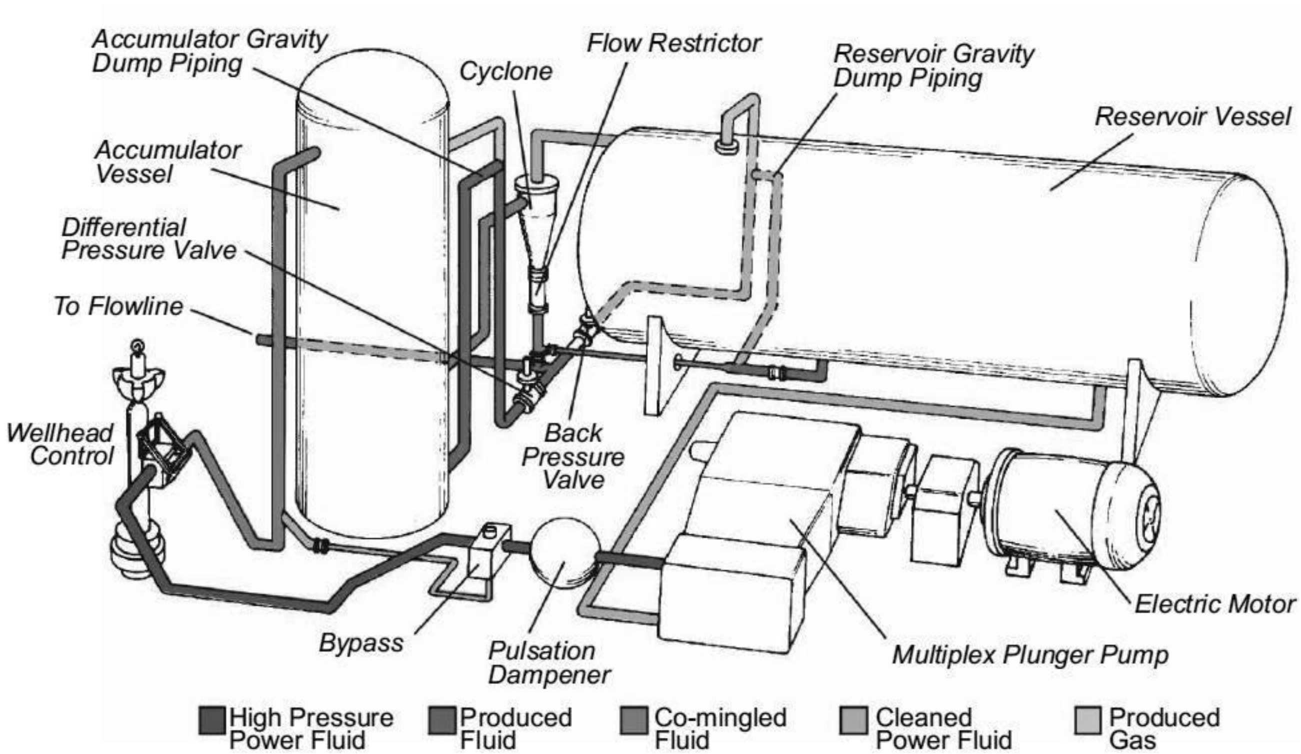
Performance

- Fine-tune the selections for optimal application fit by varying nozzle and throat combinations.
- The combination nomenclature can be seen in the table illustration. If a matching nozzle and throat pair were selected, such as a 7 nozzle and a 7 throat, it would be a 7A combination
- If one size smaller throat were selected, it would be termed an A- (as in 7A- in this example).
- One size bigger throat would be a B (as in 7B in this case) and two sizes bigger would be a C (7C in this case). It is unusual to select bigger than a I) combination.

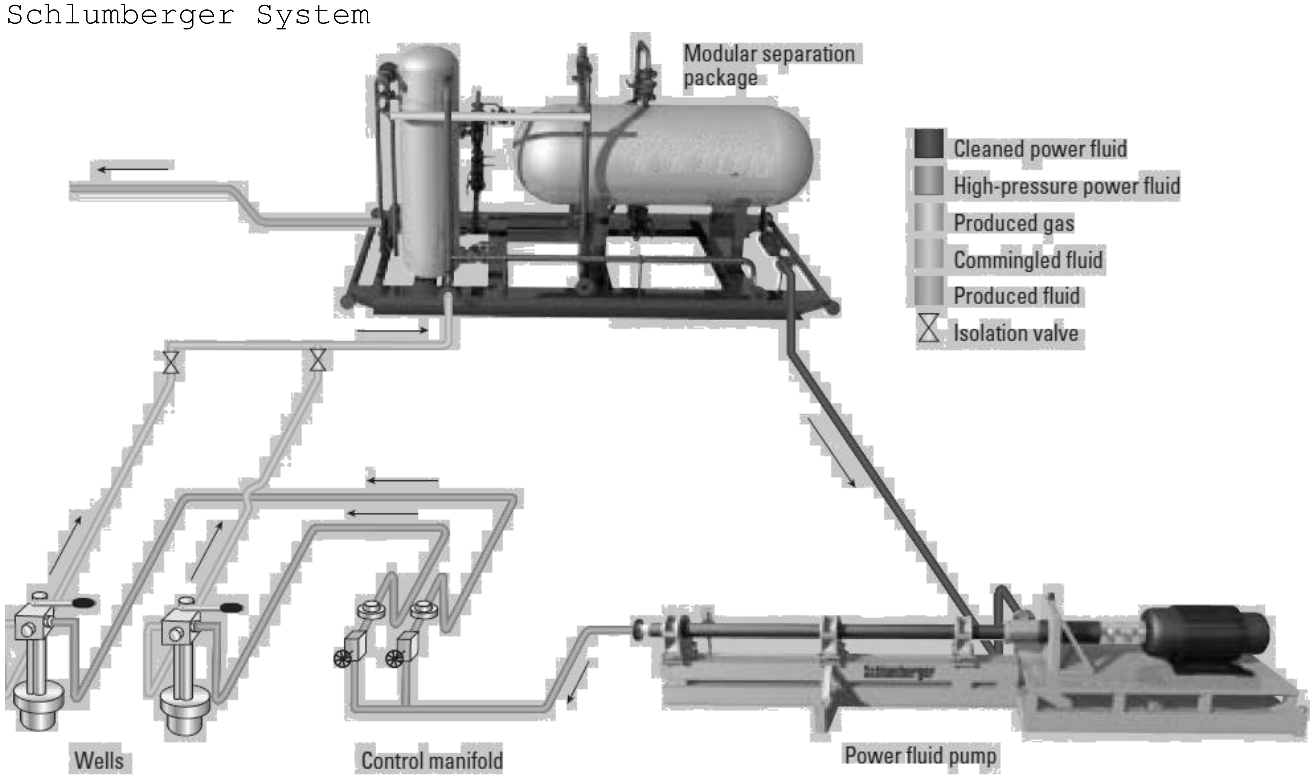
Permanent Production Applications		Temporary Services Applications
· Viscous oil wells		· Well cleanup
· High-solid-loaded wells		· Standard-flow well testing
· Gas well dewatering		· Reverse-flow well testing
· Deviated oil wells		· Drillstem testing
· Coal bed well dewatering		
· Retrofitted gas lift wells		
· Artificial lift in multiple zones		

Weatherford Application of Freestyle JP

Well type:	Gas; four directional, one vertical
Casing/tubing:	5 in., 17 lb/ft; 2-7/8 in.; 1-1/4 in.
Depth in:	8,000 ft
Flow rate:	200 bbl/d
BHA details:	Coiled tubing
Objectives	I Results
• Dewater wells producing up to 200 bbl/d of water. Progressing cavity pumps (PCPs) had been used previously, but numerous problems were experienced including rod/tubing wear, broken rotors resulting from torque problems, surging, and corrosion. The high volume of gas precluded use of rod pumps. Electric submersible pumps (ESPs) were ruled out because of the difficulty in producing at rates lower than 300 bbl/d, the high-angle deviation of the wells, and lack of electricity at the site.	• The freestyle jet pumps are producing at rates up to 800 bbl/d and at less than 50 bbl/d by changing only the nozzle and/or throat. This task was quickly accomplished, as freestyle operation capability meant that the pump could be circulated in and out hydraulically. • The jet pumps had no trouble dealing with the high deviation angles or other problems encountered by PCPs. The jet pumps have proven highly reliable in removing the required volumes of water to enable gas flow.



Weatherford System
Schlumberger System



Weatherford System Typical Size

Description	accumulator	Reservoir
Vessel size - diameter and seam length	36" x 36"	60" x 120"
Manway size	11"x18"	14" x 18"
Std. working pressure	225 psi	175 psi
Optional working pressure	300 psi	240 psi
Fluid capacity	4-1/3 bbls	41 bbls