Chair of Petroleum and Geothermal Energy Recovery

Master's Thesis

Jet pump performance analysis and optimization plan

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Danksagung / Acknowledgement

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To conclude, I acknowledge all that I have accomplished and survived, big and small.

Abstract

By observing the oil and gas industry we can estimate that up to 90%, if not more, of the world's wells are currently using artificial lift systems and the rest, will be using artificial lift at a certain point of their lives if not abandoned due to economic reasons.

The selection of the most economic system is necessary for operators to achieve the maximum potential of the production of their fields, which also arises challenges that come along with artificial lifting operations. One of the major challenges is workover.

The unavailability of workover rigs and the mean time between failures have contributed to the decrease of the profitability of the wells. These issues pushed engineers to think of an alternative. A jet pump was initially implemented as a backup for an electric submersible pump that failed to reduce the production deferment until the start of the workover. However, several failures have occurred in a short period of time and the selection of the optimum pump was a challenge.

This master thesis was conducted to investigate the actual jet pump performance and review all failure and recommend any optimization possibilities. The aim of this thesis is also to design jet pump using state of the art integrated asset modeling.

Finally, a jet pump design tool was created to select the optimum nozzle-throat size, and determine injection rates to reach the highest production potential by also taking into consideration possible cavitation problems.

Kurzfassung

Wenn wir die Öl- und Gasindustrie beobachten, wird ersichtlich, dass nahezu 90 %, wenn nicht sogar mehr, der weltweiten Bohrungen derzeit „Artificial-Lift“ Systeme verwenden, und der Rest wird den künstlichen Auftrieb an einem bestimmten Punkt seines Lebens nutzen, wenn er nicht aus wirtschaftlichen Gründen aufgegeben wird.

Die Auswahl des wirtschaftlichsten Systems ist notwendig, damit die Betreiber das maximale Produktionspotential ihrer Ölfelder erreichen können, was auch Herausforderungen mit sich bringt, die mit „Artificial-Lift“ Vorgängen einhergehen. Eine der größten Herausforderungen ist das Workover.

Die Nichtverfügbarkeit von „Workover-Anlagen“ und die Zeitintervalle zwischen den Ausfällen haben zur Verringerung der Rentabilität der Bohrungen beigetragen. Durch diesen Umstand sahen sich die Ingenieure gezwungen, eine Alternative diesbezüglich zu finden. Als „Backup“ für die bereits verwendete „Electrical-Submersible-Pump“ wurde eine „Jet-Pump“ implementiert, da das bestehende System die resultierende Produktionsverzögerung bis zum Beginn des „Workover“ nicht reduzieren konnte. Innerhalb kürzester Zeit sind jedoch mehrere Fehler aufgetreten und die Auswahl der optimalen Pumpe gestaltete sich äußerst schwierig.

Diese Masterarbeit entstand in Zusammenarbeit mit der OMV, um die tatsächliche Leistung dieser „Jet-Pump“ zu untersuchen. Dabei wurde eine hochmoderne Software zur Anlagenmodellierung verwendet, um alle Fehleranalysen zu überprüfen und Optimierungsmöglichkeiten zu empfehlen.

Schlussendlich wurde ein eigenes „Jet-pump design tool“ erstellt, mit welchem es möglich ist, die optimale „Nozzle-throat-size“ auszuwählen und die notwendige Einspritzrate zu bestimmen, um das höchste Produktionspotential, unter Berücksichtigung des Kavitationsproblems, zu erreichen.

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1 Introduction

The first oil well was successfully drilled in 1859 in the middle of a farm in northwestern Pennsylvania. Since that, the world oil consumption is increasing year after year and today our planet consumes about 90million barrels per day.

However, at a certain point of the well life, the reservoir is unable to provide either enough energy to produce the reservoir fluids at economic rates, because it is depleted or due to increased backpressure on the well. To satisfy the world's increasing needs for oil and gas consumption and to reach maximum profitability, companies have developed artificial lift systems to supply the required energy to the production system and to, eventually, enhance the production rate.

Several lifting mechanisms are available. Engineers are required to evaluate the advantages and disadvantages, operating costs, and production capabilities of every mechanism and select the most economical and profitable option. Therefore, a good understanding of the various components of the system and their interactions is required to determine what method will lift at the desired rates and from the required depths.

Jet pumping is one of the artificial lift systems used in the petroleum industry and was first described by Gosline and O'Brien in 1933. However, the prototype was not installed in an oil well until 1970 and has gradually gained acceptance as an efficient artificial lift system. Because it is a dynamic pump, jet pumps have characteristic performance curves similar to an electric submersible pump or that of a centrifugal pump.

A jet pump is small compared to other pumps and adaptable to all existing hydraulic pump bottom-hole assemblies and since there are no moving parts, a jet pump will tolerate a power fluid of poor quality, free gas, and sand production. However, they typically require higher pump intake pressures than conventional pumps to avoid cavitation problems.

One of the main advantages of jet pumps is the possibility of a free pump installation that allows the operator to remove and replace a pump using the existing surface hydraulic power fluid system without the need for a slick line. Also, in cases where workover rigs are unavailable or have a high cost, especially in fields where the mean time between failure is very short, jet pumps are considered as an alternative to avoid high production deferment volumes and high operating expenditures.

This problem was encountered in the field this thesis is studying, where workover rigs are not always available and therefore are very expensive. In addition, the waiting time for intervention is rather high, which induces high production deferment volumes. Combined with a low mean time between failures for the existing ESP systems, these issues tremendously affect the economics of the ESP operations reducing the wells profitability.

The other challenge that companies are facing is designing jet pumps, especially if they are using the integrated asset modelling. The pumps studied in this thesis are designed using JEMS, a jet pump modeling software that cannot be included in the integrated asset model.

Another alternative is PROSPER. However, designing a jet pump using this software is also challenging since it requires jet pump loss coefficients as input data. These coefficients are rather confidential data, and the purpose of this thesis is to estimate these loss coefficients in order to create reliable jet pump models that can be later used in the integrated asset model.

2 Jet Pump Fundamentals

2.1 Jet Pump Components

A jet pump is a hydraulic pumping system for artificial lift consisting of two main parts, a surface pump to transmit the fluid downhole and a downhole pump driven by the injected high-pressure fluid. Unlike other pumps, jet downhole pumps have no moving parts and consist mainly of three components: nozzle, throat and diffuser. A typical configuration of a jet pump is shown in Figure 1.

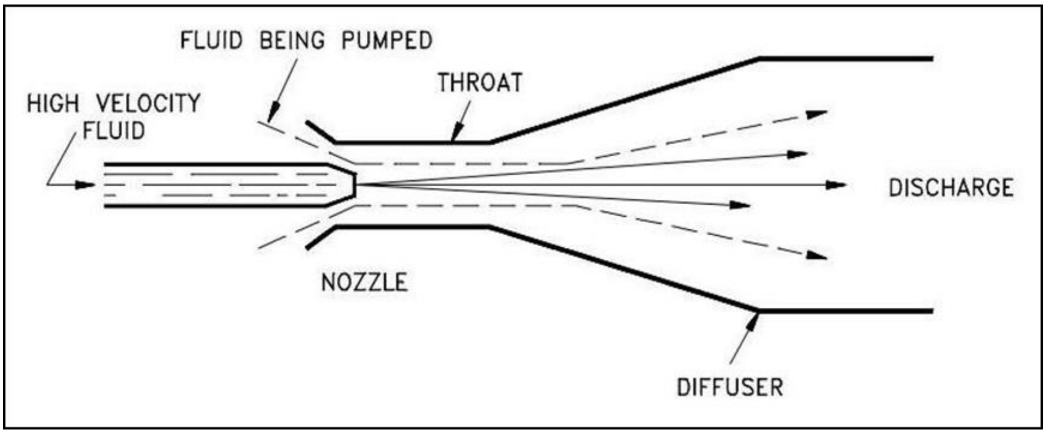


Figure 1: Typical jet pump configuration. [1]

2.1.1 Nozzle

The nozzle is a flow restriction and it is considered the engine of the jet pump, as it is responsible for converting high-pressure fluid (potential energy) to high-velocity fluid (kinetic energy).

2.1.2 Throat

The second part of the downhole jet pump is the throat, which is usually of a bigger size than the nozzle. The throat-entry is connected to the pump intake, which allows reservoir fluid to enter the jet pump. The reservoir fluids flow into the pump due to the pressure drop at the nozzle exit and throat entry area. The throat, also called mixing chamber, is responsible for mixing the injected power fluid and the produced fluid by exchanging momentum before reaching the diffuser section. [2]

The throat should be sufficiently long to accomplish the mixing of the two fluids and short enough to limit frictional losses. El-Sawaf reports that an optimum throat length is 7.25 times the throat diameter. According to Prabkeao and Aoki, the throat length decreases as the nozzle-throat diameter ratio increases. [2] [3]

The size of a jet pump depends on the cross section between the nozzle and the throat and also known as combinations. All jet pump designs in the following chapters will be using jet pumps with nozzle and throat sizes as listed in table 1:

Table 1: Nozzle and throat dimensions. [1]

No.	Nozzle area (in^2)	Diameter (in)	No.	Throat area (in^2)	Diameter (in)
1	0,0024	0,055279	1	0,0063	0,089674
2	0,0031	0,062376	2	0,0080	0,101187
3	0,0039	0,070384	3	0,0102	0,114177
4	0,0050	0,079419	4	0,0130	0,128835
5	0,0063	0,089615	5	0,0166	0,145375
6	0,0080	0,10112	6	0,0211	0,164038
7	0,0102	0,114102	7	0,0269	0,185097
8	0,0130	0,12875	8	0,0343	0,20886
9	0,0166	0,145279	9	0,0436	0,235673
10	0,0211	0,16393	10	0,0555	0,265929
11	0,0269	0,184975	11	0,0707	0,300068
12	0,0342	0,208722	12	0,0900	0,338591
13	0,0436	0,235518	13	0,1146	0,382059
14	0,0555	0,265754	14	0,1460	0,431108
15	0,0706	0,299871	15	0,1859	0,486453
16	0,0899	0,338368	16	0,2366	0,548903
17	0,1145	0,381808	17	0,3013	0,619371
18	0,1458	0,430824	18	0,3836	0,698886
19	0,1856	0,486133	19	0,4884	0,788608
20	0,2363	0,548542	20	0,6219	0,889849

Nozzle and throat diameters are sized so that the areas of different sizes are in a geometric progression. Each size is larger in flow area than the preceding size by a factor of 4/pi. The area of the nozzle to the area of the throat is called are ration, and this characteristic is used to compute pump performance in a well. Area ratios are identified by X, A, B, C, D. and E. Each ratio is associated with a nozzle number as shown in Table 2.

Table 2: Nozzle and throat combination possibilities. [1]

X	n-1
A	n
B	n+1
C	n+2
D	n+3
E	n+4

The number of the nozzle is referred to with a "n" in the table above; for example, if a combination 9B is used, the number "9" refers to the number of the nozzle and the throat number would be 10. Another example, a number 3 nozzle with a number 5 throat would make a 3C combination.

2.1.3 **Diffuser**

The diffuser is a conical tube that expands from the throat exit to the inner diameter of the tubing and it is responsible of converting the velocity (kinetic energy) of the mixed fluids to a pressure (potential energy) high enough to lift the fluid to the surface. Teamia et al. suggested a diffuser angle of 5.5°. However, the length and angle of the diffuser is difficult to determine without experiments. [4]

2.1.4 **Power Fluid**

Power fluid is a liquid pumped from the surface down to the jet pump's nozzle by using a reciprocating piston or centrifugal pump. The fluid can be pumped in a direct circulation system or in a reverse circulation system.

- Direct circulation system: the power fluid is injected through the tubing and the commingled fluids are produced through the annulus.
- Reverse circulation system: the power fluid is pumped through annulus and the commingled fluids are produced through the tubing.

There are two basic types of hydraulic pumping systems, open power fluid system and closed power fluid system.

In a closed power fluid system (CPFS) as shown in Figure 2 the reservoir fluid and the power fluid are never allowed to intermix throughout the entire system. An extra string of tubing is required both down hole and on the surface. Downhole, the extra string is used to bring the used power fluid back to the surface. On the surface, the extra string is used for carrying just the spent power fluid to the power fluid tank.

The complicated completion of the bottom hole design, makes the closed power fluid system more expensive than the open system. Therefore, the CPFS is less popular and is used much less than the open power fluid system.

However, a closed system is more common on offshore platforms due to the non-availability of space. Since the power fluid conditioning and reservoir tank only needs to be large enough to provide an adequate volume of power fluid to feed the multiplex pump, the size of the power fluid tank is relatively small and almost all of the produced fluid can be put directly into the flow line.

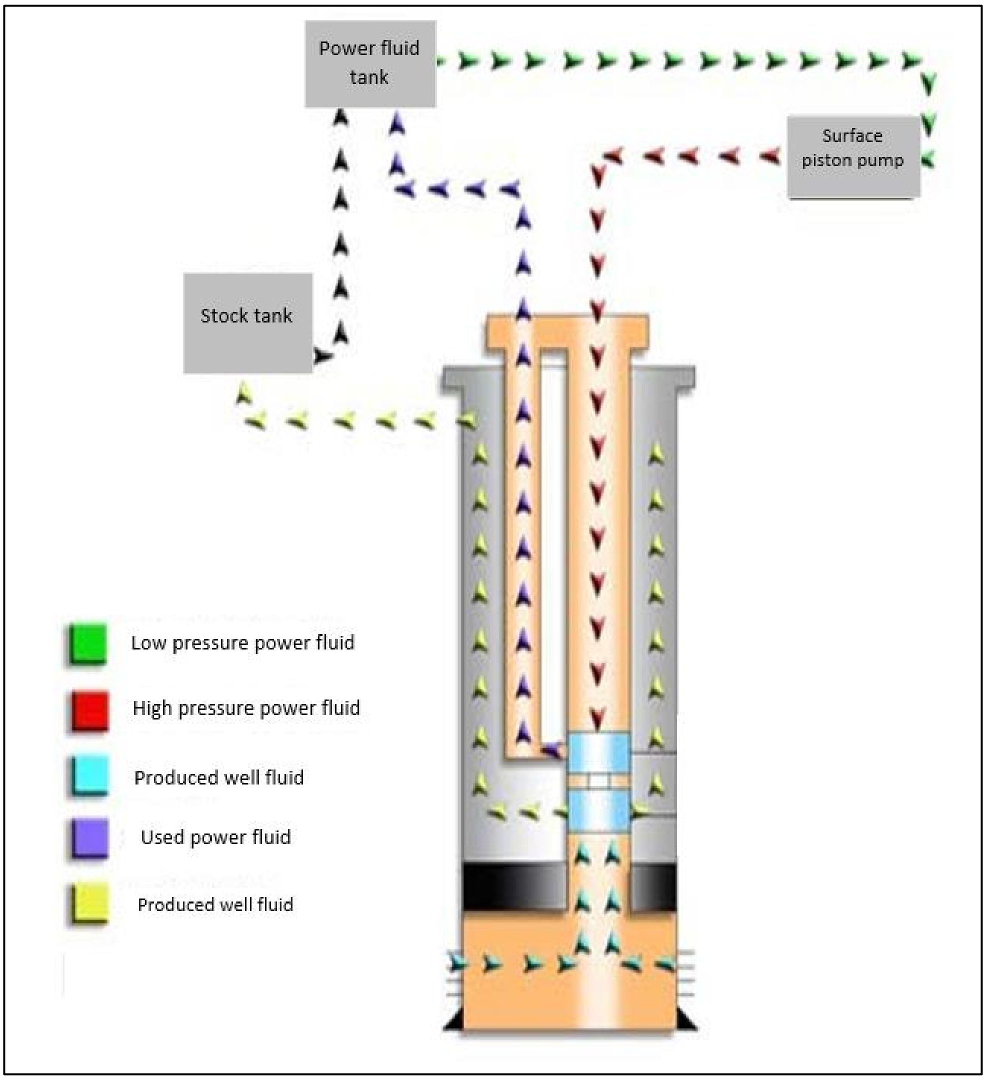


Figure 2: Closed power fluid system configuration. [5]

In an open power fluid system (OPFS) as shown in Figure 3, the power fluid mixes with the produced fluid down hole and then both fluids are returned to the surface in a commingled state. A jet pump can only operate in an open power fluid system as there is no physical boundary between the power fluid and the produced well fluid and they are allowed to mix. This system requires three down-hole fluid conduits to operate the pump: a tubing to contain the pressurized power fluid and direct it to the pump, a casing-tubing annulus to contain both power fluid and produced fluid and lift them to the surface. The casing below the pump is the third and last conduit and it provides the path for the produced fluids to flow into the inlet of the pump.

Since this is the simpler and the more economical of the two systems, it is by far the most commonly used system. Besides the simplicity and the economic advantage of the OPF system, the circulated power fluid is ideal for carrying corrosion, scale, and paraffin inhibitors to extend the life of the subsurface equipment. In addition, if there are emulsions downhole in the produced fluids, emulsion breakers can be added to the power fluid stream.

Also, when highly corrosive production fluids are being lifted, the clean power fluid can reduce the concentration of the corrosive elements by approximately 50-80 percent due to its diluting effect and reduces the viscosity to make lifting the heavy crude easier. Moreover, for production fluids that are extremely viscous or have a high paraffin content, the OPF system allows the circulation of heated liquids or dissolving agents through the power fluid in order to reduce the viscosity and remove any waxy build ups which might otherwise affect production. [5]

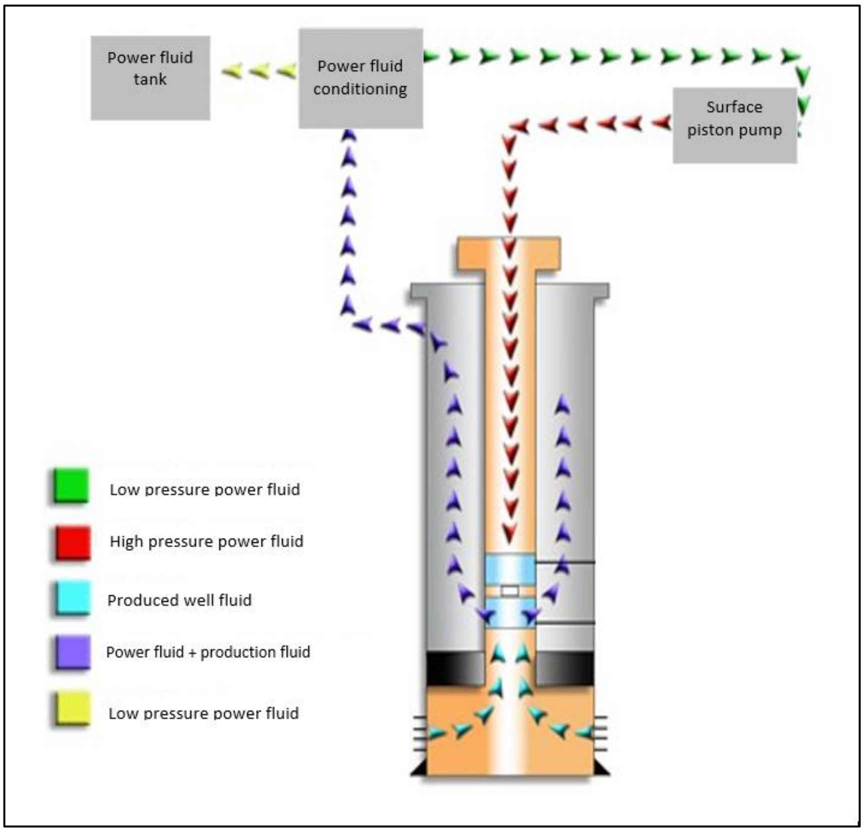


Figure 3: Open power fluid system configuration. [5]

The liquids used as power fluid are typically those produced from the well: water or crude oil. The selection of the power fluid depends on each fluid's advantages and limitations as shown in Table 3 and 4. [6]

Table 3: Oil as power fluid. [6]

Advantages	Limitations
Oil has a natural lubricity that water does not has which reduces the wear rates in both downhole and surface equipment.	Oil has potential fire hazard in case of a spill
Compatibility with produced reservoir fluids.	Pollution in case of a leak
	To high viscosity which may cause high friction losses
	Paraffin problems

Table 4: Water as power fluid. [6]

Advantages	Limitations
Cheap: lower cost installations compared to oil which requires bigger fluid tanks and a settling tank.	Water has no lubricity: Lubricants should be added to the power fluid
Environmentally friendly: fire hazard and surface contamination are reduced compared to oil.	Corrosion: corrosion inhibitor can be injected into the power fluid for corrosion control
Availability: produced water can be reused as power fluid after solid particles removal.	High salinity: salt crystallizing around some of the equipment like valves etc. and salt deposit in general can plug and damage equipment.

Power oil has a natural lubricity that water does not. This characteristic is important when operating the surface pistons pump due to their close tolerance clearances. Also, the compressibility of oil is far greater than the compressibility of water. This means that neither the surface pump nor the subsurface pump components are exposed to as much fluid hammer effect with oil as with water.

However, the high-viscosity of the oil compared to water can mean excessive friction losses in the system. This in turn increases the operating pressure and, consequently, the horsepower requirements for lifting the well.

On the other hand, water has low lubrication qualities which sometimes requires a chemical additive for lubrication (a surfactant) when using hydraulic piston pumps. Frequently, other chemicals are also used such as, scale and corrosion inhibitors to prevent corrosion and they are easily added at the multiplex pump suction via a chemical pump.

Beside usual chemical treatment for corrosion and scale depositions, salt crystals will occasionally be a problem in hydraulic systems using water as power fluid. The presence of the fresh water will dissolve the salt crystals and therefore, salt deposit problem can usually be solved by injecting fresh water with the power fluid.

Diesel has been also used as power fluid, but it is not as common as oil and water for the following reasons: Diesel is expensive compared to water or crude oil and it is highly flammable and may cause environmental problems in case of spill.
Gas is also another fluid that could be considered as a power fluid. However, its physical properties do not allow it to be as efficient as water and crude oil. When gas is flowing through the nozzle, the pressure drops leads to expansion of the gas and a temperature decrease and therefore hydrates will form. As result, the nozzle and throat area may be plugged.

Different power fluids can be employed with jet pump. However, the selection of one or the other depends on several factors. Compatibility between the power fluid and the reservoir fluid should be taken into consideration since permanent emulsions may form, making separation more difficult. Due to the increased water cut and the high viscosity of produced crude oil, water is more frequently used as power fluid. Also, due to ecological reasons, local restrictions and environmental reasons, it is considered to prohibit the use the produced crude as a power fluid. [6]

2.1.5 **Pump Installation**

The free-pump installation is one of the most significant advantages of hydraulic pumping systems. Free-pump installations permit circulating the pump to the bottom, producing the well, and circulating the pump back to the surface (retrieve the pump) using reverse circulation.

The pump is run in the hole by placing it in the power-fluid string and pumping fluid down the tubing until the pump reaches the setting depth and enters the seal bores. During normal pumping, this valve is held open by well fluid drawn into the pump suction. During pump-out, the normal flow of fluids is reversed at the surface and pressure applied to the discharge flow path of the pump. This reversal of flow permits the pump to be circulated to the surface, a process that normally takes 30 minutes to 2 hours, depending on depth, tubing size, and the circulating flow rate. Figure 4 shows a typical jet pump in and out circulation and installation:

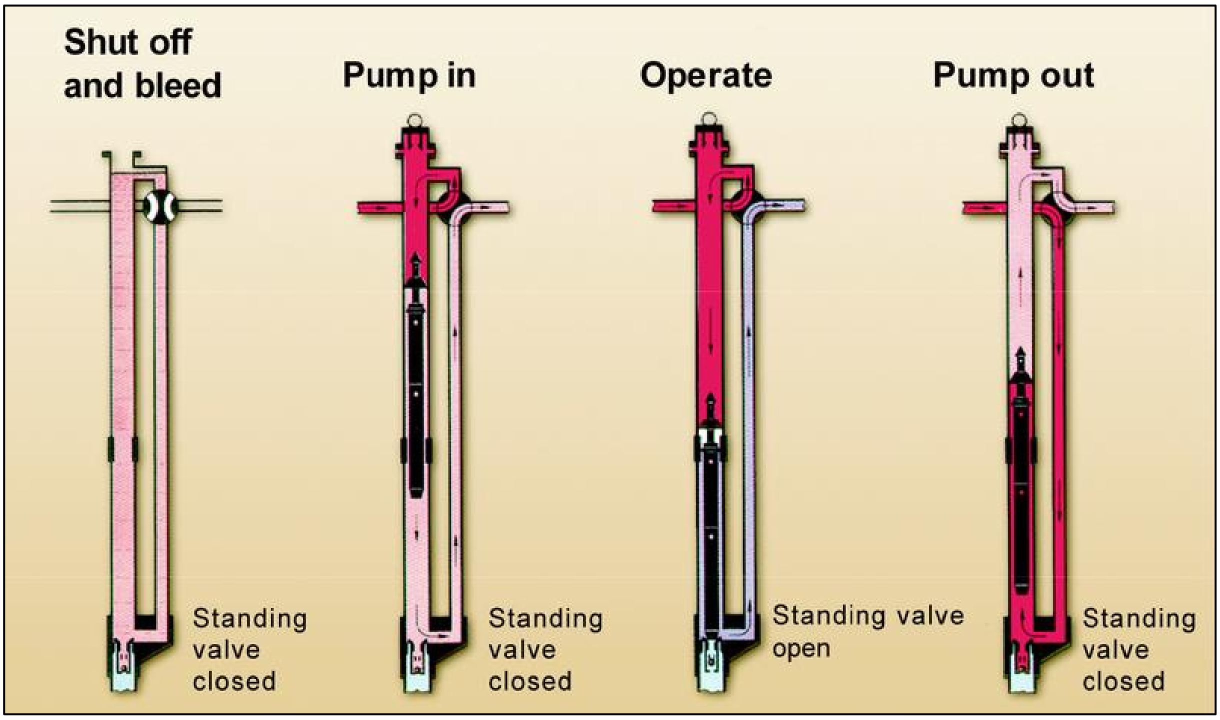


Figure 4: Jet pump in and out circulation. [7]

The benefits of being able to circulate the downhole pump in and out of the well is reducing the downtime and the ability to operate without the need for a workover unit, cable, or rod removal.

2.1.6 **Auxiliary Equipment**

A successful operation of any hydraulic downhole pumping installation depends on the effectiveness of the surface conditioning system in supplying clean power fluid for the surface pump and down-hole pump.

The presence of gas, solids, or abrasive materials in the power fluid will seriously affect the operation and wear life of the subsurface pump as well as the surface power unit. Therefore, the primary objective in conditioning crude oil or water for use as power fluid is to make it as free of gas and solids as is practical. In addition to removing gas and solid material, chemical treatment of the power fluid at the surface will also increase the wear life of the pumping equipment. [8]

The power fluid conditioning unit is composed of four main components as shown in Figure 5.

- Reciprocating multiplex pump
- Accumulator or “vertical vessel"
- Reservoir vessel or “horizontal vessel “
- Cyclone separator

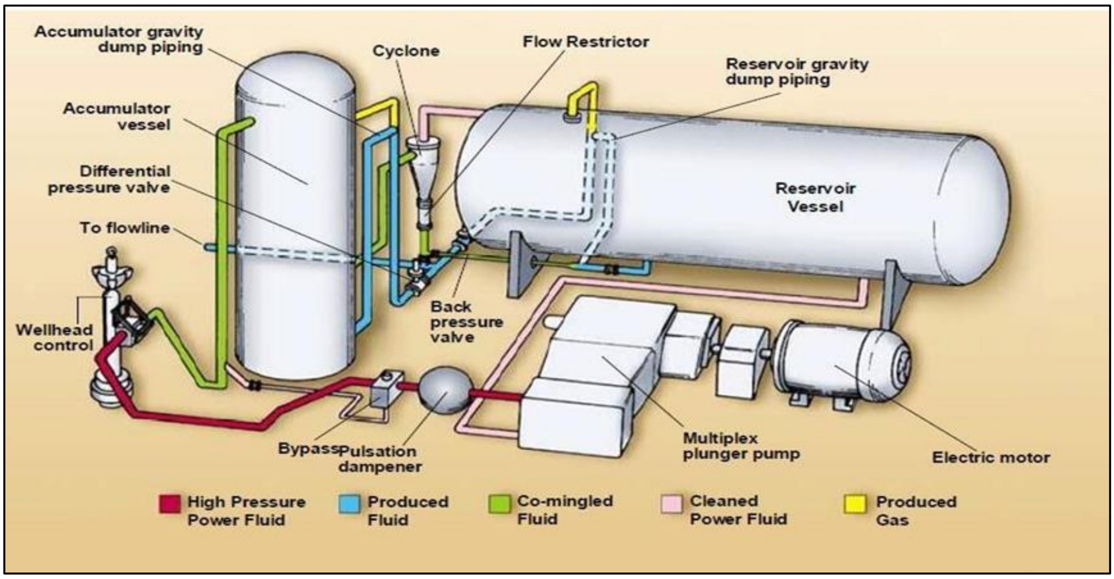


Figure 5: Surface fluid conditioning unit. [8]

The purpose of a surface fluid conditioning unit is to provide a constant supply of processed power fluid to operate the subsurface pumps. Produced fluid is a mixture of reservoir fluids and power fluid. Comingled fluids travel through a vertical separator, also called accumulator vessel, where first separation occurs. The produced fluids then travel through a cyclone separator to allow the separation of impurities such as solid particles. Finally, the partially cleaned fluid travels through a horizontal separator, also called reservoir vessel, where the final conditioning of the power fluid occurs. Then it is pumped downhole at high pressure. [8]

2.1.6.1 Reciprocating Multiple Pump

The reciprocating multiplex pump is a positive displacement pump and mainly composed of a plunger or piston to move media through a cylindrical chamber. They are also called high pressure pumps because they can deliver high pump pressures and are capable of handling both viscous fluids and solids. Multiplex pumps require a lot of maintenance and lubrication to avoid any unexpected pump failures such as a leakage or cavitation damage. [9]

2.1.6.2 Vertical Vessel and Horizontal Separator

Vertical separators are used for primary separation and fluid pressure reduction. Separated gas is discharged from the upper outlet of the vessel in order to flow through a vent system while the oil and water are separated in the vessel by the effect of gravity and different densities before entering the horizontal separator unit for further separation.

A horizontal separator receives the produced well fluid and power fluid from the vertical separator for further treatment and it is also designed as a power fluid reservoir to feed the surface pump. A typical vertical and horizontal separator configuration is shown in Figure 6. [10]

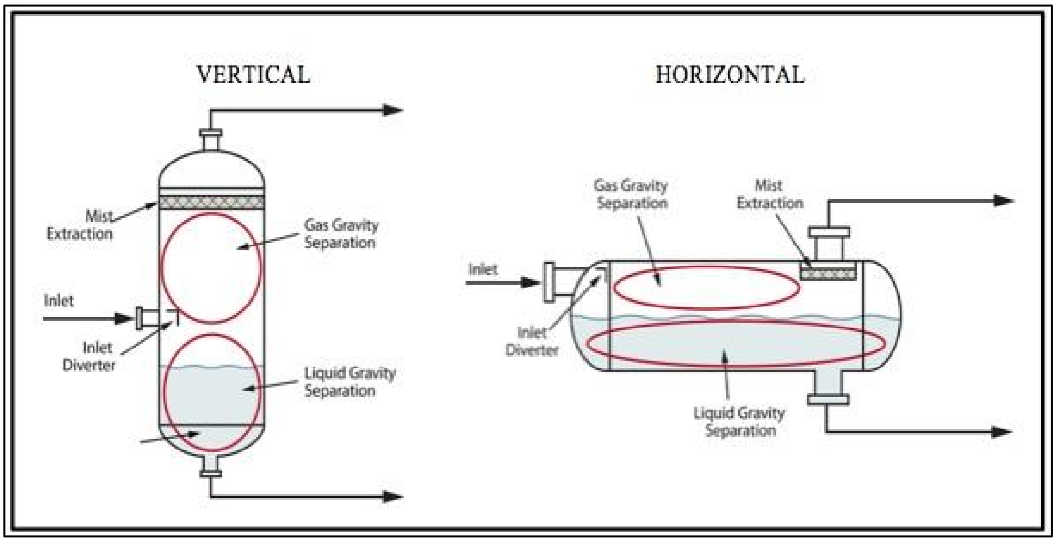


Figure 6: Typical horizontal and vertical separator configuration. [10]

2.1.6.3 Cyclone Separator

Fluids and solids enter the cyclone via an inclined feed pipe located at the top of the vessel and flow in a spiral pattern starting from the top and ending at the bottom. A typical cyclone separator is shown in Figure 7.

The high-speed rotation of the mixture is established due to the conical shape of the vessel. This creates a centrifugal force that pushes heavy and denser particles, which have higher inertia (solids), toward the walls of the separator, while lighter particles remain in the center of the separator and rise gradually until reaching the upper outlet of the conical vessel. [11]

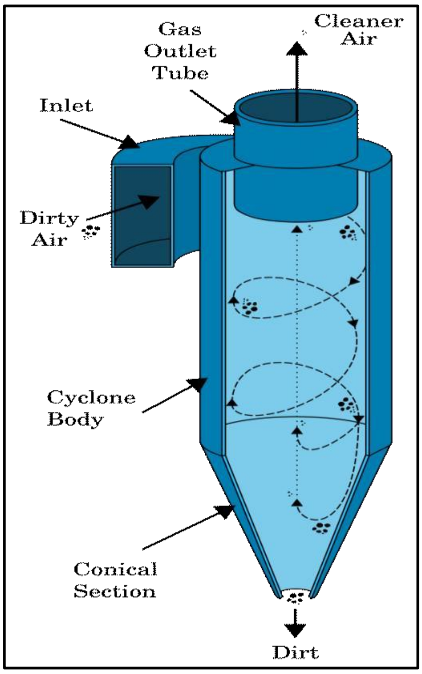


Figure 7: Cyclone separator. [11]

2.2 Working Principle

The pressure drawdown at the nozzle exit allows the reservoir fluid to flow to the well with a specific production rate that was designed when selecting the jet pump nozzle and throat combination. The selection of the optimum nozzle throat combination is explained in detail in chapter 2.3.

Velocity changes throughout the main parts of the jet pump is presented with the black curve in Figure 8. Pressure changes are presented with the red curve and can be divided into four main pressures or sections as listed below:

- P_N: Nozzle pressure
- P_d: Diffuser pressure
- P_s: Suction pressure
- P_a: Throat pressure

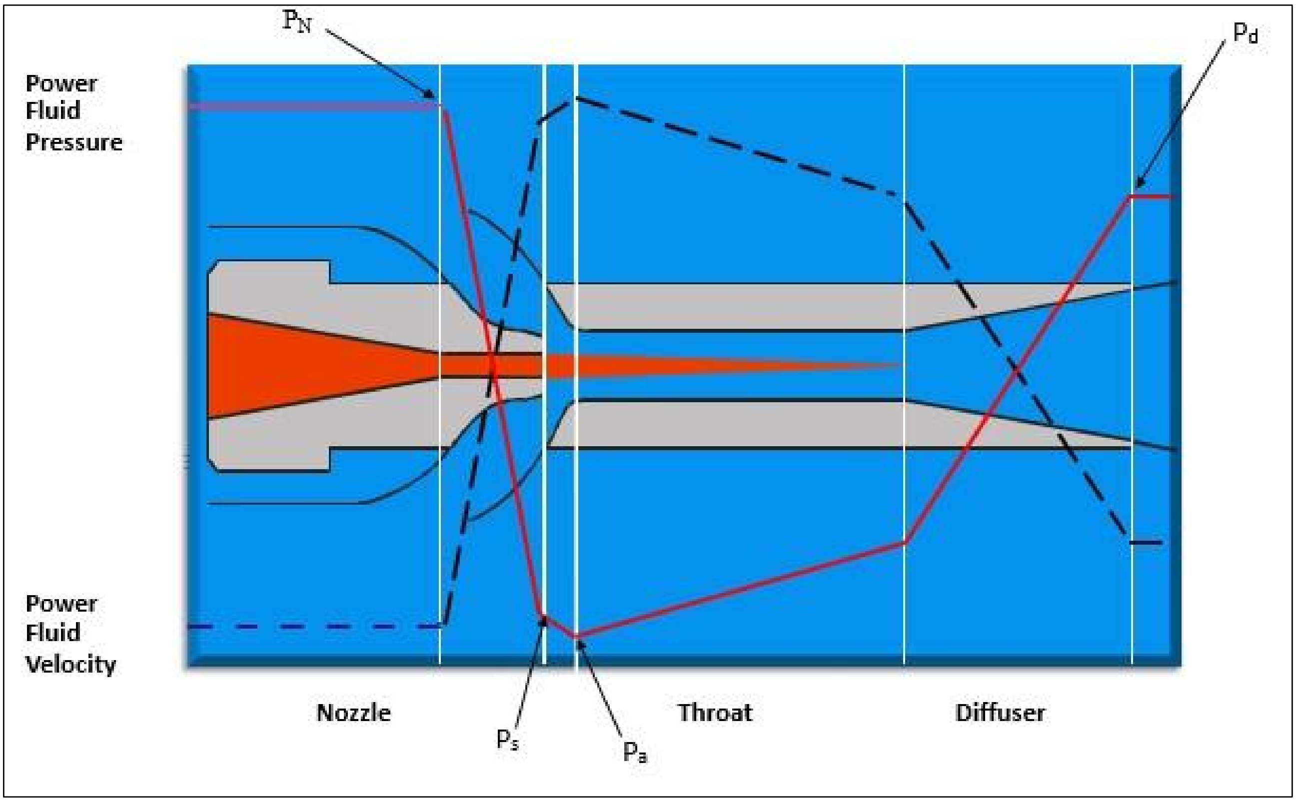


Figure 8: Power fluid pressure and velocity changes throughout the jet pump. [12]

The high-speed fluid is introduced to the well's low velocity fluid which creates a dragging action at the boundary between them due to the interaction of high-speed moving fluid particles and low speed fluid particles.

This dragging action is responsible for the mixing of both power fluid and produced reservoir fluid by momentum transfer. The momentum is transferred from the liquid with high velocity, in this case the power fluid, to the low velocity fluid, produced reservoir fluid.

This dragging action occurs at the area between the nozzle exit and the throat entrance. Both fluids are completely mixed and forms a homogeneous fluid once they reach the throat, also called mixing chamber.

The fluid mixture leaves the throat and enters a diffuser area. The area increase has an opposite effect of that from the nozzle. Fluid mixture leaves the diffuser after its pressure is increased and its velocity is decreased at the same time. The discharge pressure of the diffuser should be high enough to lift the mixed fluid to the surface. However, pump discharge pressure or the liquid pressure after exiting the diffuser is slightly lower than the pressure at the nozzle area. [12]

2.2.1 Main Equations Governing Fluid Flow in the Jet Pump

The working principle of the jet pump has been known since 1938 and it is based on Bernoulli's principle which states that an increase in velocity occurs simultaneously with a pressure decrease. Conservation of mass and momentum also play a major role in the way jet pumps work. The three main equations governing the fluid flow through a jet pump are explained in this chapter.

2.2.1.1 Continuity Equation

The continuity equation states that for an incompressible, uniform and steady state flow, the quantity of fluid entering at one side of tube is equal to the quantity of fluid leaving it, knowing that there is no leak or source in the tube.

Consider a tube with an inlet cross-section A_1, velocity V_1, outlet cross-section A_2, and velocity V_2 as shown in Figure 8. The mass of fluid flowing through cross-section A_1 is equal to the mass flow through cross-section A_2 at the same point in time.

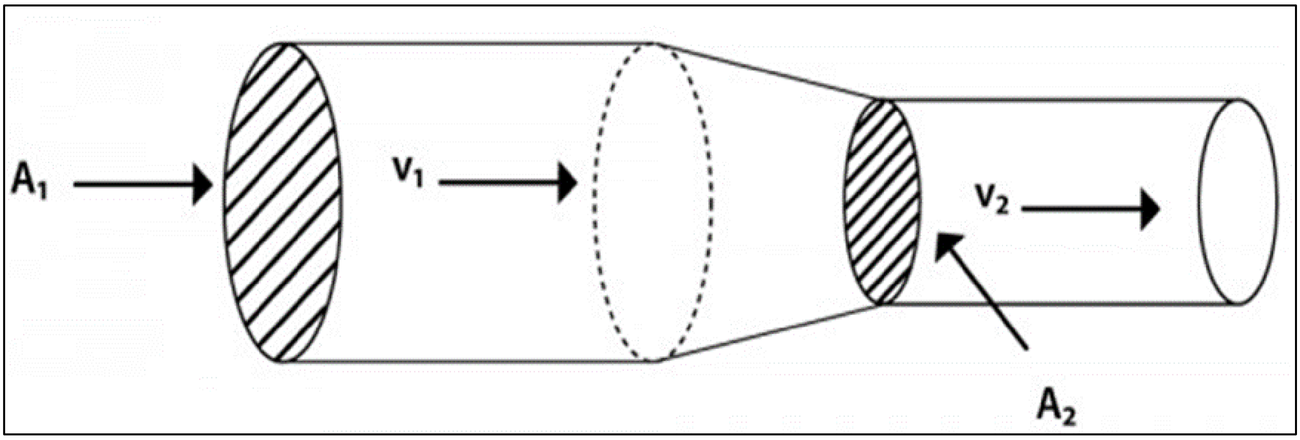


Figure 9: A representation of nozzle-throat cross section. [13]

The continuity equation can be expressed mathematically as shown in **eq. 1**:

$$\mathrm{V_1 \times A_1 = V_2 \times A_2 = Constant = Q} \tag{1}$$

The continuity equation for the nozzle can be expressed as shown in **eq. 2**

$$\mathrm{Q_1 = V_n \times A_n} \tag{2}$$

The continuity equation for the suction area can be expressed as shown in **eq. 3**

$$\mathrm{Q_3 = V_s \times A_s} \tag{3}$$

The continuity equation for the throat can be expressed as shown in **eq. 4**

$$\mathrm{Q_2 = Q_1 + Q_3 = V_t \times A_t} \tag{4}$$

V_n	average velocity of power fluid in the nozzle, (ft/s)
V_s	average velocity of power fluid in the suction area, (ft/s)
V_t	average velocity of power fluid in the throat, (ft/s)
A_n	nozzle area, (in^2)
A_s	suction area, (in^2)
A_t	throat area, (in^2)
Q_1	flow rate in the nozzle/ Injection rate (Stb/d)
Q_3	flow rate in the suction area/ Production rate (Stb/d)
Q_2	flow rate in the throat/,(Stb/d)

2.2.1.2 Bernoulli´s Principle

Bernoulli principle states that within a horizontal pipe that changes diameter, high velocity regions will be under less pressure than lower velocity regions. The velocity of a fluid increases when it flows through a narrow-restricted section as shown in Figure 7.
Fluid pressure speed up the fluid by pushing it through the small cross section (nozzle). This means that the pressure on the wider cross-section has to be larger than pressure on the smaller cross-section.

Bernoulli principle is derived from energy and rate of work equations and is as shown in equation 5. [13]

$$\left(\frac{P}{\rho} + \frac{1}{2} \times V^2 + gh\right)_2 = \left(\frac{P}{\rho} + \frac{1}{2} \times V^2 + gh\right)_1 \tag{5}$$

For incompressible fluids, density is constant along a streamline and therefore, **eq. 6** can be derived:

$$\mathrm{P} + \frac{1}{2} \times \rho \mathrm{V}^2 + \rho \mathrm{gh} = \text{constant} \tag{6}$$

The total head equation is obtained by dividing the previous equation by the specific weight ρg as shown in **eq. 7**

$$\frac{\mathrm{P}}{\rho \mathrm{g}} + \frac{1}{2\mathrm{g}} \times \mathrm{V}^2 + \mathrm{h} = \mathrm{H_t} = \text{constant} \tag{7}$$

H_t	total head , (ft)
ρ	density of the power fluid, (lb/ft^3)
A	cross section area, (in^2)
P	power fluid pressure, (psi)
G	gravitational constant, (ft/s^2)
V	velocity of power fluid, (ft/s)
M_q	mass flow rate, (lb/s)
h	Height, (ft)

The performance of the jet pump depends on four parameters based on Gosline and O'Brien theories and extensive laboratory tests and they are described as follow:

- Dimensionless flow ratio (M): Is a function of the flows in the pump and can be calculated as shown in **eq. 8**

$$M = \frac{Q_3}{Q_1} \tag{8}$$

- Dimensionless area ratio (R): Is a geometric characteristic of the jet pump and is the ratio of the nozzle area to the throat area as shown in **eq. 9**

$$R = \frac{A_n}{A_t} \tag{9}$$

- Dimensionless head ratio (H): The parameter H is the ratio of pressure increase experienced by the production fluid to the pressure loss suffered by the power fluid in the pump. The dimensionless head ratio is approximated by the static pressure in the pump and therefore, its calculated as shown in **eq. 10**

$$H = (P_2 - P_3)\,/(P_1 - P_2) \tag{10}$$

- Pump efficiency (E): The efficiency of the jet pump is defined as the ratio of the power gained by the produced fluid to the power lost by the power fluid as shown in **eq. 11**

$$Efficiency = E = \left(\frac{Q_3}{Q_1}\right) \times \left(\frac{P_2 - P_3}{P_1 - P_2}\right) * 100 \qquad (11)$$

M	dimensionless flow ratio, (-)
R	dimensionless area ratio, (-)
H	dimensionless head ratio, (-)
E	pump efficiency, (%)
A_n	nozzle area, (in^2)
A_t	throat area, (in^2)
Q_1	flow rate in the nozzle/ Injection rate rate (Stb/d)
Q_3	flow rate in the suction area/ Production rate, (Stb/d)
P_1	power fluid injection pressure, (Psi)
P_3	pump suction/intake pressure, (Psi)
P_2	pump discharge pressure, (Psi)

A plot of these equations showing dimensionless head ratio H versus dimensionless flow ratio M for different values of area ratio R. The respective efficiencies are also plotted as a function of M. A plot of dimensionless performance curve is shown in Figure 9.

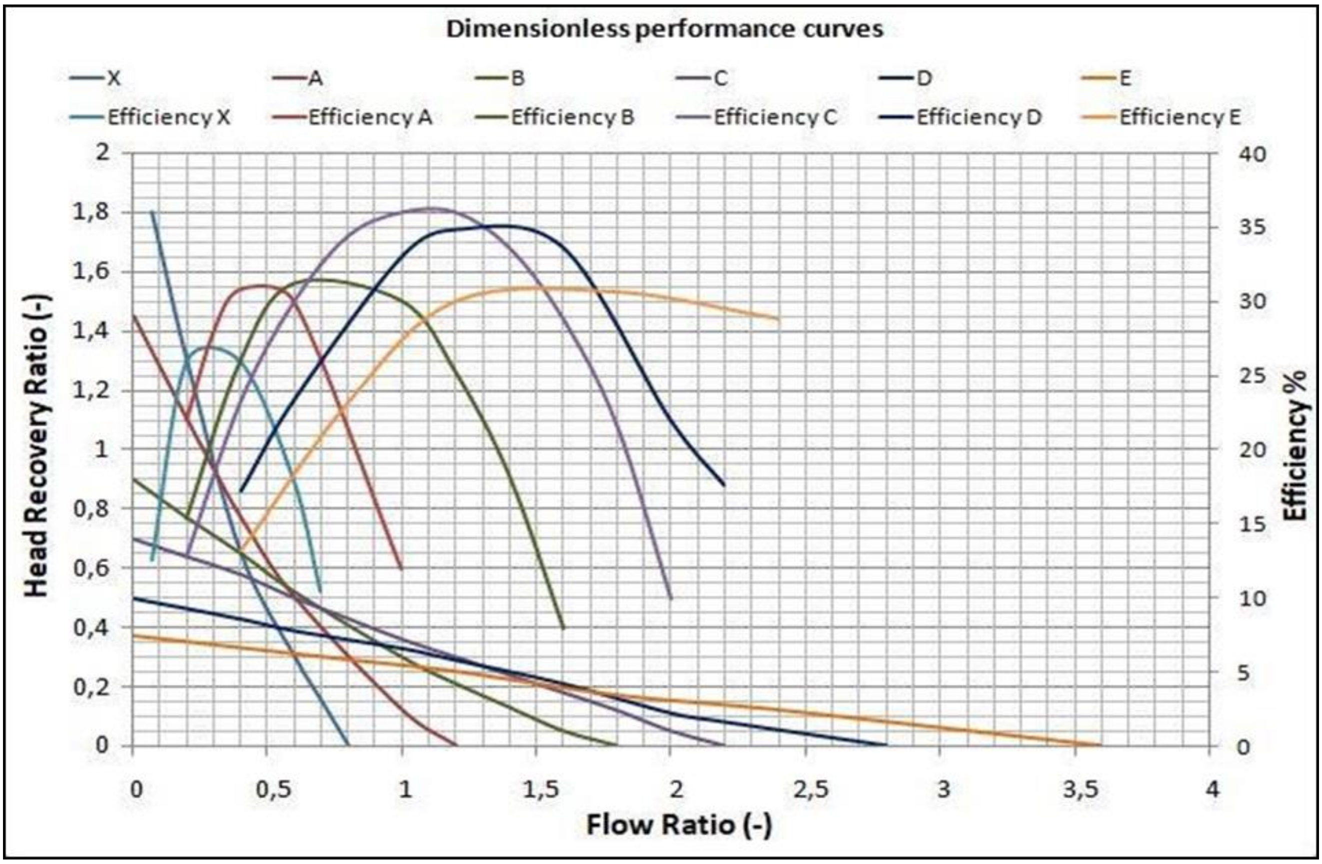


Figure 10: Dimensionless performance curves. [2]

The high head pump is typically employed in deep wells. The low head pump, on the other hand is employed in shallow wells. For example, the X ratio is for high lift and low production rates; the C ratio is for low lift and high relative production rates.

The maximum efficiency point for the combination A occurs at approximately M=0,5. This means that for every barrel of produced fluids, two barrels of power fluids must be injected in the well. These curves are dimensionless, so they are valid for jet pumps of the same area ratio regardless of nozzle number.

2.3 Nozzle-Throat Combination Selection

Selecting the appropriate nozzle-throat combination is one of the main challenges that engineers must face when planning a jet pump installation. Different possible combinations are available and therefore, companies use different tools to choose the optimum combination to be installed in the well. These tools are dependent on the jet pump manufacturer or provider. A jet pump must have a capacity that is sufficient to obtain the rate of production that the well is capable of producing.

At the same time, the required surface horsepower must be kept at a reasonable level. The balance of the process involves staying within the operating limitations for a particular installation. The most common limitations are power fluid injection pressure and/or rate, and space limitations (such as for offshore installations).
When designing a jet pump or any other artificial lift, the goal is also to reach the highest efficiency. Therefore, the designing steps that will follow considers the highest efficiency for each pump ratio as shown in table 6 and its equivalent dimensionless head ratio H, dimensionless flow ratio M and area ratio R.

Table 5: Jet pump dimensionless flow, head and area ratios at highest efficiency point. [2]

Ratio	M at max efficiency [-]	H at max efficiency [-]	Efficiency [%]	Area Ratio R [-]
X	0,38	0,74	28	0,48
A	0,47	0,66	31	0,38
B	0,75	0,42	31,8	0,30
C	1	0,32	32,2	0,24
D	1,3	0,25	32,5	0,18
E	1,9	0,17	33	0,15

Once the jet pump performance at highest efficiency are defined, the next step is to calculate surface injection rates and pressures, pump intake and discharge pressure for each of the ratios in order to define the nozzle size, or in other words, we define possible jet pump combinations. [2]

The first step is to calculate the power fluid flow rate using **eq. 12**:

$$Q_1 = Q_{wt} \times \left(\frac{Bo}{M}\right) \tag{12}$$

The produced flow rate is then defined using power fluid flow rate and flow ratio at highest efficiency as shown in **eq. 13**. Flow ratio M is the ratio of production flow rate to the power fluid rate.

$$Q_3 = Q\_1 \times M \tag{13}$$

The pump discharge and intake pressures are calculated as shown in **eq. 14** and **eq. 15**

$$P_3 = P_2 - H\left(\Delta P_{hydrostatic} - \Delta P_{friction}\right) \tag{14}$$

$$P_2 = P_{wh} + \Delta P_{hydrostatic} - \Delta P_{friction} \tag{15}$$

The next parameter to be calculated is the injection pressure and it is determined as shown in **eq. 16**

$$P_1 = P_{imax} + \left(\Delta P_{hydrostatic} - \Delta P_{friction}\right) \tag{16}$$

Once the previous parameters are defined, the nozzle area can be calculated using **eq. 17** below:

$$A_n = \frac{Q_1}{1214.5 \times \left(\frac{P_1 - P_3}{s_g}\right)^{0.5}} \tag{17}$$

P_1	power fluid pressure at the nozzle inlet, (psi)
P_2	discharge pressure in the annular space, (psi)
P_3	intake pressure, (psi)
P_{wh}	wellhead Pressure, (psi)
P_{imax}	maximum Injection pressure at the surface, (psi)
B_o	oil volume factor, (bbl/STB)
A_n	nozzle area, (in^2)
$\Delta P_{friction}$	pressure loss due to friction, (psi)
S_g	power fluid specific gravity, (-)
$\Delta P_{hydrostatic}$	hydrostatic pressure, (psi)
M	dimensionless flow rate, (-)
Q_1	injection flow rate, (Stb/d)
Q_3	production flow rate, (Stb/d)
Q_{wt}	well test flow rate, (Stb/d)

The performance of a jet pump depends on the size of the nozzle and the annular area between the throat and the jet stream of the nozzle. As the nozzle size and throat size increase, the flow capacity increases. Larger nozzle flow area allows for a greater nozzle flow and a larger annular area allows for larger volumes of oil, water and gas to pass into the throat for the same differential pressure as shown in Figure 11.

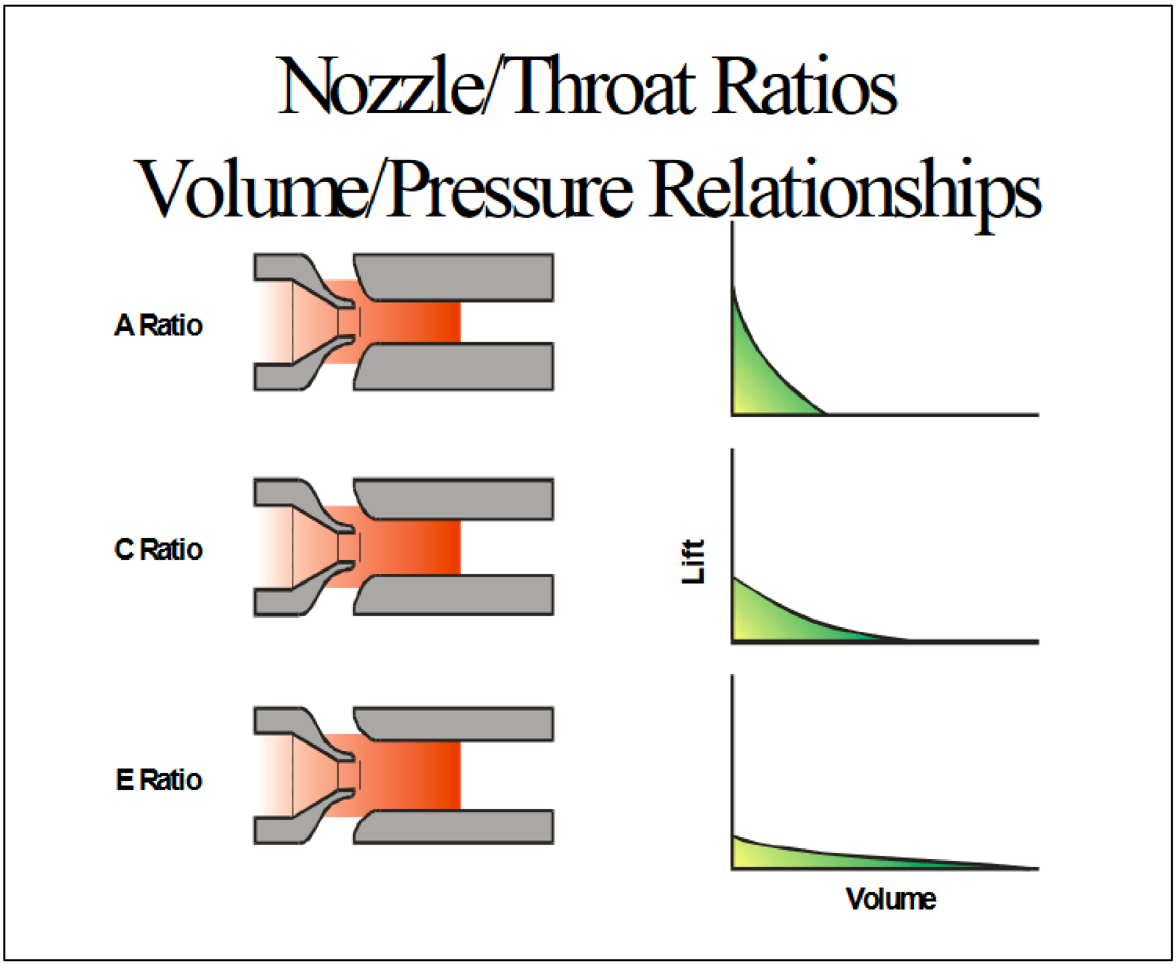


Figure 11: Volume/Pressure Relationships for Different Area Ratios. [5]

The ratio of the nozzle area to the throat area equals the area ratio. As the area ratio progressively decreases, then the throat has a greater annular area to allow the flow of produced fluids around the nozzle. Greater production rates are possible for a constant differential pressure, but less discharge head will develop due to the reduced volume of power fluid available to interact with the greater volume of produced fluid. [5]
The reverse is true for the larger ratio numbers. Lower production rates occur as the throat size decreases since the annular area available for fluid flow progressively decreases as shown in Figure 11.

However, an optimum size is usually selected within a horsepower constraint. The power required depends on the nozzle size, the operating pressure, the depth of the well, and especially the pump intake pressure.

2.4 Design Considerations

2.4.1 Cavitation

One of the major issues of jet pumps is their sensitivity to cavitation which has a high potential of occurring when the pump is not properly designed. Cavitation is a phenomenon in which rapid changes of pressure in a liquid lead to the formation of small vapor-filled cavities. Cavitation usually occurs in high velocity areas such as pumps and turbines.

Once vapor bubbles collapse against pump walls or impellers, they result in extensive erosion and reduction in pump efficiency. There are two types of cavitation: production cavitation and power fluid cavitation:

- Production cavitation is the result of a pressure drop in the produced fluid at the entrance of the throat and is due to a fluid rate that is too high for the flow area that is available. Based on the continuity equation, the higher the volume for a given flow area, the higher the velocity and therefore the lower the local static pressure. This results in the creation of vapor cavities or "bubbles" in the produced fluid stream. These bubbles are produced in sufficient quantity that they choke the flow as it enters the throat and no more production is possible. The damage to the jet pump occurs when these vapor bubbles collapse, usually against the wall of the throat. The rapid collapse of cavitation bubbles in a liquid typically produces shock waves. This type of cavitation can also occur in centrifugal pumps.

- Power fluid cavitation on the other hand occurs in the suction area between nozzle and throat and results from high velocities and the pressure drop of power fluid after exiting nozzle area. The pressure drop is due to the interaction of the high velocity power fluid stream with a produced fluid stream of insufficient volume and rate. Cavitation caused by power fluid may cause damage in the constant diameter section of the throat or in the diffuser as shown in Figure 12. [12]

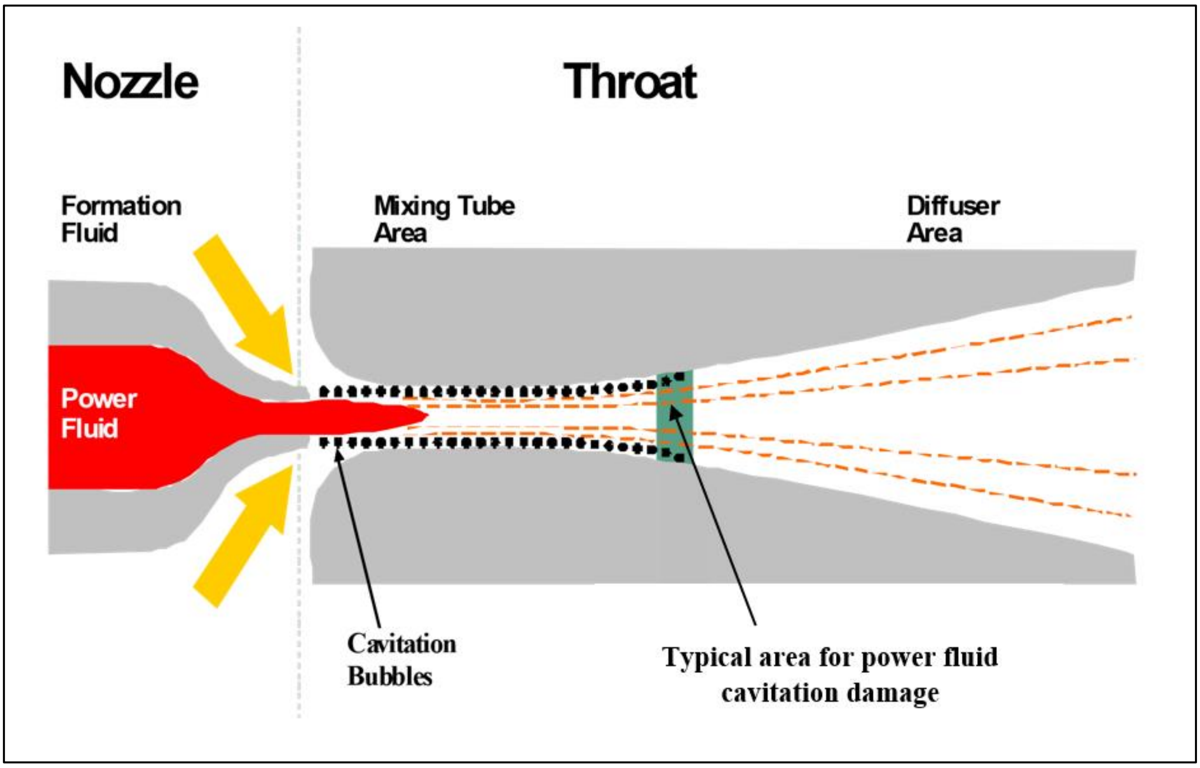


Figure 12: Schematic of cavitation bubbles and damage due to power fluid cavitation. [14]

The geometrical changes introduced to the perfectly shaped throat section as well as the diffuser and even minor changes in their diameters increases turbulence, friction and losses. Also, the change of nozzle throat area ratio will decrease jet pump efficiency and sometimes leads to equipment breakdown. In Figure 13 a typical nozzle cavitation damage is shown:

Figure 13: Damaged nozzle due to cavitation.

However, to prevent cavitation, the production rate should be lower than the cavitation rate and the pressure should be high enough to avoid cavitation. Net positive suction head (NPSH) of the pump is also a key parameter in the analysis of cavitation and it is composed of two main parameters: available and required net positive suction heads.

Available net positive suction head (NPSHa) is the absolute pressure head minus the vapor pressure of a liquid at the suction port of the pump. Required net positive suction head (NPSHr) is the minimum pressure required at the suction port of the pump and is given by the pump manufacturer. The available net positive suction Head (NPSHa) should be larger than the required net positive suction head (NPSHr) for the pump system to operate without cavitation.

When designing a jet pump, a cavitation flow rate can be calculated to prevent the pump from working in cavitation conditions.

Cavitation index I_c should be defined at first using **eq. 21**:

$$I_c = \frac{P_1 - P_b}{P_1 - P_2} \tag{21}$$

Cavitation rate is then calculated as shown in **eq. 22**

$$\mathrm{Q_c} = \mathrm{R_c} \times \mathrm{Q_1} \tag{22}$$

Where R_c is the cavitation flow ratio and is defined by **eq. 23**

$$\mathrm{R_c} = \left(\frac{1-\mathrm{HP}}{\mathrm{HP}}\right) \times \left(\frac{\mathrm{P_3}}{\mathrm{I_c} \times (\mathrm{P_1} - \mathrm{P_3}) + \mathrm{P_3}}\right)^{0.5} \tag{23}$$

HP is the required horse power and define by **eq.24**

$$\mathrm{HP} = \mathrm{q_1} \times \mathrm{P_1} \times \mathrm{E} \times \mathrm{C} \tag{24}$$

P_1	power fluid pressure at the nozzle inlet, (psi)
P_2	discharge pressure in the annular space, (psi)
P_3	Intake pressure, (psi)
P_b	bubble point pressure, (psi)
I_c	cavitation index, (-)
R_c	cavitation ratio, (-)
Q_c	cavitation flowrate, (Stb/d)
HP	horsepower, (hp)
E	efficiency, (%)
C	conversion factor, (-)

2.4.2 Erosion Damage

Erosion refers to the loss of material on the equipment surface, tubing for example, due to the mixture of production fluid and solid particles at high velocities. We differentiate between three types of erosion: sand or solid particle erosion, liquid droplet erosion and erosion-corrosion.

The design phase is important to manufacture all parts (exposed to high velocity flow) out of abrasive resistant materials, such as tungsten carbide. These parts are mainly nozzle, and throat. By using abrasive resistant materials, the demand for retrieving the pump for maintenance is reduced.

2.4.3 Emulsion

An emulsion is a dispersion of one liquid as droplets in another immiscible liquid, which is a continuous or external phase. The phase that is present in the form of droplets is the dispersed or internal phase. Crude oil emulsions form when oil and water (brine) are in contact with each other when there is sufficient mixing, and when an emulsifying agent or emulsifier is present.

The stability of the emulsion depends on the types and the amount of surface-active agents, which commonly act as emulsifying agent or emulsifiers. Emulsifying agents form interfacial films around the droplets of the dispersed phase and create a barrier that prevents coalescence of the droplets

At low temperature and presence of asphaltene and waxes, water forms a stable emulsion with crude oil and cannot be removed. In this case, emulsion treating methods must be used. Emulsion treating processes require a combination of the following: use of demulsifier, more settling time, heat, and electrostatic coalescing.

Besides emulsion, erosion and cavitation issues, limitations of surface units should also be taken into consideration. Due to surface pump limitations, we should select a jet pump combination with equivalent pressure and rate that our surface pump can deliver. Therefore, the combination that ensures the highest possible production rate is not necessarily the optimum combination. [15]

2.5 Advantages and Limitations

2.5.1 Advantages

There are numerous advantages to hydraulic jet pumping systems compared to sucker-rod, ESP or gas-lift systems. One major advantage is that it will operate over a wide range of well conditions such as deviated well setting depths of as much as 20,000 feet and production rates of 35,000 bpd and they are able to operate for extended periods of time without the need for intervention. Production rates can be varied by simply adjusting the power fluid injection rate.

Power fluid can be pumped down-hole through a standard circulation or in a reverse circulation completion. The benefit of reverse circulation is the ability to run the downhole pump into and out of the well with a reduced downtime and no need for pulling tubing, cable, or rods.

Chemicals can be added to the power fluid to control corrosion problems, paraffin and scaling problems. Hot water can also be used in a reverse circulation to control paraffin problems. However, maintenance costs are low compared to other pumps and are easily and quickly retrieved and replaced when maintenance is required. Jet pumps are also suitable for low gravity crude oils and capable of handling high gas liquid ratios. Jet pumps can also be used in wells with high deviation angles. [16]

2.5.2 Limitations

Power fluid injection rates are typically twice the production rate and high injection rates of the power fluid requires high surface facility investment. Produced fluids treatment as well as conditioning of the power fluid are mandatory to have a supply of clean power fluid.

Sand or other solid particles in the power fluid must be removed as they can damage the surface power fluid pump as well as the nozzle in the downhole pump section. Also, high pressure surface lines and high injection rates of as much as 5000 psi might possibly be a safety hazard and should be taken into consideration. [16]

However, other artificial lift methods are available and they can deliver higher production rates. More details concerning other artificial lift methods are listed in Table 7.

Table 6: A general comparison of different artificial lift systems. [16]

	Jet Pump	Gas Lift	Sucker Rod	ESP
Common applications	-Moderate-low GLR wells -Mod–low PI wells -Deviated wells -Alternative to gas lift	-Offshore and Onshore wells -deep wells -deviated wells -moderately productive wells	-Onshore wells -Low PI wells	-High rate, low GLR wells -High WOR wells -Alternative to gas lift
Production rate	100-30 000 bpd	10- 50 000 bpd	2-1500 bpd	+/- 120000 bpd
Efficiency	Fair to poor, maximum efficiency for ideal case is 30%	typically 20 to 30%	Excellent total system efficiency. Typically 50 to 60%	Typically total system efficiency is about 50% for high rate wells but for rates <1000bpd, efficiency is <40%.

Capital Cost	Relatively low to moderate. Cost increases with higher horsepower. Requires surface treating and high pressure pumping equipment.	Well gas lift equipment costs are low, but compression cost and gas distribution system may be high	Low to moderate. Cost increase with depth and larger surface units.	High cost for power generation and cabling. Relatively low capital cost if electric power available
Operating Cost	High power cost to pump power fluid. Low pump maintenance cost with properly sized throat and nozzle. No moving parts in pump, simple maintenance procedures.	Low, Gas lift systems have a very low OPEX	Low for shallow to medium depth (<7000ft) and low rates (<400bpd)	Moderate to high. Costly interventions are required to change out conventional ESP completion. ESP workover is generally costly. However, productivity and improved run life can offset these costs.

3 Jet Pump Performance Assessment

When designing an artificial lift system, several challenges are raised such as matching the well productivity, the reservoir characteristics, artificial lift capabilities and surface facilities. Among the most important factors to consider are reservoir changing pressure and well productivity. However, the jet pump production system offers a great flexibility to changing well conditions.

Surface operating parameters such as pressure and rate can be adjusted with little lost production. Changing the jet pump size to adapt to well conditions and increase production can be made at low costs compared to other artificial lift system and do not require a lot of time due to the jet pump flexibility and simple configuration.

The first jet pump was installed in Tunisia in 2015 and since jet pumps are not a common artificial lift system in Tunisia, challenges raised tremendously. Despite the pump´s simple mechanism, engineers had to face a high number of failures in a short period of time. A detailed investigation of pump failures and the performance of the pump was necessary to develop an improvement plan.

A first sight on the problem shows that surface equipment failures, especially the surface pump, and downhole issues such as plugged nozzle and jet pump damage have occurred multiple times at regular intervals.

3.1 Jet Pump Performance Overview

A workshop was done in Tunisia in 2016 to analyze the failure that occurred in an oil field operating mainly with jet pumps. The study showed that the jet pumps had failed more than 40 times in less than half a year.

Liquid foaming, old equipment and inappropriate jet pump design are all likely have contributed to this huge number of failures in a short period of time. These failures are resulting in approx. 7600 bbl./month of deferred production, as shown in Figure 14. The blue curve and the red curve show consecutively, the production deferment and shut in hours from the month of January 2016 until June 2016.

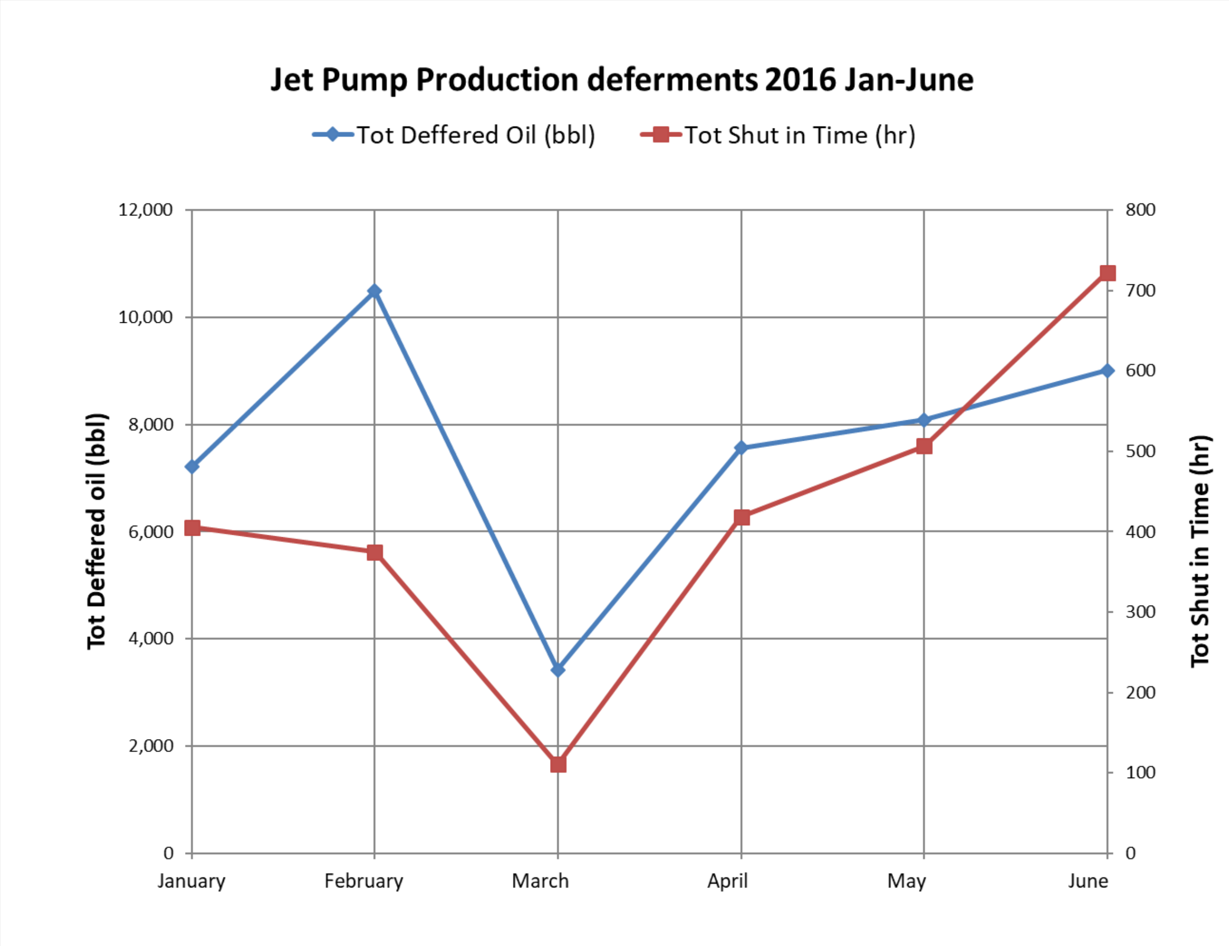


Figure 14: Jet pump production deferments from January to June 2016.

48% of oil deferment was mostly a planned deferment for production optimization (changing of the jet pump combination, well testing, completion optimization…)

30% of oil deferment was a result of the surface pump failure (leakage, piston failure, maintenance…)

22% of oil deferment was a result of several other issues such as foaming, plugged nozzle, surface unit’s leakage, and electrical engine failure as shown in Figure 15.

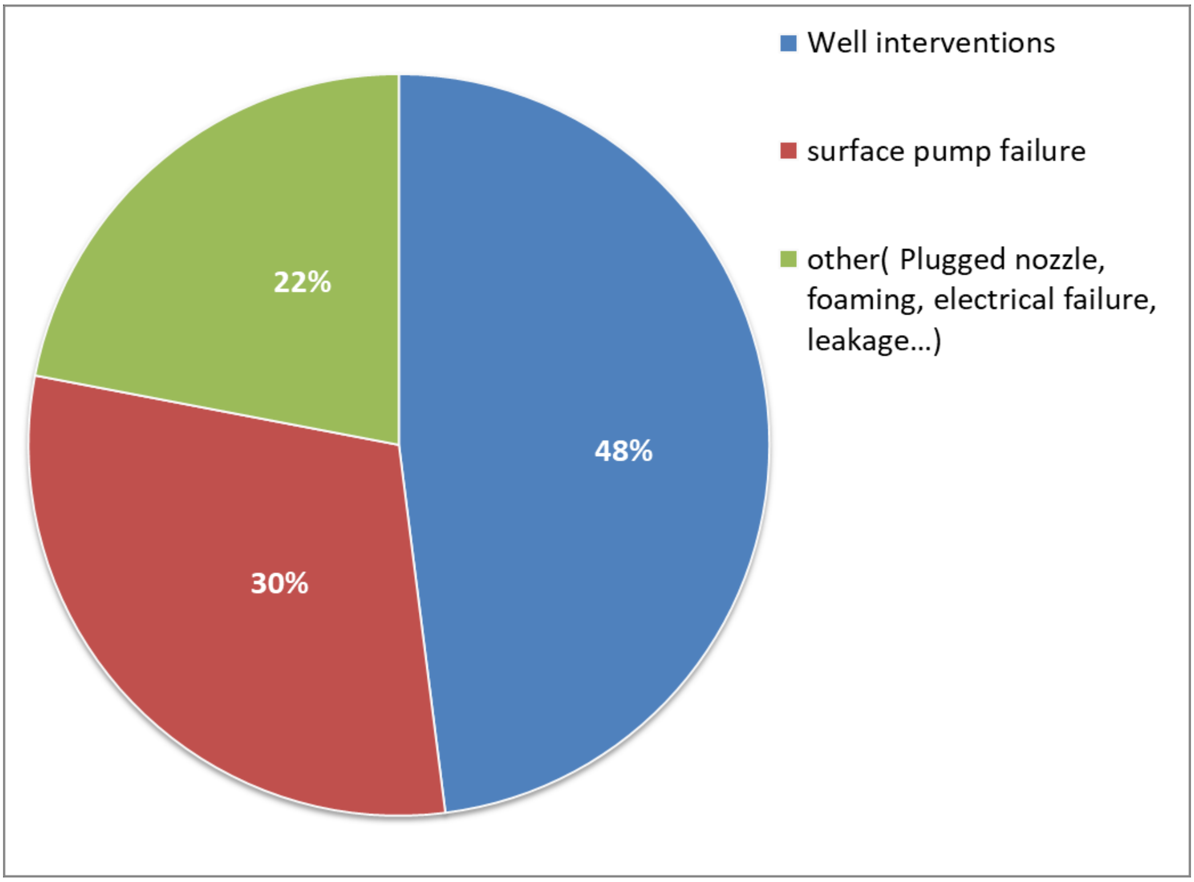


Figure 15: Jet pump failure causes.

3.2 Fault Tree of Jet Pump

Fault tree analysis (FTA) is an important method in reliability engineering to understand how systems or a mechanism can fail. A fault tree is established based on collecting its main failure modes, starting with a potential undesirable accident, also called a top event, and then determining its causes. Then the qualitative and quantitative analysis of a fault tree is discussed. The analysis proceeds by determining how the top event can be caused by individual or combined lower level failures or events.

A fault tree was used in a workshop in 2016 to investigate the failures of the jet pumps and their origins.

Fault trees that are discussed and analyzed in the subsequent section to follow as listed below are:

- Vibration fault tree
- Surface pump fault tree
- Diesel engine fault tree

3.2.1 Vibration Fault Tree

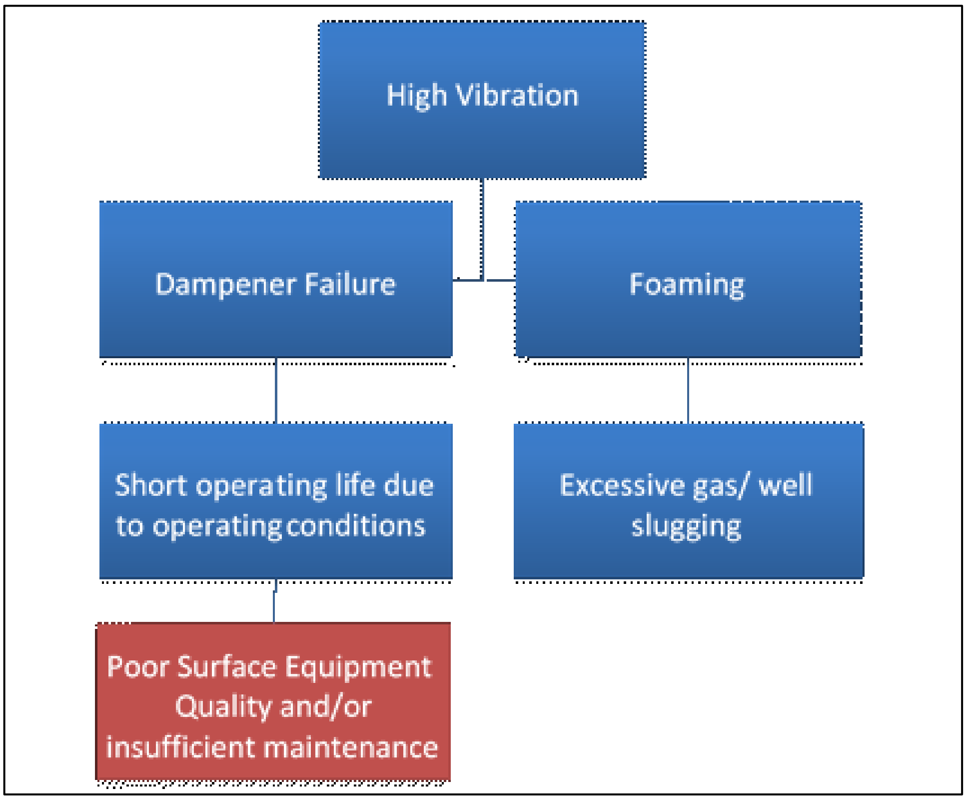


Figure 16: Vibration fault tree.

As shown in Figure 16, low maintenance frequency and poor surface equipment quality have contributed to the resulted failure. Continuous vibration with unsupported or improperly supported discharge line of the surface pump and poor welding quality has contributed to the dampener failure and the broken discharge line welds as shown in Figure 17.

A pressure dampener, also called pulsation dampener, is a spherical shaped vessel widely used in various fluid power systems to reduce vibration induced by pumps and are installed at the inlet or outlet of the pump. The vibration induced by the pump in fluid systems may be severe enough to cause damage of equipment if a dampener is not installed.

High vibration and excessive agitation have contributed in air bubble creation in the power fluid as well as produced fluid which leads to foaming formation. Foaming is an undesirable phenomenon in pumps, turbine and hydraulic systems in general as it may lead to misinterpretation of fluid levels and inappropriate power fluid and produced fluid separation.

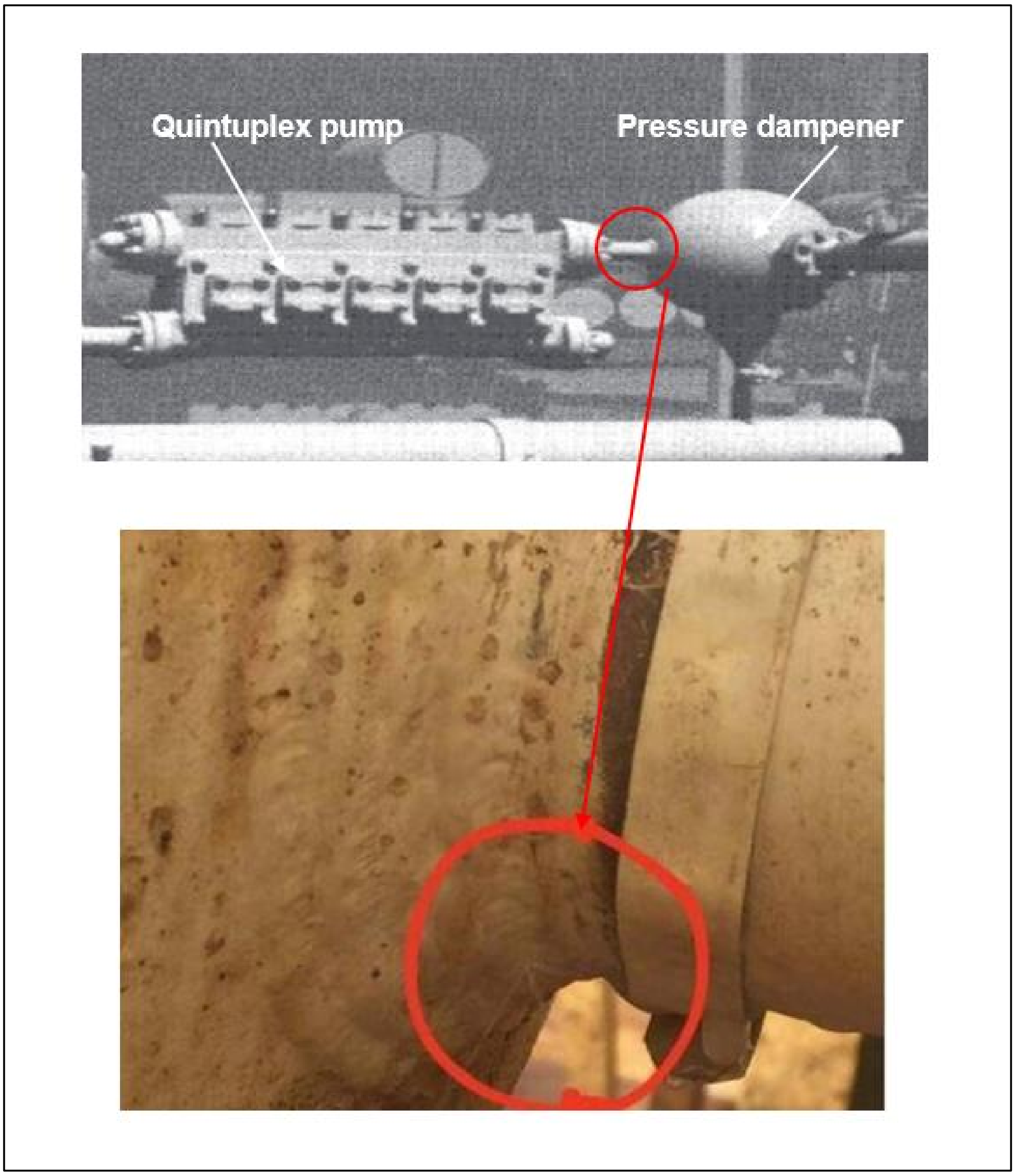


Figure 17: Broken discharge line and dampener.

3.2.2 **Surface Pump Fault Tree**

Surface pump failures are one of the major causes for jet pump failure and decreased performance. The corresponding fault tree can be divided into two main fault trees as shown in Figures 18 and 19:

- Surface pump suction pressure
- Surface pump discharge pressure

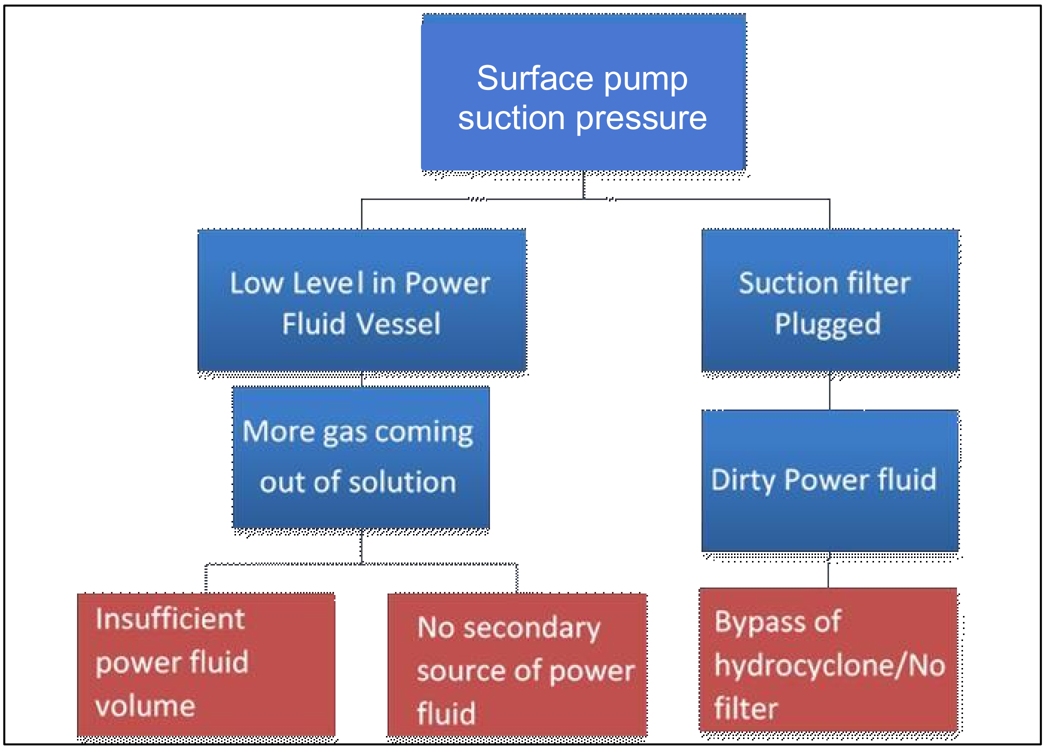


Figure 18: Surface pump suction pressure fault tree.

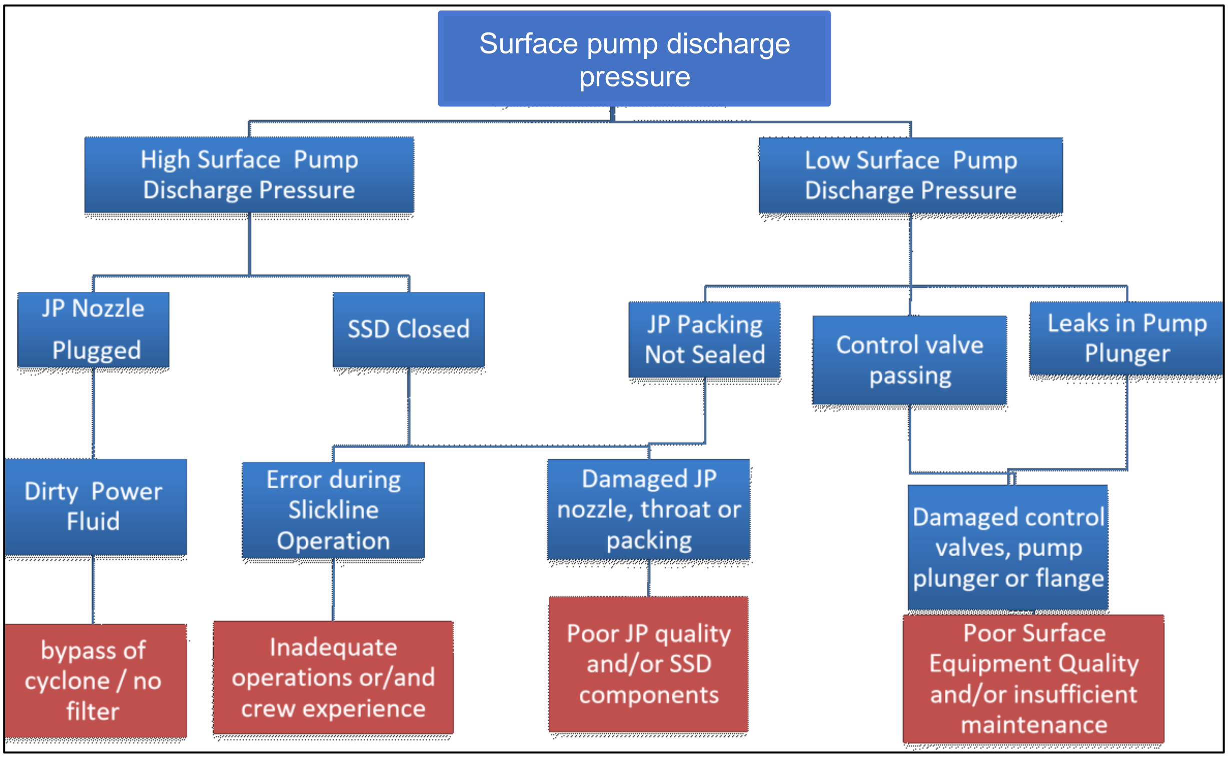


Figure 19: Surface pump discharge pressure fault tree.

Poor quality of the surface equipment, inadequate operations and crew experience and insufficient maintenance has resulted in severe failures in the surface pump and power fluid circuits.

As shown in Figures 18 and 19, plugged nozzle, valves or power fluid circuit is one of the major issues affecting the pump performance and causing major pressure changes. Paraffin build-up or obstruction (solid particles and other impurities) located in the power fluid circuit, or in the flow line accumulates through time and prevents the power fluid from entering the pump. In this case, the operating pressure increases slowly while power fluid rate remains constant or gradual reduced. If proper solutions are not applied, the power fluid circuit and the pump inlet and outlet will be fully plugged, pressure will increase tremendously while power fluid flow will essentially stop.

Leak in power fluid surface or downhole circuits is another major problem. Surface and subsurface flowlines and tubing should be checked for leaks and repaired as needed as well as the surface pump plungers. Sudden decrease in operating pressure with a constant power fluid rate is one of the indicators of the occurrence of a leak in one of the power fluid circuits and therefore, maintenance and continuous checking for leaks is mandatory to avoid sudden operating pressure changes and production deferment.

3.2.2.1 Surface Pump Stuffing Box Failures

During an asset inspection done in April 2018 major issues were found concerning the surface pump. The stuffing box adjustment nut was broken completely which is related to how the plungers packing are tightened.
Figure 20 shows a standard nut bushing gland assembly. Both components must be perfectly fitted and tightened.

Figure 20: A standard nut and bushing glands assembly.

When the packing assembly does not fully compress the adjustment nut, the bushing gland can be pulled back and forth hammering the bushing continuously and eventually weakening the adjusting near the thread relief resulting in failure as shown in Figure 21. In order to avoid such failure, field engineers should ensure that the stuffing box is adjusted correctly which includes cleaning and lubricating the threads before re-assembling.

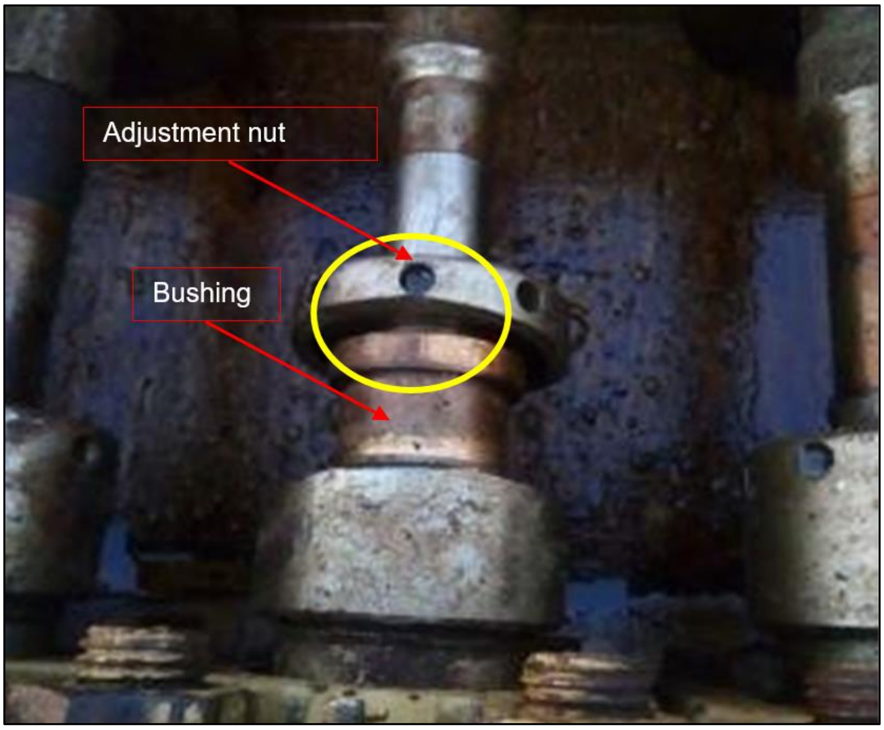


Figure 21: Broken stuffing box.

Washed out of stuffing box seal is another issue facing surface pump. The root cause is the use of an incorrect stuffing box adaptor ring, which caused a catastrophic wash out as shown in Figure 22. The use of improper torque values can cause threads to become loose over time, which can result in stress fatigue and major equipment damage.

Figure 22: Washed out stuffing box.

Stuffing box split failure is another issue that should be taken into consideration. This is not a common failure and the force required to split a stuffing box as shown in Figure 23, leads us to a potential misalignment issue via the not properly tightened stuffing box retainers as indicated in the previous failures.

Figure 23: Stuffing box split due to misalignment issue.

Crosshead, shown in Figure 24, connects the pump piston to the crank shaft through a connection rod and is responsible of transmitting the rotating movement of the crank to a reciprocating movement of the pump piston. Misalignment in the crosshead and cross head housing is generally caused by excessive heat or lack of lubrication. This may lead to the breakage of the cross head and bending of the pump piston and connection road.

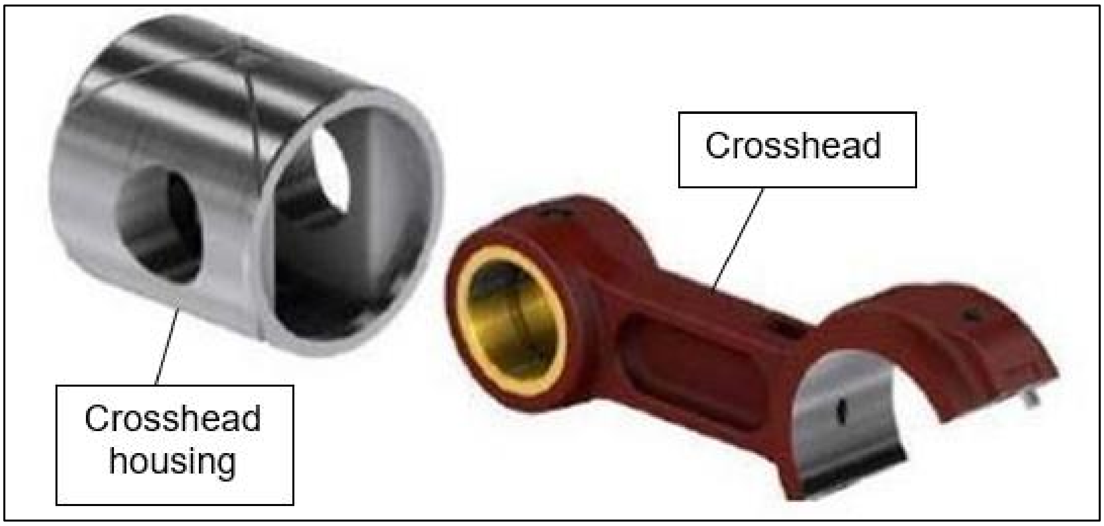


Figure 24: Surface pump cross heads.

3.2.2.2 Plunger Failures

Plunger packing life is shortened due to various reasons such as improper packing of plungers and adjusting of stuffing box nuts and improperly tightened retainers. Improper plunger lubrication is another root cause for plunger failures. Oil being used doesn't have the proper stickiness qualities. Also, inadequate plunger lubrication volumes are used in inappropriate areas. A typical pump plunger and lubricating tubing configuration are shown in Figure 25.

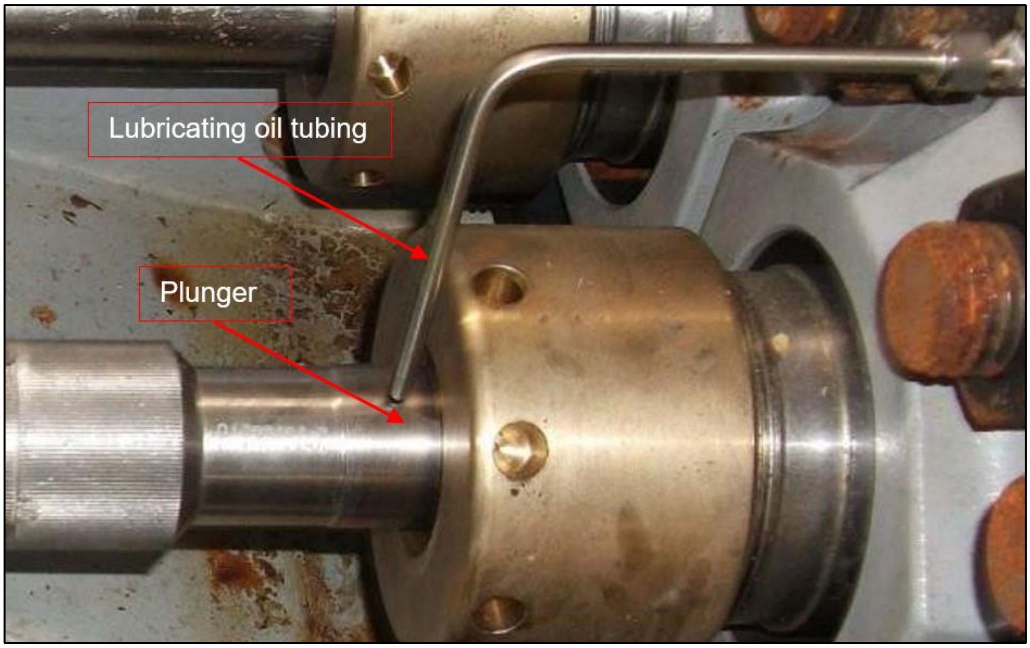


Figure 25: Surface pump plunger and perfectly located lubrication system.

3.2.3 Diesel Engine Fault Tree

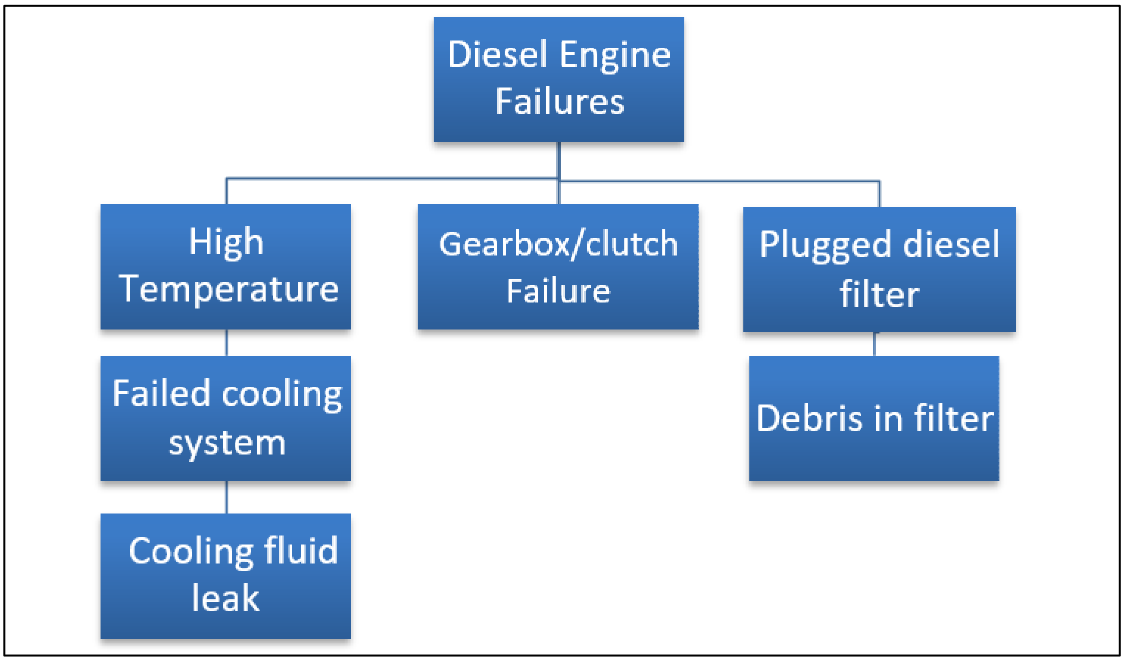


Figure 26: Diesel engine fault tree.

High temperature, gearbox failure and plugged diesel engine circuit have contributed to the diesel engine failures which provides power to run the surface pump. As a result, the surface pump failed several times to inject the power fluid downhole and thus, resulted in production deferment. These failures were a result of the lack of cooling fluid due to leakage in the system or the accumulation of debris as shown in Figure 26.

Despite all the failures reported, production deferment in Tunisia due to well intervention was reduced tremendously since 2016. Jet pump combinations were rarely changed, and jet pumps were installed in the well using free style method. This resulted in time and money savings compared to slick line interventions and reduced oil deferment.

Maintenance is done more frequently to prevent surface pump damage which also resulted in production deferment reduction. Corrosion inhibitors injection and power fluid quality control reduced nozzle plugging and corrosion. No plugged nozzle or corroded equipment was retrieved from wells until today.
Despite all the improvement since the installation of the first jet pump, more improvement can be done when it comes to the maintenance and equipment quality.

4 Jet Pump Performance Optimization

4.1 Integrated Assets Modelling (IAM) and Prosper

Integrated Asset Modeling (IAM) is the term used in the oil industry for computer modeling of both the subsurface and the surface elements of a field. Oil and gas assets usually consist of several interconnected systems and processes, which have always been modeled separately.

IAM proved to be an effective way of oilfields modeling that captures the interaction between the standalone production models. It is also, a reliable method to combine reservoir, production and surface engineering modeling into a single software platform. This allows the simulation of entire assets taking into consideration the interaction between the surface and the subsurface pressures, mixing of different fluids, accounting for facilities constraints and identification of system limitations. In this way, optimum artificial lift methods can be implemented to meet production targets or to investigate production optimization possibilities.

The target field consists of three concessions and 25 wells: one naturally flowing well, 18 wells operated by gas lift wells, 2 wells equipped with electrical submersible pumps (ESP) and 4 wells using jet pumps. The three concessions are connected through a trunk line, which goes to a central processing facility where fluids are separated, and water is disposed.

PROSPER, **Pro**duction and **s**ystems **Per**formance, is the industry standard single well performance design and optimization software, it can model most types of well completions and artificial lifting methods. Major operators worldwide use the software and it allows building well models with the ability to address:

- Fluid thermodynamics (PVT)
- Multiphase VLP correlations
- IPR models

A critical part of the development of a jet pump modelling software was the establishment of the loss factors for the nozzle, suction, throat and diffuser as listed below:

- Nozzle loss coefficient K_n
- Suction loss coefficient K_s
- Throat loss coefficient K_t
- Diffuser loss coefficient K_d

Having values for those loss factors is critical as no calculations can be made without them. The main scope was to find appropriate loss coefficient factors that we can apply for Prosper to finalize an already existing integrated asset model as it was not possibility to connect the GAP model to the 3rd party JEMS software which is originally used to model jet pumps.

JEMS was developed in 1980 and the loss coefficients are integrated in the software. Therefore, the use of JEMS does not require any knowledge of these loss coefficients. Additionally, these loss coefficients have been found after several laboratory testing and they are considered as confidential data and every other program in use today other than JEMS just estimates them.

Theoretical and empirical approaches were applied in this thesis in order to estimate these coefficients and are discussed in details in the next chapter.

4.2 Jet Pump Design

"Jet Evaluation Modeling Software" or JEMS is a software used to model the performance of a specified jet pump nozzle/throat combination in order to deliver the desired production rate under specified conditions and parameters. The software was originally known as "4.1" and was continuously developed and updated resulting in a more sophisticated and accurate program than any other jet pump analysis program that has ever been created up to date.
Several input data are required to model a jet pump using JEMS and they are as listed below:

- Perforation depth (ft): the true vertical depth to the midpoint of the perforation interval.

- Pump true vertical depth TVD (ft): the true vertical depth to the location of the pump bottom-hole assembly BHA. If the pump vertical depth is less or greater than the perforation depth, the intake pressure is corrected from the value of the flowing bottom hole pressure at the perforation depth to the value of the pump intake pressure at the pump vertical depth. The correction is based on the average liquid gradient of the produced well fluids.

- Pump installation:
  - The parallel free system: Includes one string of tubing that conducts the power fluid supply down to the pump and bottom-hole assembly, and one string of tubing that returns the well-produced fluids and the power fluid mix to the surface. The program default value for this type of installation is (2).
  - The casing free system: Includes one string of tubing that conducts the power fluid supply down to the pump and bottom-hole assembly and returns the well-produced fluids and power fluid through the casing-tubing annulus. The program default value for this type of installation is (1).
  - Reverse flow or reverse circulation systems are special design cases that do not have a specific pump installation type number assigned. In reverse flow system power fluid goes down the casing-tubing annulus or a separate power fluid supply string and the produced fluids and power fluid mix returns up through a tubing string. Usually a parallel free installation (1) is selected to model reverse flow systems.

- Casing/Tubing inside diameter ID (in): The actual diameter is the diameter through which the fluid flows, thus installations with iron sulfide, scale, or paraffin deposits may have a diameter less than the original steel size diameter.

- Tubing length (ft): Refers to the measured depth (MD) of the power fluid supply tubing string including the length of the pump bottom-hole assembly.

- Pipe condition: There are three numerical values that represents the average pipe condition. New pipe referred to with (1), average pipe referred to with (2) and old pipe referred to with (3). This input specifies the relative roughness used in friction pressure loss calculations.

Beside the well completion data, several well test data are required to model a jet pump using JEMS and they are as listed below:

- Oil gravity (API): The value entered should be for the produced oil at stock tank conditions.

- Water cut (%): The water cut is the volume percent of water of the total fluid mixture. The water cut is measured at test conditions using metering separator to meter the oil and water volumes at stock tank conditions.

- Water specific gravity: The program default value for the water specific gravity is 1,05. However, if a more accurate or measured value of water specific gravity is available, the default value can be changed.

- Gas oil ratio GOR (scf/STB): This parameter can be determined by calculating the total gas produced per barrel of oil produced at stock tank conditions. GOR is a major factor in determining the required throat and nozzle sizes, the friction pressure drop of the produced fluids and is used to determine all the correlation derived fluid properties. The accuracy of GOR determines how well JEMS models the well performance.

- Gas specific gravity: The software accepts values between 0 and 2,0. The default value in JEMS is 0,80. However, the presence of H_2S or CO_2 will lead to higher values. The gas specific gravity is a major factor used to determine correlation derived fluid properties.

- Separator pressure (psig): The program has a default value for separator pressure of 30 psig. The operating pressure of the test or metering separator used to measure the oil, water, and gas volumes. The separator pressure is used to correct the GOR value from the test condition to the surface condition at stock tank.

- Well static bottom-hole pressure BHP (psig): The static bottom-hole pressure is the pressure the oil reservoir builds up to and stabilize at. It is one of the values used to construct an inflow performance relationship (IPR) curve for the well. JEMS assumes that the well production or flowing bottom-hole pressure is measured at the midpoint of the perforation interval. If the static pressure is not known, value of zero can be used since JEMS can proceed calculations without knowing the static bottom-hole pressure.

- Well flowing BHP (psig): The steady state producing or flowing bottom-hole pressure that corresponds to the production rate of the well. The IPR curve and its ability to model changes in the production rates for different producing bottom hole pressures is dependent on the accuracy of the reported rates and pressure.

- Well test flow rate (bpd): The total liquid flow rate of the oil and water as reported by production test on a 24 hour basis. If the actual rate is not reported after a 24 hour test period, then an estimated value of what the rate would be after 24 hour tested in entered. The multi-phase flow correlations are based on the volumes produced during a 24 hour period. Additionally, JEMS will correct the oil and water rates reported at test conditions to the oil and water volumes at stock tank conditions.

- Well head temperature (deg. F°): The temperature at of the produced well fluids and power fluid mix at the well head. The well head temperature is used to calculate the pressure gradient and it effects the calculation of the jet pump discharge pressure. If the well head pressure cannot be measured, the program has a default temperature of 100 degree F°.

- Bottom-hole temperature (deg. F°): The temperature measured at the perforation depth. The default value, if the temperature is not known, is the well head temperature plus one degree F° for each 100 feet to the perforation depth. The bottom-hole temperature effects the calculation of the jet pump discharge pressure since it affects the fluids properties.

- Power fluid: Two options of power fluids are available in JEMS. Oil is referred to with default value (1) and number (2) for water.

- Power fluid specific gravity: API value should be entered if the power fluid is an oil. If power fluid is water, specific gravity should be used.

Figure 27 shows JEMS input data interface that were listed and explained previously.

Figure 27: JEMS Input data interface.

Once input data is introduced to the software, the maximum and minimum power fluid injection pressure should be added to the software as shown in Figure 28 in order to have an appropriate combination for our production target and surface pump properties.

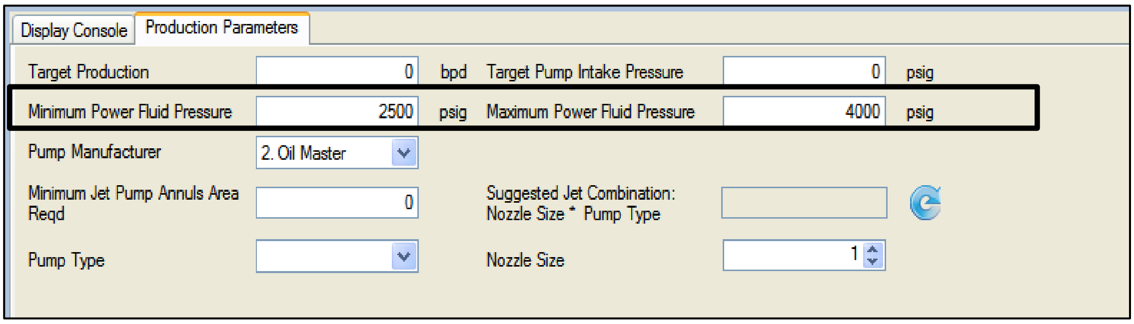


Figure 28: JEMS production parameter interface.

JEMS suggests more than one possible jet pump performance, taking into consideration only injection rate and pressure of the surface pump and target production rate. The next step is to check the cavitation rate of each solution to avoid cavitation. In addition to the JEMS output report, we can also visualize the graph of intake pressures versus production rates for different surface injection pressures.

4.2.1 **JEMS Design Example**

In this chapter we will show a well design using JEMS and compare the software result with well test data. Available data from a well located in one of South Tunisia assets is introduced to JEMS input interface as shown in Figure 29:

```
                    W E L L   D A T A   S U M M A R Y
                    =================================

     Customer     : OMV                                Date  : 09/03/2017
     Field & Well : Nada-1
     Location     : Anaguid East - South Tunisia       Run ID: 094148-02

 1. Perforation Depth       (ft) :  8530.0  13.Producing GOR  (scf/STB) :  650.0
 2. Pump Vertical Depth     (ft) :  6927.0  14.Gas Sp. Gravity (air=1.) :  0.957
 3. Pump Instl  (1) Standard Flow           15.Separator Press    (psig) :   75.0
    (2)Rev. Flow (3)Parallel Flow:  1       16.Well Static BHP    (psig) : 3350.0
 4. Casing ID               (in) :  6.094   17.PI            (STB/d/psi) :   1.02
 5. Tubing OD               (in) :  3.500   18.N/A when PI is used
 6. Tubing ID               (in) :  2.992   19.Well Head Temp  (deg. F) :  100.0
 7. Return Tubing ID        (in) :  N/A     20.Bottom Hole Temp(deg. F) :  180.0
 8. Tubing Length           (ft) :  7065.0  21.Bypass Gas? (1)yes (2)no :  2
 9. Pipe Cond (1)new(2)avg(3)old :  1       22.Power Fld (1)oil (2)water:  2
10.Oil Gravity             (API) : 39.000   23.Power Fld API/Sp. Gravity:  1.200
11.Water Cut                 (%) :  45.00   24.Bubble Point Press(psia) : 2134.6
12.Water Specific Gravity        :  1.200   25.Well Head Press    (psig) :  180.0
```

Figure 29: JEMS input data.

The next step is to define the maximum and the minimum pressure of the surface pump as well as a production target rate. As a result, the program suggests a jet pump combination and defines the output parameters as shown in Figure 30:

```
Oilmaster 11A  Jet Pump Performance Summary for User Specified Production
rate of  1200. STB/d

Production Rate        =   1200. STB/d
Injection Pressure     =   2506. psig
Injection Rate         =   2147. bpd
Horsepower to Jet Pump =    105. hp
Pump Intake Pressure   =   1487. psig
Discharge Pressure     =   3005. psig
Cavitation Rate        =   1444. STB/d
```

Figure 30: 11A combination performance summary for a specific production target (1200 STB/d).

As shown in Figure 29, JEMS suggested a jet combination which is in our case a combination 11A. Referring to Table 1, we can directly determine the nozzle and the throat areas:

- The nozzle size number is 11, it corresponds to 0,0269 in^2 of nozzle area
- The throat size number is 11, it corresponds to 0,0707 in^2 of throat area

JEMS also defines the pump operating parameters as listed below:

- Production rate: The rate set as 'target rate' in the input data.
- Injection pressure: The pressure obtained at the discharge of the surface pump and, it is the pressure with which the power fluid goes to the well.
- Injection rate: The rate at which the power fluid flows to the well.
- Horse power to jet pump: Engine power.
- Pump intake pressure: Flowing bottom hole pressure at pump depth
- Discharge rate: The rate of the commingled fluid (Production + Power Fluid).
- Cavitation rate: The production rate at which the pump will start cavitating.

JEMS suggests an injection rate and pressure of 2147 bpd and 2506 psig in order to reach a production of 1200 bpd. JEMS also shows that the pump will risk a cavitation damage at a rate of 1444 bpd and therefore, the production rate should not reach or exceed the cavitation rate.

Jet pump performance summary for a specified production rate and sensitivity to different injection pressures are shown in Figure 31:

```
Injection Pressure   =   2600. psig
         Production Rate        =   1230. STB/d
         Injection Rate         =   2177. bpd
         Horsepower to Jet Pump =    111. hp
         Pump Intake Pressure   =   1453. psig
         Discharge Pressure     =   2995. psig
         Cavitation Rate        =   1416. STB/d

Injection Pressure   =   2800. psig
         Production Rate        =   1281. STB/d
         Injection Rate         =   2236. bpd
         Horsepower to Jet Pump =    122. hp
         Pump Intake Pressure   =   1393. psig
         Discharge Pressure     =   2972. psig
         Cavitation Rate        =   1367. STB/d

Injection Pressure   =   3000. psig
         Production Rate        =   1329. STB/d
         Injection Rate         =   2293. bpd
         Horsepower to Jet Pump =    134. hp
         Pump Intake Pressure   =   1336. psig
         Discharge Pressure     =   2958. psig
         Cavitation Rate        =   1320. STB/d

Injection Pressure   =   3200. psig
         Production Rate        =   1371. STB/d
         Injection Rate         =   2347. bpd
         Horsepower to Jet Pump =    147. hp
         Pump Intake Pressure   =   1287. psig
         Discharge Pressure     =   2948. psig
         Cavitation Rate        =   1279. STB/d
```

Figure 31: Sensitivity to different injection pressures for the selected combination 11A.

The last two options shows possible production rates higher than the previous options. However, the production rates in both situation are higher than the cavitation rates which increase the risk of damaging some parts or the entire pump.

- Production rate (1329 bpd): production rate is higher than cavitation rate (1320 bpd)
- Production rate (1371 bpd): production rate is higher than cavitation rate (1279 bpd)

Intake pressures vs. production rates for different surface injection pressures are shown in Figure 32 below as well as the cavitation curve.

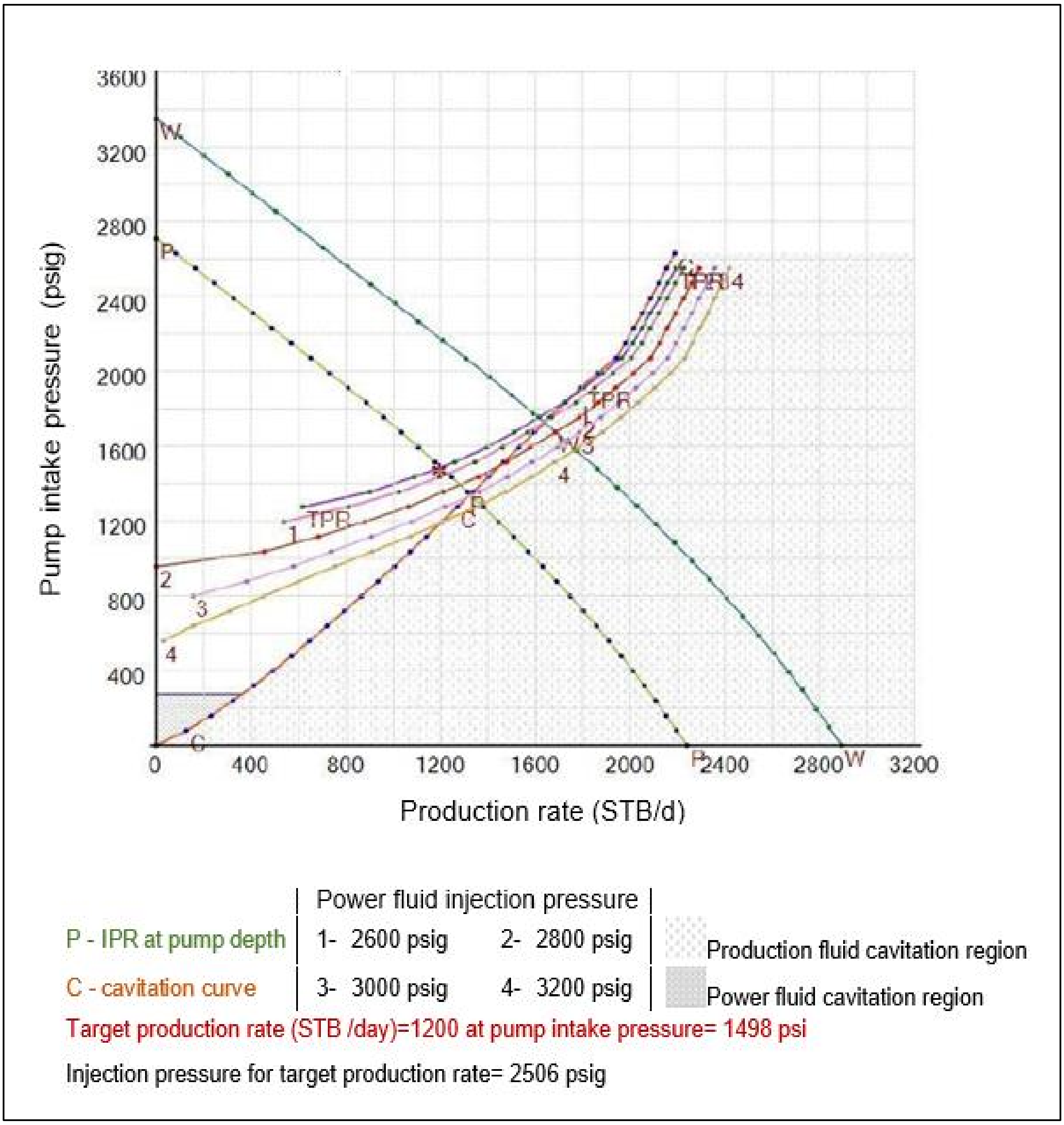


Figure 32: JEMS intake pressures versus production rates graph.

- W-W line represents the inflow performance relationship (IPR) curve.

- P-P line represents the inflow performance relationship (IPR) curve at pump depth.

- TPR curve: TPR stands for target production rate. This is the injection pressure that yields the target production rate when it intersects with the P-P line. This intersection point should not lie in the cavitation regions.

- 2-2, 3-3, TPR (or 1-1) and 4-4 lines- These represents the sensitivities on various Injection pressures. The range of these pressures is set with the input data in “Minimum Power Fluid Pressure” & “Maximum Power fluid Pressure”. Their intersection points with the P-P line should not fall in the cavitation region. If they do, the range of injection pressure set in the ‘Min & Max Power Fluid Pressure’ should be changed. For example, if the intersection points of some the lines fall in the ‘production cavitation’ region, the values of the pressure ranges set initially in the input data should be decreased.

- C-C Line: This curve defines cavitation regions and cavitation rate: This is the rate where TPR line intersects with the C- C Line. If that is the case we either change the nozzle- throat combination or we change the value of the target production rate, depending upon in which cavitation region does the intersection falls in.

4.3 Prosper Well Design

The main objective of this chapter is to finalize and update PROSPER models that have been initially provided and quality checked by the company operating the jet pumps studied in this thesis. A part of the PROSPER files, well data were also gathered and were quality checked and studied in order to be used in the jet pump JEMS modelling and PROSPER designs update.

Unlike JEMS, PROSPER requires inserting a jet pump combination and loss coefficients in the input data as shown in Figure33. PROSPER gives the possibility to randomly select a jet pump combination from a wide list of combinations (nozzle-throat sizes).

The nozzle, throat, diffuser and suction loss coefficients are part of the JEMS program and the user of the program is not required to have any knowledge of these coefficients. The loss coefficients values are considered to be confidential, they can only be determined through several laboratory testing and measurements and the only company who established these coefficients is the company who developed JEMS.

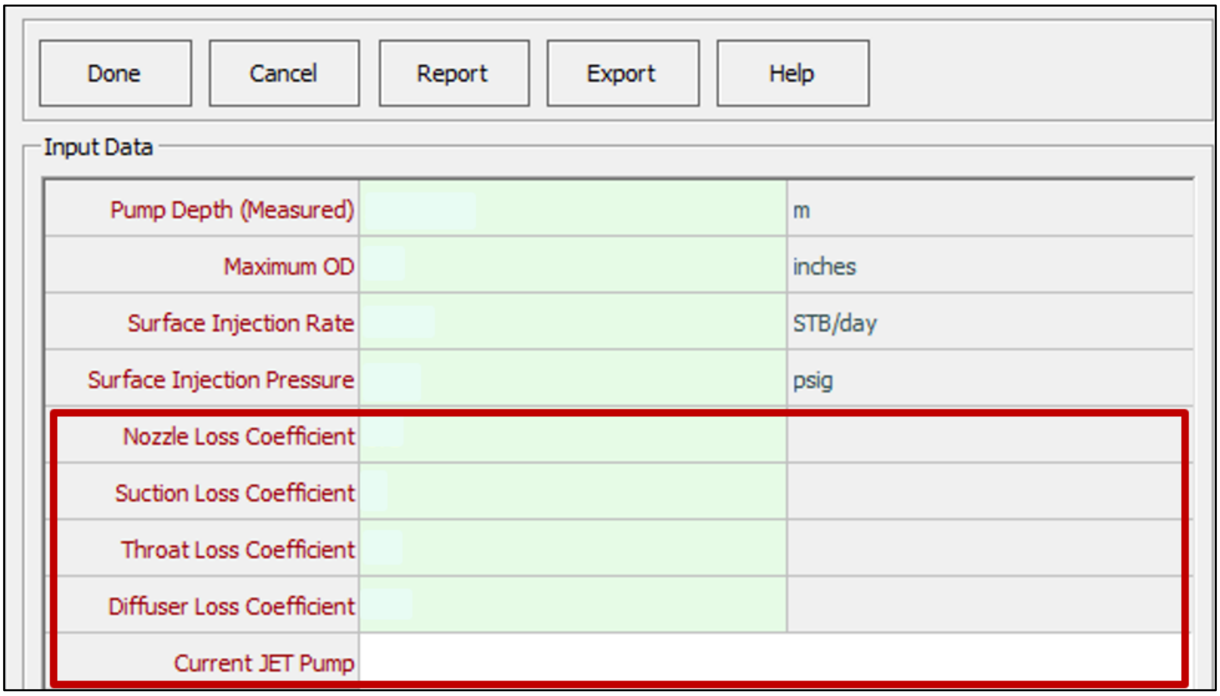


Figure 33: PROSPER input data interface.

The first step to finalize the PROSPER model of a well “AMANI-1” is to determine the appropriate jet pump combination using JEMS. The well input data are as shown in Figure 34:

```
       Field & Well : Amani-1
       Location     : Anaguid East                        Run ID: 105156-14

1. Perforation Depth        (ft) :  8364.0  13.Producing GOR  (scf/STB) :  635.0
2. Pump Vertical Depth      (ft) :  8315.0  14.Gas Sp. Gravity (air=1.) :  0.900
3. Pump Instl  (1) Standard Flow            15.Separator Press   (psig) :  112.0
   (2)Rev. Flow (3)Parallel Flow:  1        16.Well Static BHP   (psig) : 2850.0
4. Casing ID                (in) :  6.094   17.Well Flowing BHP  (psig) : 2078.0
5. Tubing OD                (in) :  3.500   18.Well Test Flow Rate(STB/d): 1040.0
6. Tubing ID                (in) :  2.992   19.Well Head Temp  (deg. F) :  120.0
7. Return Tubing ID         (in) :  N/A     20.Bottom Hole Temp(deg. F) :  180.0
8. Tubing Length            (ft) :  8364.0  21.Bypass Gas? (1)yes (2)no :  2
9. Pipe Cond (1)new(2)avg(3)old  :  1       22.Power Fld (1)oil (2)water:  2
10.Oil Gravity             (API) : 39.700   23.Power Fld API/Sp. Gravity:  1.200
11.Water Cut                 (%) :  21.00   24.Bubble Point Press(psia) : 2223.9
12.Water Specific Gravity        :  1.200   25.Well Head Press   (psig) :  120.0
```

Figure 34: Well “AMANI-1” JEMS input data.

Once the well data is inserted in JEMS, the program well suggest a jet pump combination as well as the power fluid injection pressure, injection rate and the pump cavitation rate. For the studied well” AMANI-1” a combination 9B was suggested as shown in Figure 35.

Referring to Table 1, we can directly determine the nozzle and the throat areas:

- The nozzle size number is 9, it corresponds to 0,0166 in^2 of nozzle area
- The throat size number is 10, it corresponds to 0,0555 in^2 of throat area

```
Oilmaster  9B  Jet Pump Performance Summary for User Specified Production
rate of  1350. STB/d

Production Rate         =   1350. STB/d
Injection Pressure      =   2637. psig
Injection Rate          =   1399. bpd
Horsepower to Jet Pump  =     72. hp
Pump Intake Pressure    =   1810. psig
Discharge Pressure      =   2732. psig
Cavitation Rate         =   1510. STB/d
```

Figure 35: Well “AMANI-1” JEMS output data.

After determining the optimum jet pump combination, the next step is to finalize the PROSPER model. The existing PROSPER file was updated using the latest well test results. Water cut as well as gas oil ratio (GOR) and power fluid injection pressure and rate ad pump depth were inserted in the input data as shown in Figure 36:

Top node pressure	140	psig
Water Cut	19.6	percent
Total GOR	800	scf/STB

Input Data

Pump Depth (Measured)	2534.55	m
Maximum OD	2.8	inches
Surface Injection Rate	1280	STB/day
Surface Injection Pressure	3400	psig
Nozzle Loss Coefficient		
Suction Loss Coefficient		
Throat Loss Coefficient		
Diffuser Loss Coefficient		
Current JET Pump	WFT - 9B - Nozzle (9) Throat (10) An 0.017 At 0.056 R 0.297	

Figure 36: “AMANI-1” PROSPER input data.

The next step to finish the PROPSPER model is to determine the nozzle, suction, throat and diffuser loss coefficients. These loss coefficients are dependent on nozzle and throat area ratio, flow rates and pressures.

Many theories have suggested different loss coefficients that are suitable for a specific jet pump such as Cunningham and Gosline and others. Different loss coefficients suggested by different theories are listed in Table 10 below:

Table 7: Loss coefficients suggested by different theories. [17]

	K_n	K_s	K_t	K_d	$K_t + K_d$
Gosline	0.15	0	0.28	0.10	0.38
Cunningham	0.10	0	-	-	0.30
Petrie et al.	0.03	0	-	-	0.20
Sanger	0.09	0.008	0.098	0.102	-

Each of the theories above has suggested a completely different values for the loss coefficients than the others simply because each of them has built its theory on a different nozzle and throat size and area. As shown in Figure 37 below different sizing of the nozzle and throat are available depending on the manufacturer of the pump.

Kobe				National				Guiberson			
Nozzle		Throat		Nozzle		Throat		Nozzle		Throat	
No	Area	No.	Area	No.	Area	No.	Area	No.	Area	No.	Area
1	0.0024	1	0.0060	1	0.0024	1	0.0064	DD	0.0016	000	0.0044
2	0.0031	2	0.0077	2	0.0031	2	0.0081	CC	0.0028	00	0.0071
3	0.0040	3	0.0100	3	0.0039	3	0.0104	BB	0.0038	0	0.0104
4	0.0052	4	0.0129	4	0.0050	4	0.0131	A	0.0055	1	0.0143
5	0.0067	5	0.0167	5	0.0064	5	0.0167	B	0.0095	2	0.0189
6	0.0086	6	0.0215	6	0.0081	6	0.0212	C	0.0123	3	0.0241
7	0.0111	7	0.0278	7	0.0103	7	0.0271	D	0.0177	4	0.0314
8	0.0144	8	0.0359	8	0.0131	8	0.0346	E	0.0241	5	0.0380
9	0.0186	9	0.0464	9	0.0167	9	0.0441	F	0.0314	6	0.0452
10	0.0240	10	0.0599	10	0.0212	10	0.0562	G	0.0452	7	0.0531
11	0.0310	11	0.0774	11	0.0271	11	0.0715	H	0.0661	8	0.0661
12	0.0400	12	0.1000	12	0.0346	12	0.0910	I	0.0855	9	0.0804
13	0.0517	13	0.1292	13	0.0441	13	0.1159	J	0.1257	10	0.0962
14	0.0668	14	0.1668	14	0.0562	14	0.1476	K	0.1590	11	0.1195
15	0.0863	15	0.2154	15	0.0715	15	0.1879	L	0.1963	12	0.1452
16	0.1114	16	0.2783	16	0.0910	16	0.2392	M	0.2463	13	0.1772
17	0.1439	17	0.3594	17	0.1159	17	0.3046	N	0.3117	14	0.2165
18	0.1858	18	0.4642	18	0.1476	18	0.3878	P	0.3848	15	0.2606
19	0.2400	19	0.5995	19	0.1879	19	0.4938			16	0.3127
20	0.3100	20	0.7743	20	0.2392	20	0.6287			17	0.3750
		21	1.0000							18	0.4513
		22	1.2916							19	0.5424
		23	1.6681							20	0.6518
		24	2.1544								

Figure 37: Jet pump nozzle and throat sizes from different manufacturer. [2]

Referring to Table 1, the jet pumps implemented in the field this thesis is studying are sized as listed in “NATIONAL” nozzle and throat sizes table. Also, Cunningham loss coefficient theory was based on “NATIONAL” nozzle and throat sizes. Therefore, for the next step of PROSPER design we will assume that the loss coefficients are as suggested by Cunningham and as listed below:

- K_n=0,1
- K_t+K_d=0.30
- K_s=0

Theories listed above showed that K_t and K_d are dependent: Kobe: (K_t+K_d=0.38) and Cunningham: K_t+K_d=0.30). To check the dependency of throat and diffuser loss coefficients, several PROSPER simulations were created.

The input data used in the simulation is shown in Table 8 below as well as the output of each simulation. “Amani-1” PROSPER file will be used for all the simulation since it is already quality checked and updated and therefore, the main focus will be only to define the loss coefficients.

Table 8: Throat and diffuser PROSPER sensitivity simulation input and output.

Well “AMANI-1” jet pump combination (**9B**)			
Kn=0,1 and Ks=0, Injection pressure=3400 psi, Injection rate=1280 bpd			
Kt	Kd	Flow rate (bpd)	Pump intake pressure (psi)
0,15	0,15	958,96	2192,70
0,1	0,2	958,96	2192,70
0,165	0,135	958,96	2192,70

For this simulation, nozzle loss coefficient as well as suction loss coefficient were kept constant and they are consecutively 0,1 and 0 as suggested by Cunningham. Throat and diffuser loss coefficients were changed every time, however, their sum was kept 0,30 in order to check the sensitivity of PROSPER output to K_t and K_d.

As shown by this simulation, the production flow rate and the pump intake pressure did not change when changing the throat or the diffuser loss coefficients as long as their sum was kept the same (0,30). Other PROSPER design output were also not affected when changing throat and diffuser loss coefficients such as the pump discharge pressure and the pump power requirements.

The same simulation was applied on different wells in the field this thesis is studying. Independently from the well characteristics, its perforation depth, pump installation depth or the used jet pump combination, PROSPER simulations has showed no sensitivity to throat and diffuser loss coefficients as long as the sum in kept constant. Therefore, for the next simulations performed in this thesis, K_t+K_d is assumed to be 0,30 as suggested by Cunningham and $K_t=K_d=0,15$.

The next step is check the sensitivity of the PROSPER design to the suction loss coefficient. For this simulation, the same PROSPER file for the well "AMANI-1" was used. Nozzle, Throat and diffuser loss coefficients were inserted in the input data interface as shown in Figure 38 and they are as suggested by Cunningham:

- $K_n=0,1$
- $K_t=K_d=0,15$

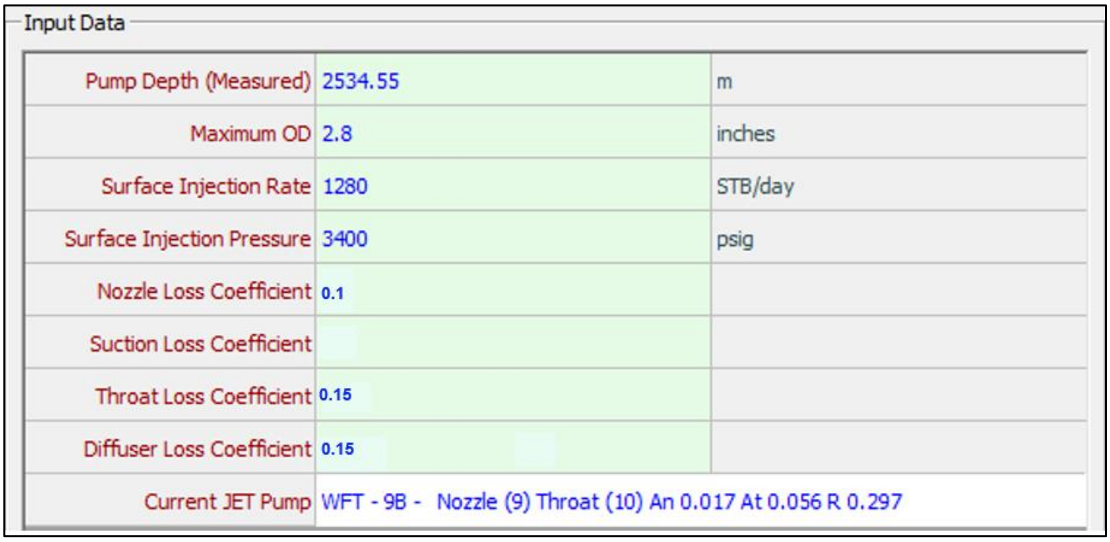


Figure 38: Input data for the suction loss coefficient PROSPER sensitivity simulation.

Different suction loss coefficients were used in the input data and the PROSPER output of the different simulations were compared. The used suction loss coefficients and the final results are as shown in Table 9:

Table 9: Suction loss coefficient PROSPER sensitivity simulation input and output.

Well "AMANI-1" jet pump combination (**9B**)		
$K_n=0,1$ and $K_d= K_t=0,15$, Injection pressure=3400 psi, Injection rate=1280 bpd		
K_s	Flow rate (bpd)	Pump intake pressure (psi)
0	948,96	2192,70
0,10	931,80	2234,48
0,15	905,3	2285,31
0,20	882,4	2301,0

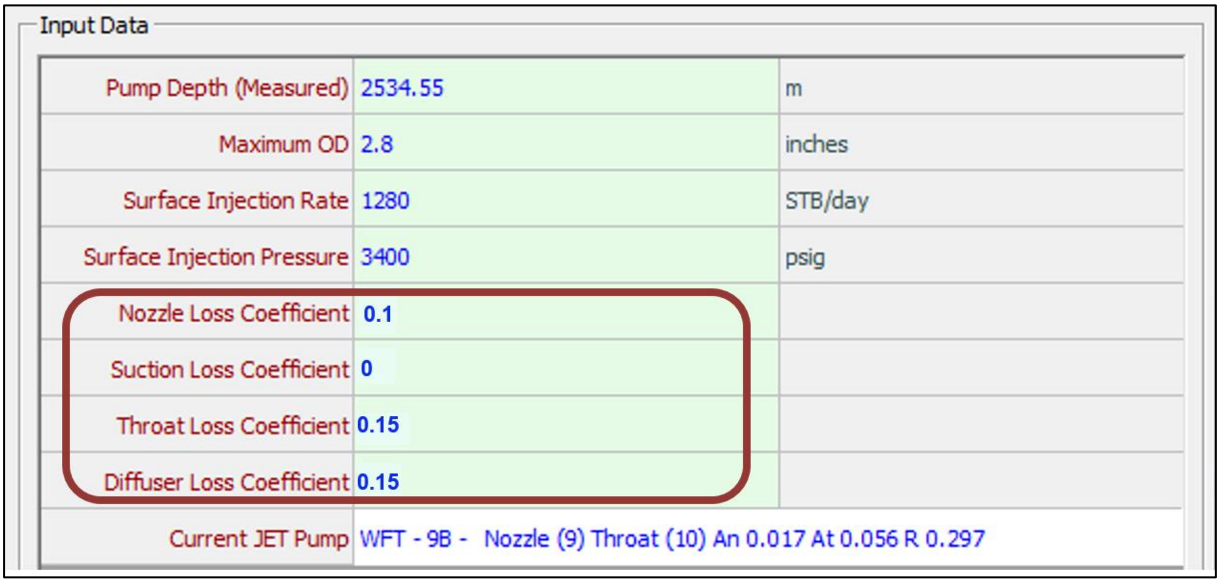
The same simulation was applied to different other wells with different characteristics and operating with other jet pump combinations than 9B. However, PROSPER output showed minor to non-sensitivity to K_s compared to its sensitivity to the nozzle loss coefficient value which will be shown in the next simulations. Therefore, for all PROSPER jet pump models, we will assume that suction loss coefficients is equal to zero (K_s=0).

The previous simulations were executed to check the sensitivity of PROSPER to each of the loss coefficients without taking into consideration the similarity of the final output to the actual well data. The main goal is to check the loss coefficients suggested by Cunningham and there effect on PROSPER jet pump modeling.

The production flow rate and pump intake pressure in each of the previous simulations were different from the actual production flow rate and pump intake pressure of the studied well "AMANI-1". However, there are four loss coefficients, which means four different unknowns. Since the throat, diffuser and suction loss coefficients showed minor to non-effect on the PROSPER model output, they will be considered for the rest of the simulations as listed below:

- K_t+K_d=0.30 and K_t=K_d=0,15
- K_s=0

The next steps is to estimate the nozzle loss coefficient, in order to match the PROSPER files output with the actual well data. A PROSPER model was created using all loss coefficients suggested by Cunningham and they are as shown in Figure 39 below:


Input Data

Pump Depth (Measured)	2534.55	m
Maximum OD	2.8	inches
Surface Injection Rate	1280	STB/day
Surface Injection Pressure	3400	psig
Nozzle Loss Coefficient	0.1	
Suction Loss Coefficient	0	
Throat Loss Coefficient	0.15	
Diffuser Loss Coefficient	0.15	
Current JET Pump	WFT - 9B - Nozzle (9) Throat (10) An 0.017 At 0.056 R 0.297	

Figure 39: Prosper design input with a jet pump combination 9B using Cunningham loss coefficients.

The output of the PROSPER model is shown in Figure 40 below:

Label	Value	Units
Liquid Rate	958.969	(STB/day)
Oil Rate	771.011	(STB/day)
Water Rate	187.958	(STB/day)
Gas Rate	17.378	(1000Sm3/d)
Solution Node Pressure	2336.01	(psig)
dP Friction	-7.35941	(psi)
dP Gravity	3407.13	(psi)
dP Total Skin	0	(psi)
dP Perforation	0	(psi)
dP Damage	0	(psi)
dP Completion	0	(psi)
Completion Skin	0	
Total Skin	0	
Pump Intake Pressure	2192.69	(psig)
Pump Discharge Pressure	3396.45	(psig)
Average Rate Through Pump	1325.19	(RB/day)
Pump Head Generated	1147.6	(m)
Pump Power Requirement	124.19	(hp)
Pump Efficiency	28.6449	(percent)

Figure 40: Prosper design output with a jet pump combination 9B using Cunningham loss coefficients.

As mentioned previously, the loss coefficients suggested by Cunningham were used in the previous simulations to check Prosper sensitivity to each of the coefficients without taking into consideration the final output accuracy.

Prosper showed a possible production rate of 959 bpd. However, the daily reports and well tests have shown a production rate of 1156 bpd. A comparison between the PROSPER output and the well test data are shown in Table 10:

Table 10: A comparison between “AMANI-1” well testing and PROSPER model results.

	Well testing results	**Prosper model results**
Injection pressure (Psi)	3400	3400
Injection rate (bpd)	1280	1280
Total Liquid Rate (bpd)	**1156**	**948.96**
Oil Rate (bpd)	929.50	761,01
Water Rate (bpd)	227.40	187,95
Pump intake pressure (Psi)	**2230**	**2192.69**

As shown in Table 10, the PROSPER model estimated a production rate, almost 200 bpd less than the actual production rate of the well. The Cunningham loss coefficients were used to create a PROSPER model of another well "NADA-1" operating with a jet pump combination 11A.

However, the output results were also different from the actual well data. Production flow rate estimated by PROSPER is lower than the flow rate of the well "NADA-1" with a 25% and it is as shown in Table 11 below:

Table 11: A comparison between "NADA-1" well testing and PROSPER model results.

	Well testing results	**Prosper model results**
Injection pressure (Psi)	3200	3200
Injection rate (bpd)	1950	1950
Total Liquid Rate (bpd)	**~445**	**~330**
Oil Rate (bpd)	**~336**	**~247**
Water Rate (bpd)	**~109**	**~85**

As shown in the examples above, PROSPER simulation output using Cunningham loss coefficients showed different results from the actual well data. Since the sensitivity of PROSPER to the nozzle loss coefficient is not determined, the next step is to estimate the nozzle loss coefficient and its sensitivity to different well input data such as injection rate and pressure and the jet pump combination.

PROSPER calculates well hydraulics and other well properties that are subjected to changes in the wellbore. Different modules may be used depending on the artificial lift system, to verify performance and perform sensitivity analysis for changing conditions. The jet pump module is based on the work by Brown and Hatziavramidis. The equations describing the energy exchange is based on Cunningham energy balance equations. In his model, throat-inlet, nozzle and diffuser is described by the Bernoulli's energy conservation and throat by the momentum balance equation.

Hatziavramidis states that constant value loss coefficients are valid for a specific range of nozzle-throat-area ratios. Constant value loss coefficients are also likely in cases with high Reynolds Numbers. An experimental loss coefficient equation based on Reynolds Number is suited for jet pump-parts that are subjected to high velocity flow and is shown in equation 24 below. [18] [19]

Nozzle loss coefficient K_n: Calculated using Reynolds's number for different injection rates using eq. 24:

$$K_n = \frac{379}{R_e^{0.63}} \tag{24}$$

A plot of Loss Coefficients calculated with equation 24 as a function of Reynolds Numbers are shown in Figure 41.

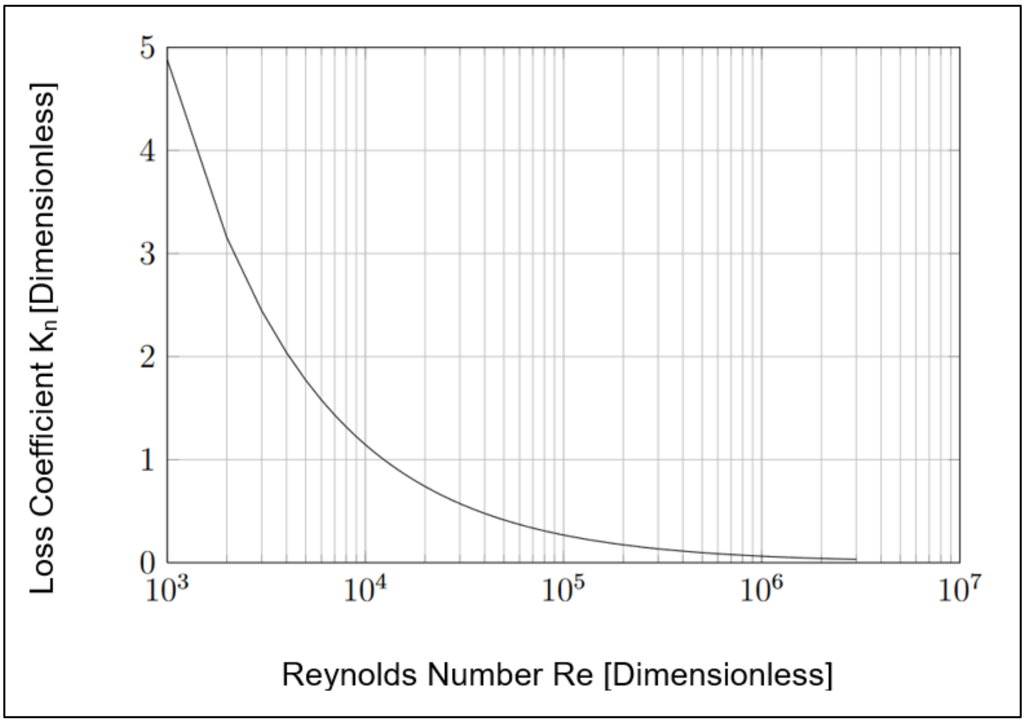


Figure 41: Loss coefficient K_n as a function of Reynolds number Re. [19]

The nozzle loss coefficients can also be expressed as a function of the velocity or the injection rate of the power fluid instead of the Reynolds number. A plot of the nozzle loss coefficient versus injection rate, for a jet pump combination 9B is shown in Figure 42 below:

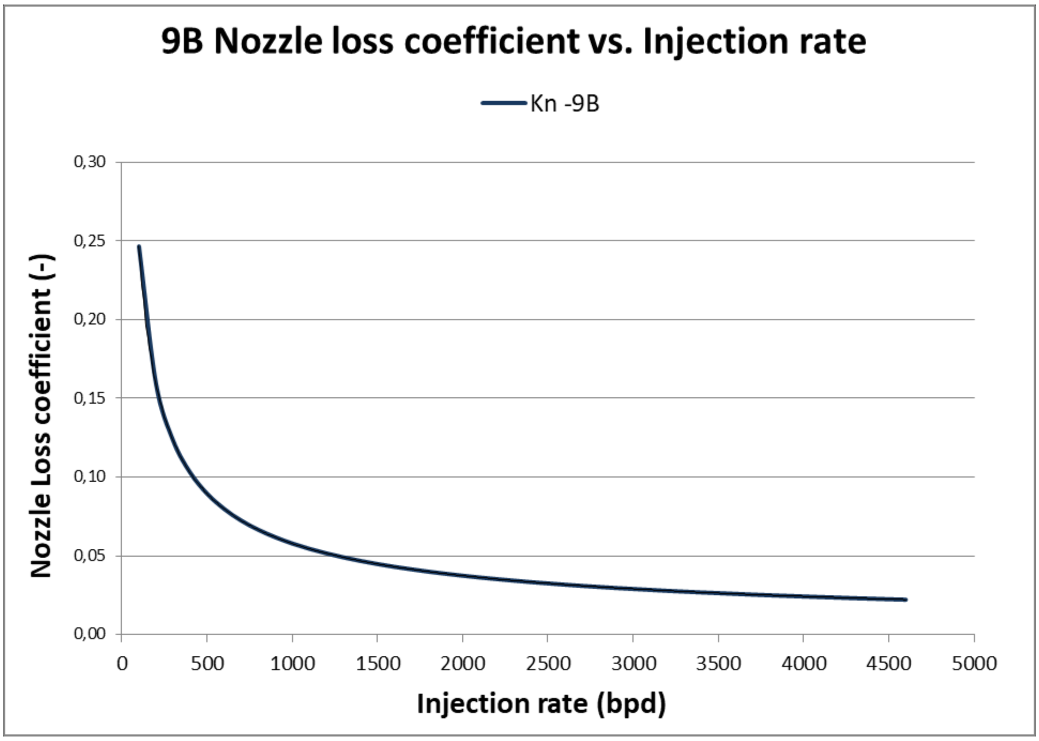


Figure 42: Reynolds loss coefficient K_n as a function of injection rate.

Since we have a pattern of the nozzle loss coefficient changes as a function of the power fluid injection rate, the next step was to randomly change the nozzle loss coefficient in the PROSPER input interface until the jet pump Prosper model matches the well data including daily production rates as well as pump intake pressure.

For the same well “AMANI-1”, a nozzle loss coefficient was calculated as a function of Reynolds number using equation (24). A different nozzle loss coefficient is inserted in the PROSPER input. The simulation results for “AMANI-1” with a specific injection rate and pressure using the random nozzle loss coefficient matched the well data. Both Reynolds and The randomly inserted PROSPER loss coefficients for a jet pump combination 9B are shown in Table 12 below:

Table 12: Reynolds and PROSPER nozzle loss coefficient for a jet pump combination 9B.

injection rate (bpd)	1280		
Jet pump 9B	Reynolds nozzle loss coefficient	PROSPER nozzle loss coefficient	Δv factor
Nozzle	0,0496	0,154	3,10

Δv describes the difference between Reynolds and Prosper loss coefficients. This factor is used to plot PROPSER nozzle loss coefficients curve as shown in Figure 43. The red curve represents the PROSPER nozzle loss coefficients for a jet pump combination 9B, referred to as modified loss coefficients, as a function of the injection rate.

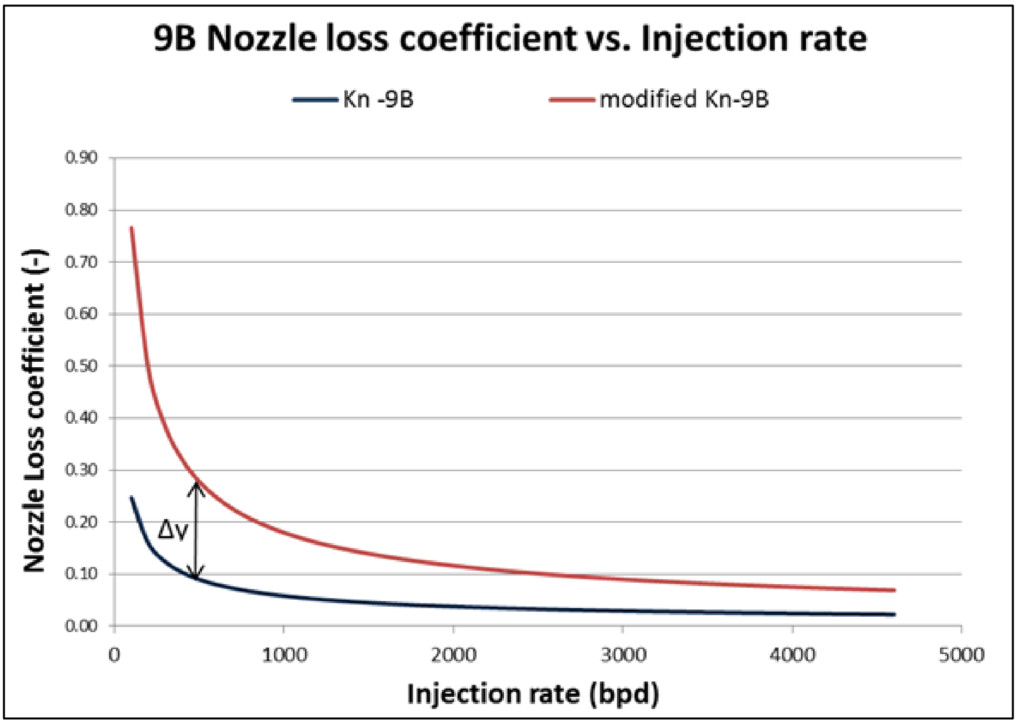


Figure 43: 9B Reynolds and modified nozzle loss coefficient vs. injection rate curves.

Different range of injection rates were used to check the accuracy of the estimated PROSPER nozzle loss coefficient presented by the red curve in Figure 43. Used injection rates and loss coefficients for this simulation are listed in the Table 13:

Table 13: Different injection rates and modified loss coefficient used for Prosper modeling of a 9B jet pump combination.

Pressure [psi]	Injection rate [bpd]	Kn [-]
3400	1260	0,155
3400	1280	0,154
3400	1335	0,151
3400	1355	0,149

The nozzle loss coefficient listed in Table 13 were randomly inserted and changed in PROSPER until the design output matched the well data and daily reports. Different nozzle loss coefficients used in this simulation are plotted as dots on the modified PROSPER nozzle loss coefficients curve as a function of injection rate as shown in Figure 44 below:

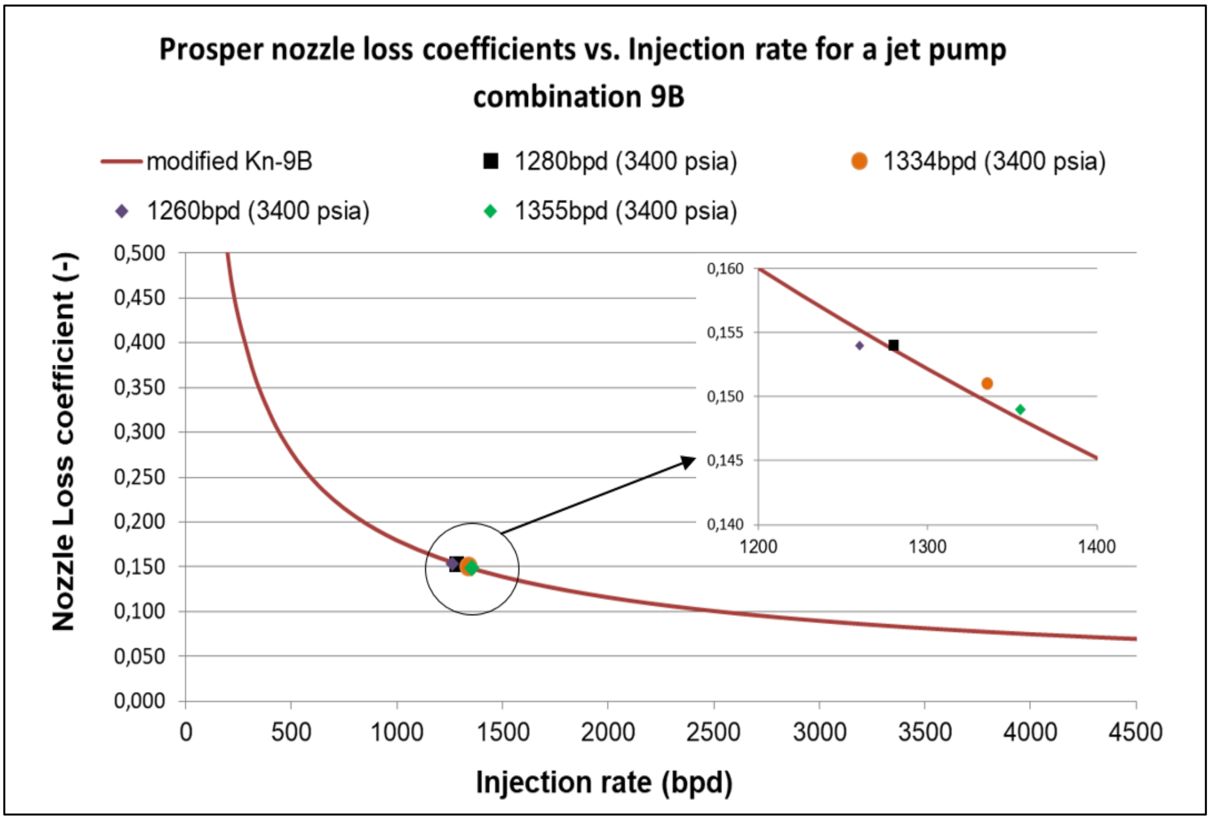


Figure 44: Modified PROSPER nozzle loss coefficients vs. injection rate for a jet pump combination 9B.

The nozzle loss coefficients in Figure 44 are almost colliding with the modified nozzle loss coefficient curve and PROSPER simulations using these coefficients showed accurate results when compared to the actual well production data. Therefore, the modified PROSPER nozzle loss coefficients versus injection rate curve can be used for further PROSPER modeling of a jet pump performance with a 9B combination.

The next step is to further estimate the nozzle loss coefficients for a different jet pump combination. For optimization purpose, a jet pump combination 11B was implemented in the same well “AMANI-1”. The selection of the new combination was done using JEMS as shown in Figure 45:

```
                          W E L L   D A T A   S U M M A R Y
                          =================================


        Field & Well : Amani-1
        Location     : Anaguid East

  1. Perforation Depth       (ft) :  8364.0  13.Producing GOR  (scf/STB) :  877.0
  2. Pump Vertical Depth     (ft) :  8315.0  14.Gas Sp. Gravity (air=1.) :  0.900
  3. Pump Instl  (1) Standard Flow           15.Separator Press   (psig) :  130.0
     (2)Rev. Flow (3)Parallel Flow:  1       16.Well Static BHP   (psig) : 3000.0
  4. Casing ID               (in) :  6.094   17.Well Flowing BHP  (psig) : 2187.0
  5. Tubing OD               (in) :  3.500   18.Well Test Flow Rate(STB/d): 1116.0
  6. Tubing ID               (in) :  2.992   19.Well Head Temp  (deg. F) :  120.0
  7. Return Tubing ID        (in) :  N/A     20.Bottom Hole Temp(deg. F) :  180.0
  8. Tubing Length           (ft) :  8364.0  21.Bypass Gas? (1)yes (2)no :  2
  9. Pipe Cond (1)new(2)avg(3)old :  1       22.Power Fld (1)oil (2)water:  2
 10.Oil Gravity             (API) : 41.300   23.Power Fld API/Sp. Gravity:  1.200
 11.Water Cut                 (%) :  19.00   24.Bubble Point Press(psia) : 2300.0
 12.Water Specific Gravity        :  1.200   25.Well Head Press   (psig) :  135.0
================================================================================
    Oilmaster 11B  Jet Pump Performance Summary for User Specified Production
    rate of  1600. STB/d

   Production Rate        =   1601. STB/d
   Injection Pressure     =   1753. psig
   Injection Rate         =   2066. bpd
   Horsepower to Jet Pump =     71. hp
   Pump Intake Pressure   =   1780. psig
   Discharge Pressure     =   2455. psig
   Cavitation Rate        =   1931. STB/d
```

Figure 45: JEMS input data and combination 11B performance summary.

The same method used to estimate PROSPER modified nozzle loss coefficients for a jet pump combination 9B will be used to calculate Reynolds nozzle loss coefficient and estimate the PROSPER coefficients for an 11B combination. Both coefficients and the difference factor Δv are shown in Table 14:

Table 14: Reynolds and PROSPER nozzle loss coefficient for a jet pump combination 11B.

injection rate (bpd)	1861		
Jet pump 11B	Reynolds nozzle loss coefficient	PROSPER nozzle loss coefficient	∆v factor
Nozzle	0,04650	0,245	5,269

A plot of the Reynolds and PROSPER nozzle loss coefficients versus injection rate, for a jet pump combination 11B is shown in Figure 46 below:

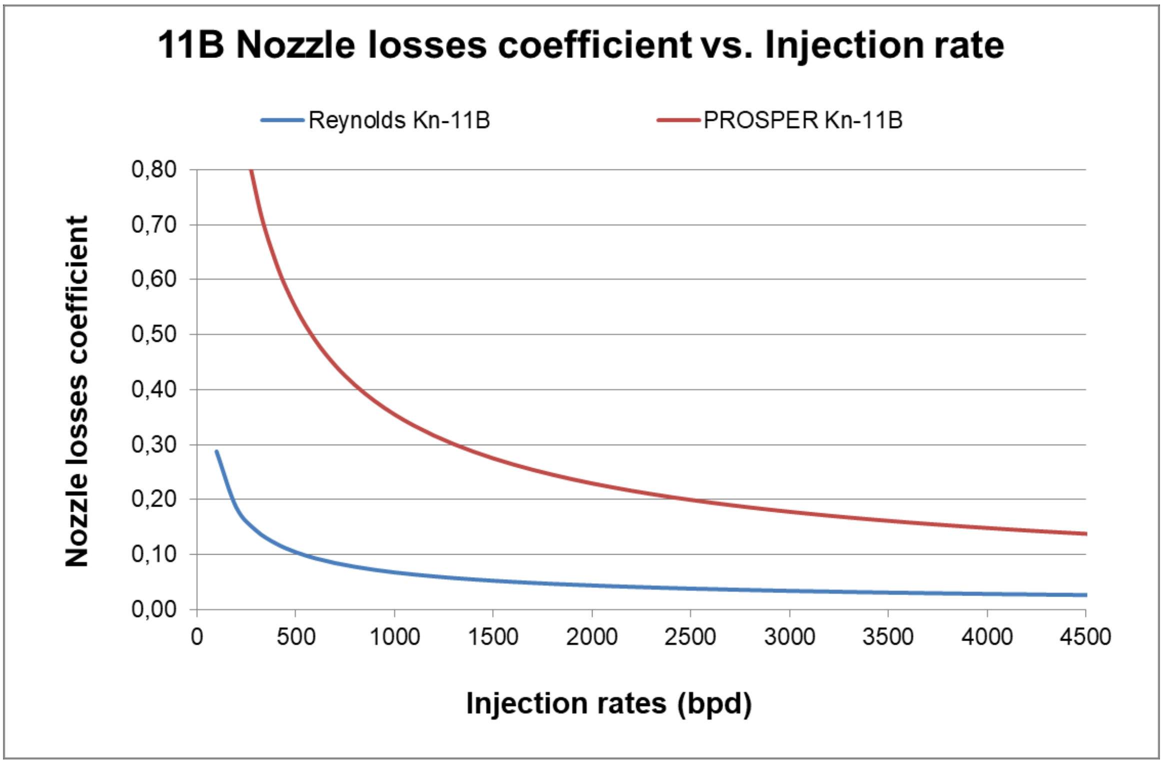


Figure 46: 11B Reynolds and PROSPER nozzle loss coefficient vs. injection rate curves.

Different range of injection rates were used to check the accuracy of the estimated PROSPER nozzle loss coefficient presented by the red curve in Figure 46. Used injection rates and loss coefficients for this simulation are listed in the Table 15:

The nozzle loss coefficient listed in Table 15 were randomly inserted and changed in PROSPER until the design output matched the well data and daily reports of the well “AMANI-1”.

Table 15: Different injection rates and modified loss coefficient used for Prosper modeling of an 11B jet pump combination.

Pressure [psi]	Injection rate [bpd]	Kn [-]
2800	1861	0,233
3000	1920	0,221
3200	1970	0,245
3400	2015	0,252

Different nozzle loss coefficients used in this simulation are plotted as dots on the modified PROSPER nozzle loss coefficients curve as a function of injection rate as shown in Figure 47:

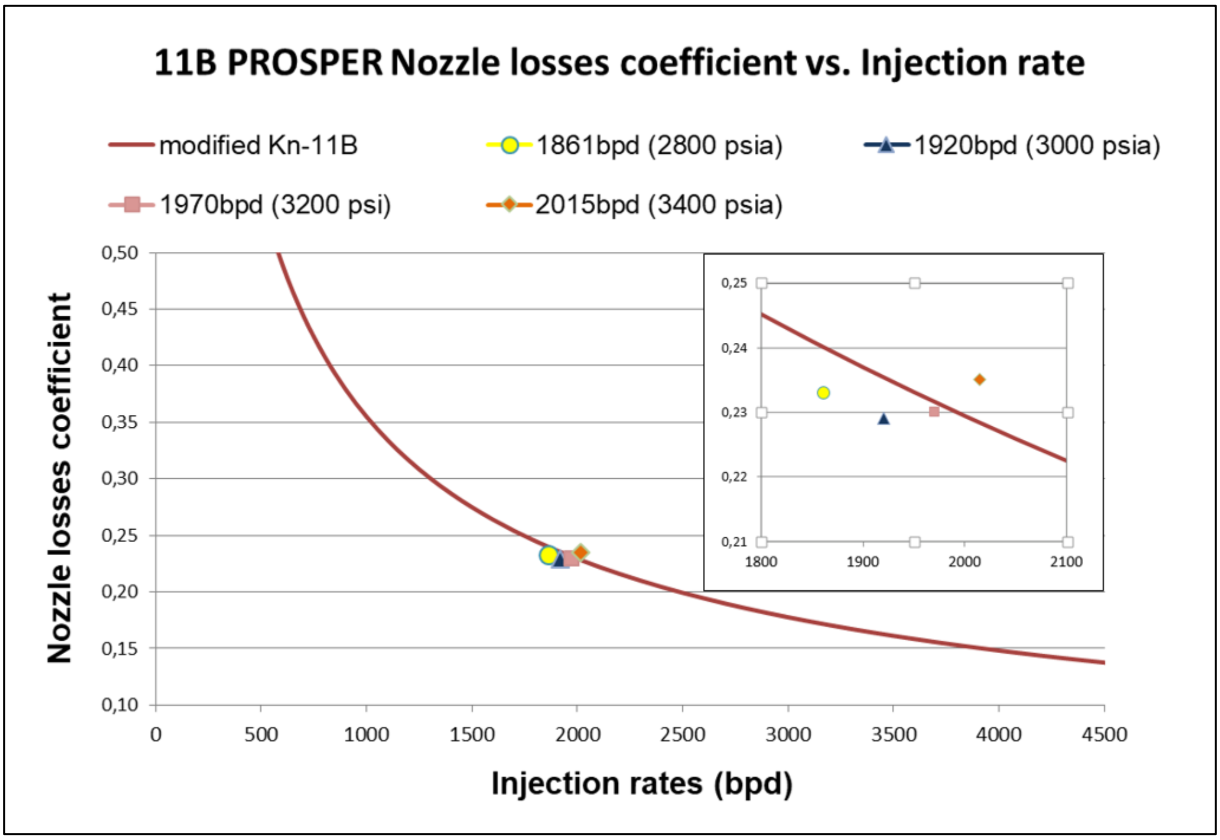


Figure 47: Modified PROSPER nozzle loss coefficients vs. injection rate for a jet pump combination 11B.

The nozzle loss coefficients in Figure 47 are almost colliding with the modified nozzle loss coefficient curve and PROSPER simulations using these coefficients showed accurate results when compared to the actual well production data. Therefore, the modified PROSPER nozzle loss coefficients versus injection rate curve can be used for further PROSPER modeling of a jet pump performance with an 11B combination.

The same method was used to model more wells with different jet pump combinations and injection rates.

A well “NADA-1” with a jet pump combination 11A will be studied. Injection rates and loss coefficients for this simulation are listed in Table 16:

Table 16: Different injection rates and modified loss coefficient used for Prosper modeling of an 11A jet pump combination.

Pressure [psi]	Injection rate [bpd]	Kn [-]
3200	2299	0,25
3200	2280	0,26
3100	2250	0,262
3500	1980	0,27
3200	1950	0,28
3300	1920	0,275

Different nozzle loss coefficients used in this simulation are plotted as dots on the modified PROSPER nozzle loss coefficients curve as a function of injection rate as shown in Figure 48:

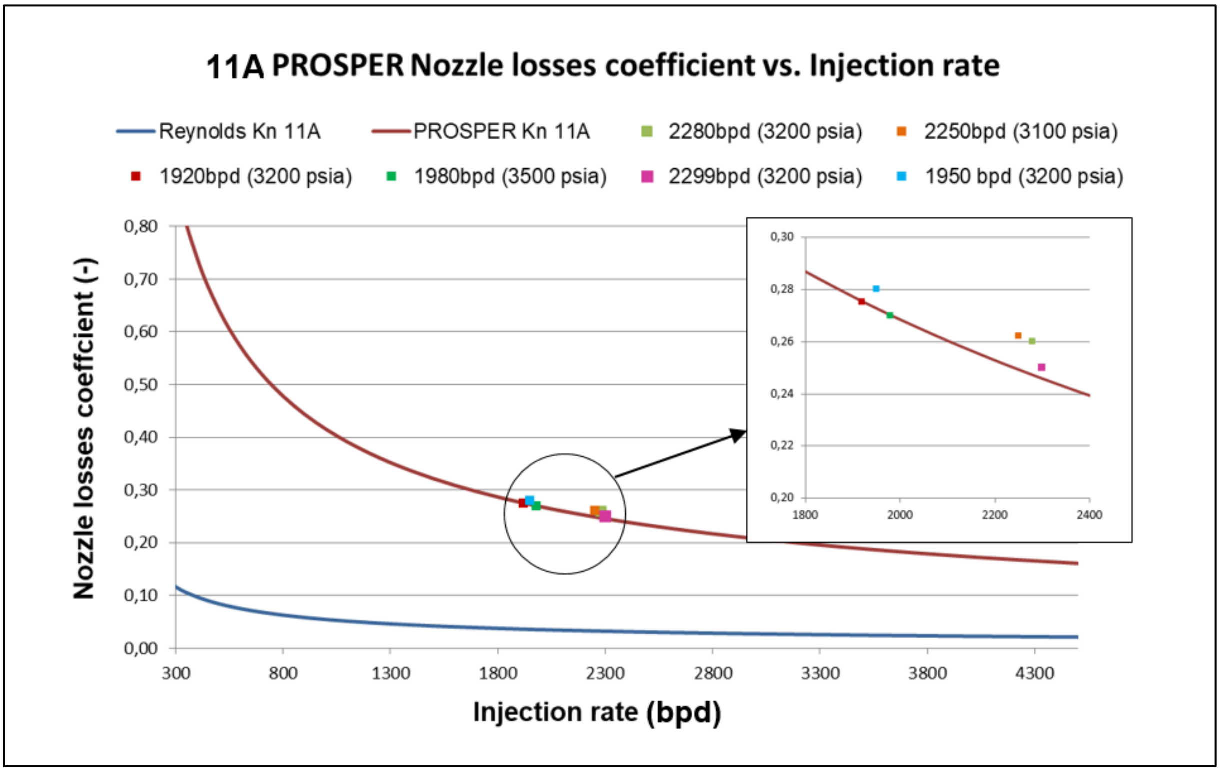


Figure 48: Modified PROSPER nozzle loss coefficients vs. injection rate for a jet pump combination 11A.

For the same well "NADA-1", the same method was used to estimate PROSPER nozzle loss coefficients for a jet pump combination 9F that was implemented for optimization purposes. Both Reynolds and PROSPER coefficients and the difference factor ∆v are shown in Table 14:

Table 17: Different injection rates and modified loss coefficient used for Prosper modeling of a 9F JP combination.

Pressure [psi]	Injection rate [bpd]	Kn [-]
2600	2450	0,065
2700	2490	0,064
2800	2535	0,063
2900	2630	0,0615
3000	2660	0,061
3200	2740	0,06

Different nozzle loss coefficients used in this simulation are plotted as dots on the modified PROSPER nozzle loss coefficients curve as a function of injection rate as shown in Figure 48 below:

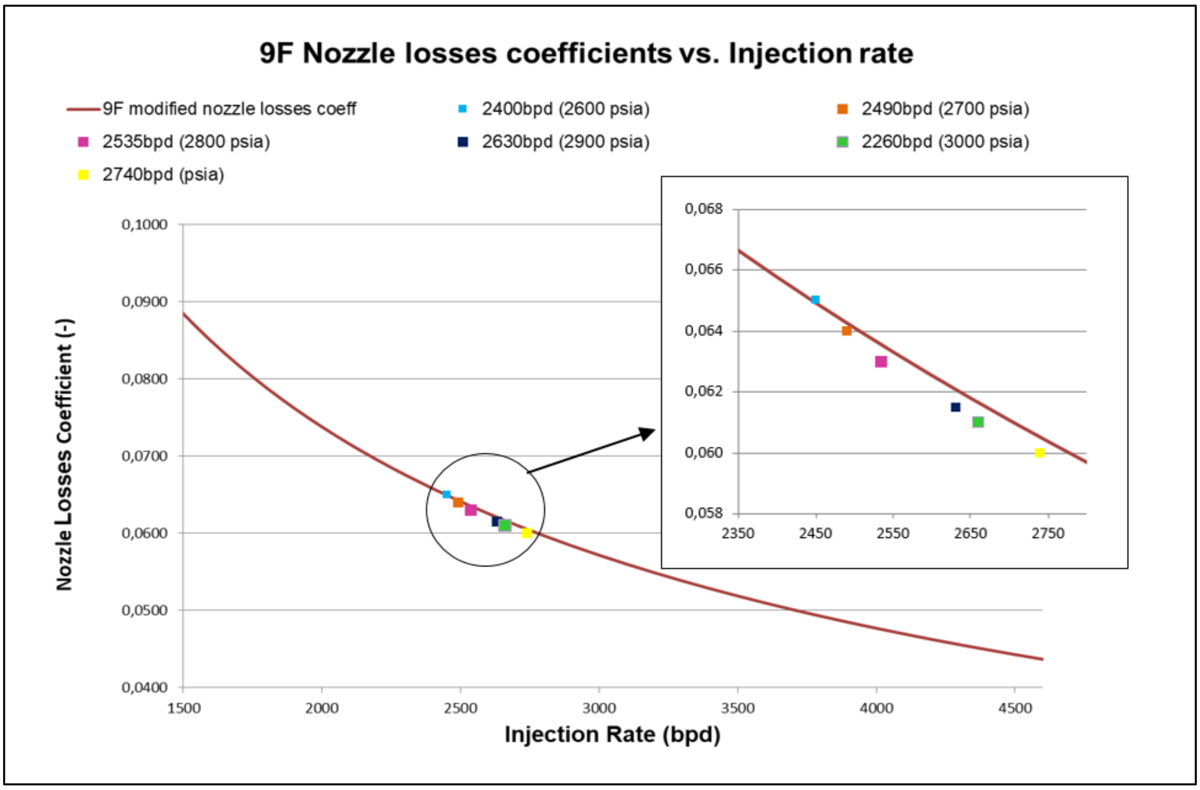


Figure 49: Modified PROSPER nozzle loss coefficients vs. injection rate for a jet pump combination 9F.

For different jet pump combinations and injection rates and pressures, nozzle loss coefficients were estimated as a function of Reynolds number and injection rate. For each combination, a modified nozzle loss coefficients curve was created as a function of the power fluid injection rate. These curves were tested on several wells and different injection rates, and they can be used for further jet pump simulations using PROSPER.

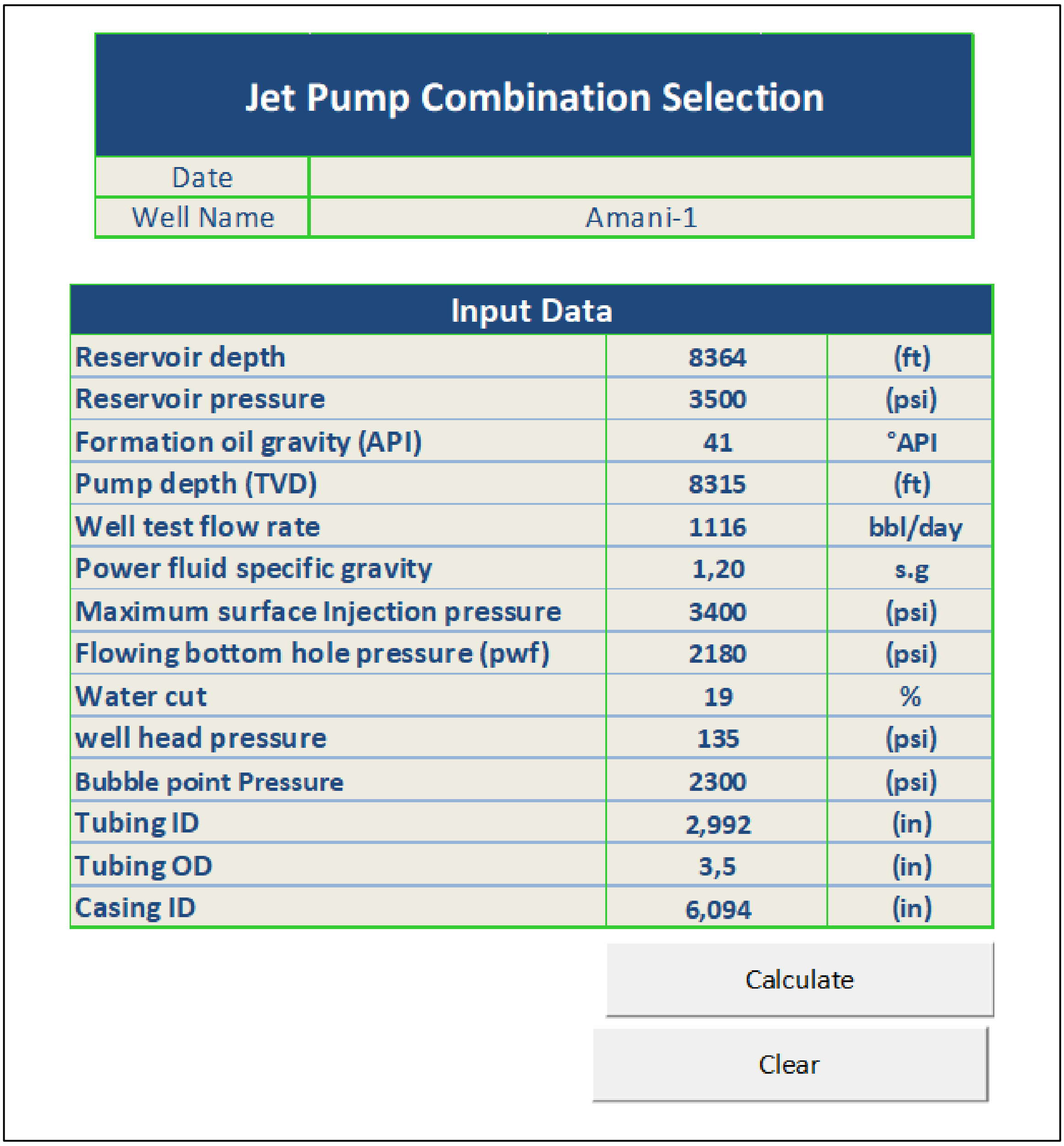
However, this study was based on data provided by the company operating these jet pumps and the loss coefficients were only estimated for the jet pump sizes that are already implemented and in operation phase. Unfortunately, the modified curves created showed no tendency between them or any relation to the different area ratios and therefore, loss coefficients for other jet pump combinations were not covered in this thesis.

Accurate PROSPER models that matched well data were created in order to finalize the integrated asset model. The next step was to create a tool that can be used to design jet pumps, select the optimum nozzle and throat combination, injection rates and pressure and check for cavitation problem.

5 Jet Pump Design Tool

The aim of this study is to create a design tool for jet pumps that allows us to select optimum combination, taking into consideration cavitation problems and surface pump limitations.

By providing input data in the input data interface of the design tool shown in Figure 50, the tool will determine the jet pump combination we can use, define the injection pressure and rate of the power fluid, calculate the intake and discharge pressure of the downhole pump as well as the required horsepower. The tool will also take in consideration the cavitation problem, and will provide us with the cavitation rate.



Jet Pump Combination Selection	
Date	
Well Name	Amani-1

Input Data		
Reservoir depth	8364	(ft)
Reservoir pressure	3500	(psi)
Formation oil gravity (API)	41	°API
Pump depth (TVD)	8315	(ft)
Well test flow rate	1116	bbl/day
Power fluid specific gravity	1,20	s.g
Maximum surface Injection pressure	3400	(psi)
Flowing bottom hole pressure (pwf)	2180	(psi)
Water cut	19	%
well head pressure	135	(psi)
Bubble point Pressure	2300	(psi)
Tubing ID	2,992	(in)
Tubing OD	3,5	(in)
Casing ID	6,094	(in)

Calculate

Clear

Figure 50: Input interface of jet pump selection tool.

When designing a jet pump or any other artificial lift, the goal is also to reach the highest efficiency. Therefore, the designing steps that will follow considers the highest efficiency for each pump ratio as shown in table 18 and its equivalent dimensionless head ratio H, dimensionless flow ratio M and area ratio R.

Table 18: Jet pump ratio's flow rate, head and efficiency.

Ratio	M@ max efficiency (-)	H@ max efficiency (-)	Efficiency %	R (-)
X	0.38	0.74	28	0.48
A	0.47	0.66	31	0.38
B	0.75	0.42	31.8	0.30
C	1	0.32	32.2	0.24
D	1.3	0.25	32.5	0.18
E	1.9	0.17	33	0.15

The next step is to calculate is to calculate the injection pressure and rates of each possible combination, from X to E and calculate the cavitation rate and the required horse power as shown in the equations (15) to (24) from chapter 2 “the jet pump fundamentals”:

To be able to calculate intake, injection and discharge pressures, friction pressure losses as well as Hydrostatic pressure gradient in the tubing and the annular area were calculated as shown in Table 19 and Table 20:

Table 19: Friction pressure losses and hydrostatic pressure through tubing.

Friction pressure losses through tubing							
	Liquid rate	V(ft/s)	Re	1/racine (f)	f	delta P friction	delta P hydrostatic
X	2937	2,93	1,64E+05	15,60	0,00411	979,03	4322,07
A	2374	2,370	1,32E+05	15,28	0,00428	666,99	4322,07
B	1488	1,485	8,29E+04	14,57	0,00471	287,92	4322,07
C	1116	1,114	6,22E+04	14,14	0,00501	172,10	4322,07
D	858	0,857	4,78E+04	13,74	0,00530	107,82	4322,07
E	587	0,586	3,27E+04	13,16	0,00577	54,99	4322,07

Table 20: Friction pressure losses and hydrostatic pressure through annular area.

Friction pressure losses through Annular area							
	Liquid rate	V(ft/s)	Re	1/racine (f)	f	delta P friction	delta P hydrostatic
X	2937	1,055	4,25E+04	13,57	0,00543	161,03	3601,73
A	2374	0,853	3,44E+04	13,24	0,00570	110,47	3601,73
B	1488	0,535	2,15E+04	12,54	0,00636	48,43	3601,73
C	1116	0,401	1,62E+04	12,10	0,00683	29,24	3601,73
D	858	0,308	1,24E+04	11,71	0,00730	18,48	3601,73
E	587	0,211	8,51E+03	11,14	0,00806	9,56	3601,73

Calculated injection, intake and discharge pressure for each combination are shown in Table 21 below:

Table 21: Injection, intake and discharge pressures and injection and production rates of different jet pump combinations.

combination	Injection rate q1 (bpd)	Production rate q3 (bpd)	Dimensionless head (H)	Intake pressure p3 (psi)	Discharge pressure p2 (psi)	Injection pressure p1 (psi)
X	2937	1116	0,74	1242	3576	6743
A	2374	1116	0,66	1365	3626	7055
B	1488	1116	0,42	2100	3688	7434
C	1116	1116	0,32	2470	3707	7550
D	858	1116	0,25	2744	3718	7614
E	587	1116	0,17	3043	3727	7667

Once the nozzle area is calculated, the tool will automatically select a nozzle from the nozzle and throat area Table 1. The tool was designed to select a nozzle with an area equal to the calculated nozzle area. If an exact equal area is not available the tool will select the first nozzle with a smaller area. The results of the selected nozzle areas for each combination are as shown in Table 22:

Table 22: Calculated nozzle area and selected nozzle number and area from manufacturer table.

combination	Calculated nozzle area An (in)	Nozzle number	Nozzle area
X	0,03571	12	0,0342
A	0,02839	11	0,0269
B	0,01838	9	0,0166
C	0,01412	8	0,0130
D	0,01110	7	0,0102
E	0,00779	5	0,0063

As shown in Table 22, a nozzle was selected for each combination from a provided manufacturer nozzle and throat areas as well as its equivalent number. The attributed nozzle's area s are slightly smaller than the calculated areas. Therefore, a recalculation of injection, intake and discharge pressures as well as the injection and production rate is recommended to determine the actual operating parameters of the pump when using the attributed nozzles. Using the same equations, pressure losses through tubing and in the annular area were recalculated and the results are shown in Table 23 and 24:

Table 23: Recalculated friction pressure losses and hydrostatic pressure through tubing.

Recalculated Friction pressure losses through tubing							
	Liquid rate	V(ft/s)	Re	1/racine (f)	f	delta P friction	delta P hydrostatic
X	2814	2,809	1,57E+05	15,53	0,00415	906,03	4322,07
A	2247	2,243	1,25E+05	15,19	0,00433	604,08	4322,07
B	1342	1,340	7,48E+04	14,41	0,00481	239,38	4322,07
C	1029	1,027	5,73E+04	14,01	0,00509	148,84	4322,07
D	791	0,790	4,41E+04	13,61	0,00540	93,25	4322,07
E	476	0,475	2,65E+04	12,84	0,00606	37,86	4322,07

Table 24: Recalculated friction pressure losses and hydrostatic pressure through annular area.

Friction pressure losses through Annular area							
	Liquid rate	V(ft/s)	Re	1/racine (f)	f	delta P friction	delta P hydrostatic
X	2814	1,010766172	4,07E+04	13,50	0,00548	149,23	3601,73
A	2247	0,807	3,25E+04	13,16	0,00577	100,23	3601,73
B	1342	0,482	1,94E+04	12,38	0,00653	40,41	3601,73
C	1029	0,370	1,49E+04	11,98	0,00697	25,35	3601,73
D	791	0,284	1,15E+04	11,58	0,00745	16,03	3601,73
E	476	0,171	6,89E+03	10,83	0,00853	6,63	3601,73

The next step is to calculate the new injection and production rates, as well as the intake discharge and injection pressures using the selected nozzle areas of each combination from X to E. The results of are shown in Table 25:

Table 25: Recalculated Injection, intake and discharge pressures and injection and production rates of different jet pump combinations.

combination	Injection rate q1 (bpd)	Production rate q3 (bpd)	Dimensionless head (H)	Intake pressure p3 (psi)	Discharge pressure p2 (psi)	Injection pressure p1 (psi)
X	2814	1069	0,74	1209	3587	6816
A	2247	1056	0,66	1340	3636	7118
B	1342	1007	0,42	2091	3696	7483
C	1029	1029	0,32	2468	3711	7573
D	791	1028	0,25	2744	3721	7629
E	476	904	0,17	3043	3730	7684

The next step is to determine the cavitation rates in order to decide which combination can be used without the risk of cavitation damage of the downhole pump. Cavitation index Ic and cavitation rates are calculated using the equations below:

Cavitation Index Ic, cavitation rate and required horsepower HP of each combination are as shown in Table 26:

Table 26: Horsepower, cavitation index and rate of different jet pump combinations.

combination	Horsepower (HP)	Cavitation Index (-)	cavitation rate (bbl/day)
X	163	1,399	1100
A	130	1,384	1389
B	78	1,369	1479
C	59	1,365	1712
D	46	1,364	1895
E	27	1,362	1599

Finally, the tool output interface will show different operating rates and pressure of each combination as well as the nozzle number and the required horsepower. The tool will also highlight in red the cases where there is a risk of cavitation. The output interface is as shown in Figure 51.

For a Surface Injection Pressure of : 3400 (psi)

Check caviation

combination	Nozzle number	I.R (bbl/day)	Q (bbl/day)	PIP (psi)	PDP (psi)	HP	C.R (bbl/day)
X	12	2814	1069	1209	3587	163	1100
A	11	2247	1056	1340	3636	130	1389
B	9	1342	1007	2091	3696	78	1479
C	8	1029	1029	2468	3711	59	1712
D	7	791	1028	2744	3721	46	1895
E	5	476	904	3043	3730	27	1599

Warning:
C.R cells in Red referes to the combinations where the cavitation limit was exceeded !!!

Recommendation:

Nomenclature:
I.R: Injection Rate
P.R: Production Rate
PIP: Pump Inateke Pressure
PDP: Pump Discharge Pressure
HP: Horse Power
C.R:Cavitation Rate

Figure 51: Output interface of a jet pump design tool.

Final results are summarized in a table as shown in Figure 51, where user can read possible jet pump combinations and there equivalent operating parameters such as pressure, rate and horsepower. The tool considers the highest possible injection rate of the power fluid depending on the available surface pump and alerts the designer in case cavitation of the downhole pump may occur to avoid damage.

6 Conclusion/ Interpretation/ Recommendations

This master thesis was done in collaboration with a company operating in South Tunisia to investigate jet pump performance using state of the art integrated asset modeling software, investigate failures and recommend any optimization possibilities in one of the fields they operate.

The oil field studied in this thesis has faced several problem. The availability of workover rigs was limited in addition, the waiting time for intervention was high, having reached more than five months for one of the wells. This resulted in high production deferment volumes combined with a low mean time between failures for the existing jet pumps. These issues have affected the field operation and reduced the wells profitability.

The very first jet pump was installed initially as a backup for an ESP pump. However, from the first month of operating, the pump has failed 40 times. One of the main reasons was the use of old equipment and the poor maintenance. The lack of experience played also a major role in the pump failure. Severe pump cavitation and wrong jet pump combination selection have both resulted in several pump retrieving operations and production deferment. The aim of this thesis was to investigate these failures to improve efficiency of jet pumps and reduce mean time between failures.

Since the first jet pump installation in 2015 until today, maintenance was done more frequently to prevent surface units damage especially the surface pump unit. Nozzle plugging, corrosion, and retrieving have been reduced tremendously and well interventions were only planned to improve productivity of the well and not because of a failure.

An assessment of the current jet pump operating conditions has shown that more improvement can be done when it comes to the maintenance and equipment quality and can be summarized in the following points:

- Regular maintenance
  - Repairing of plunger packing and fluid valves.
  - Ensure appropriate lubrication quality and frequency.
  - Regular maintenance and appropriate support of all surface unit components to avoid frequent leakage and shutdowns.

- Design and operations
  - Corrosion inhibitors injection and power fluid quality control reduced nozzle plugging and corrosion.
  - Oversized power unit is a common issue. Injection flow rate requirement is below the capability of the surface pump resulting in vibration, noise and washout. This requires a speed limitation of the engine or plunger size changing.
  - Appropriate vertical, horizontal and cyclone separators installation, control and maintenance as well as proper fluid level reading facilities on sigh.

The thesis was dedicated to design jet pumps using state of the art integrated asset modeling software to analyze actual performance and to recommend any optimization possibilities. The nozzle and throat sizes of the studied wells were already selected using JEMS. Unlike JEMS, PROSPER requires the knowledge of the jet pump loss coefficients. Based on the available wells test data and daily production reports provided from the company, several simulations were run in order to estimates these loss coefficients. The estimations were based on several theories such as the non-sensitivity of PROSPER results to the suction loss coefficient (Ks =0). The throat and diffuser loss coefficients have shown to be dependent and there sum is equal to 0,30. Therefore, the purpose of the simulations was to estimate the nozzle loss coefficients for different jet pump sizes.

An infinite number of nozzle loss coefficients were estimated and they are represented in curves as a function of the pump injection rate. A PROSPER model was created for each and every well operated with a jet pump in order to finalize the IAM. However, the study only focused on one field in South Tunisia with a limited number of jet pump combinations. Despite the progression that different nozzle sizes have shown, nozzle loss coefficients covered in this study have no tendency. In this case, nozzle loss coefficients for the rest of the jet pump combinations cannot be estimated but rather studied individually following the same method carried in this thesis.

The finally part of this thesis was to create an excel tool that selects a jet pump combination for each well. The tool was designed to obtain the highest production potential of a jet pump operating wells taking into consideration cavitation problems. The tool also provides the operating parameters such as the injection rates and pressures as well the discharge pressure and rate of the downhole pump assembly.

Even though the jet pump performance in general has improved and the production has remarkably increased, the current state of some surface equipment is unfavorable for the company future plans. Since there are serious thoughts about jet pumps as a reliable ALS and not only a backup plan, the company is considering purchasing new equipment and is planning to switch from rental to operating its own new jet pump units which will eventually improve the overall performance of the jet pump operated wells.

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Abbreviations

JP	Jet Pump
VLP	Vertical Lift Performance
PVT	Pressure Volume Temperature
IAM	Integrated Asset Modeling
ALS	Artificial Lift System
PIP	Pump Intake Pressure
PDP	Pump Discharge Pressure
NPSHa	Net Positive Suction Head available
NPSHr	Net Positive Suction Head required
BHA	Bottom-Hole Assembly
BHP	Bottom-Hole Pressure
IPR	Inflow Performance Relationship
OPFS	Open power fluid system
CPFS	Closed power fluid system

Nomenclature

V_n	average velocity of power fluid in the nozzle, (ft/s)
V_s	average velocity of power fluid in the suction area, (ft/s)
V_t	average velocity of power fluid in the throat, (ft/s)
A_n	nozzle area, (in^2)
A_s	suction area, (in^2)
A_t	throat area, (in^2)
Q_n	flow rate in the nozzle, (Stb/d)
Q_s	flow rate in the suction area, (Stb/d)
Q_t	flow rate in the throat, ,(Stb/d)
H_t	total head, (ft)
P	density of the power fluid, (lb/ft^3)
A	cross section Area, (in^2)
P	power fluid pressure, (psi)
G	gravitational constant
V	velocity of power fluid, (ft/s)
M_q	mass flow rate, (kg/s)
P_n	power fluid pressure at the nozzle inlet, (psi)
P_d	discharge pressure at the annular space, (psi)
P_s	Intake/suction pressure, (psi)
P_a	power fluid pressure at the throat inlet
P_{wh}	wellhead Pressure, (psi)
P_{imax}	maximum Injection Pressure at the surface, (psi)
B_o	oil volume factor, (bbl/STB)
$\Delta P_{friction}$	pressure loss due to friction, (psi)
S_g	power fluid specific gravity, (-)
$\Delta P_{hydrostatic}$	hydrostatic pressure gradient, (psi)
M	dimensionless flow rate, (-)
R	dimensionless area ratio,(-)
H	dimensionless head ratio, (-)
E	pump efficiency, (%)
q_1 / IR	injection flow rate, (Stb/d)
q_2 / PR	production flow rate, (Stb/d)
q_{wt}	well test flow rate, (Stb/d)
Mc	cavitation flow Ratio, (-)
Ic	cavitation Index, (-)
GOR	gas oil ratio, (Scf/Stb)
HP	horse power, (hp)

TVD	total vertical depth, (ft or m)
ID	inside diameter, (in or mm)
MD	measured depth, (ft or m)