HIVAC (HIGH VOLUME AND COMPRESSION) CAGES FOR SUBSURFACE SUCKER ROD PUMPS.

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Abstract

Inefficiencies in sucker rod pumping systems due to gas interference are major concerns for petroleum producing companies throughout the world. This paper describes an innovative cage design that is based on years of exposure to sucker rod pump inspection and repair from many varied field conditions and on compression and flow comparison testing. Observance of results obtained from working closely with the Alberta Research Council in Edmonton Alberta on numerous projects specifically related to conventional, thermal, vertical and horizontal sucker rod pumping was also influential. The creative cage design addresses the two features that are absolutely key to good standing cage performance. These two features are (1) high compression capability and (2) large flow capacity. The equipment used to perform the flow and compression testing allowed actual visual observation. The tests compared many different cage designs and demonstrated how those design differences affected cage performance.

Introduction

Reciprocating subsurface rod pumps, as we know them today, differ very little conceptually from those used for water production by the Chinese many hundreds of years ago. Hold down seal assemblies, barrel tubes, plungers, traveling valves, and standing valves are still the five major parts of a pump. However, materials, configurations, tolerances, efficiencies, and costs have all changed drastically from those early days. Bamboo barrels and plungers have been replaced with steel alloys, stainless steel alloys, and copper-nickel alloys. They may be processed by the application of contemporary technologies to resist corrosion or wear or both. Hand carved wooden balls and seats have been replaced with heat treated stainless steel, cast cobalt alloys, tungsten carbide alloys, and certain types of ceramic. Tolerances that were once measured in 10'ths of an inch (.010) are now measured in 1,000'ths (.001) and 10,000'ths (.0001) of an inch. (A blonde human head hair is approximately .001" in diameter.) For example, correctly manufactured barrel tubes will not vary more than .002" on inside diameter from end to end. Typically, barrel tubes range in length from 5 to 40 feet.

This evolution is a direct result of the new and challenging demands placed on pumps due to changes in operating conditions within which the pump must operate.

Comparison Of Pumping Applications

When the oil industry was in its infancy, crude oil was produced from wells which had been dug by hand.
Pumping or lifting depths have changed from around 30 feet in these early wells to depths greater than 16,000 feet.

Pumping temperatures have changed from 50° F in the early shallow wells to as much as 400° F in some insitu or fire flood applications and up to 650° F in some of the thermal (steam assisted)production.

Production rates have increased from as little as 2 to 5 barrels per day, produced by a small diameter pump in the early days, to as much as 9000 barrels per day produced by a 7-3/4" bore pump in the 1990's.

Horizontal drilling and production technology has changed the pump orientation from the standard vertical position to any angle from vertical to horizontal.

Due to the increase in pump jack size, stroke lengths have increased from less than 20 inches on some of the early pumps to as much as 32 feet today. Down hole stroke or net plunger travel has also been increased by the recent application of the fiberglass sucker rod. The elasticity of the fiberglass upper portion of the rod string allows the lower portion of the string to travel more than 140% of surface stroke length.

Stroke speed or strokes per minute(SPM) of the pump jack has increased from a leisurely 2 and 3 SPM in the early days to applications which exceed 20 SPM today.

We can readily see that pump application has undergone enormous change.

Traveling Valve Assembly

The traveling valve assembly is the most common major pump component damaged during operation down hole. Deterioration usually manifests itself through a change in shape and size of the ball stop and the ball guides. The ball stop is that feature inside a cage that limits the upward or linear motion of the ball. The ball guides limit the side or diametric ball travel inside the cage. The ball and seat may also suffer damage. Spalling , or orange peeling, may occur on the ball as a result of high single point loading when it makes contact with a flat ball stop. The seat may also sustain damage after either spalling of the ball or the change in tolerance between the ball and the ball guides. Either condition will allow improper contact with the sealing (lapped) surface of the seat, causing damage and therefore the loss of fluid seal. The condition described above is commonly referred to as "cage pounding", "rounded out", "seat pounded", etc. Almost invariably, this type of damage incurred by the traveling assembly during the down hole pumping operation is caused by the pumping condition known as "fluid pound".

Dynamics Of The Pumping Cycle

In order to better understand "fluid pound", we will consider the dynamics of the pumping cycle.

During "normal" pumping, while the plunger is on the up stroke cycle, the traveling ball and seat is closed and the column of fluid above the plunger is being lifted inside the tubing towards the surface. Annular fluid forces the standing ball off the seat and follows the upward motion of the plunger, thus filling that portion of the barrel tube immediately below the plunger. As the plunger slows and stops at the top of the stroke, the standing ball moves down onto the seat and the plunger starts downward. Pressure builds up between the traveling and standing valves until the hydrostatic load on the top of the traveling ball is over come and the traveling valve opens. The open traveling valve and plunger (which is hollow) fall down through the fluid in the barrel tube which is held in place by the closed standing valve. Near the bottom of the stroke, the plunger slows and stops. The traveling ball then moves down onto the seat, and the plunger starts upward, repeating the up stroke cycle.

Further to the pumping cycle, we must understand the relationship between ***pump displacement*** and ***fluid flow rates*** entering and exiting the pump.

In down hole pumps powered by conventional pump jacks, the plunger is on the upstroke half the time and on the down stroke half the time. *Pump displacement is* the theoretical production capability of the pump based on the cross sectional area of the plunger, the distance the plunger travels, and the number of strokes the plunger makes in a given time frame.

Let us consider a 2" bore pump with plunger travel of 100" and 10 strokes per minute. Cross sectional plunger area is 3.1416 $in.^2$, times 100" of travel x 10 SPM = 3,141.6 $in.^3$, or 13.6 US gallons per minute *pump displacement.* However, the *fluid flow rate,* as it enters and exits the pump, is *double the displacement rate* since it takes place in one half the time. Therefore, the flow rate entering and exiting the pump is double the 13.6 US GPM example shown above, or 27.2 US GPM.

Fluid Pound

Fluid pound occurs when the portion of barrel tube below the plunger is not completely filled with liquid during the up stoke cycle of the plunger. Due to the *incomplete barrel fillage*, the traveling valve does not immediately encounter liquid on the down stroke. It remains closed, since the pressure build up is *not sufficient* to lift the traveling ball off the seat. Downward plunger travel continues, and will increase in velocity until mid-point of the stroke is reached. At some point in this down stroke, the traveling valve comes in contact with liquid in the barrel. When this happens, tremendous hydraulic shock occurs, and immediately sufficient pressure is built up to overcome the hydrostatic load on the traveling ball. The ball is instantly catapulted off the seat, contacting the ball stop on the top inside of the cage.

The more often this "pounding" occurs, the more the ball and seat, the ball guides, and the ball stop will suffer damage from the greatly increased ball velocity. As soon as the ball and seat begin to leak, erosion of the lapped surface begins and production starts to decrease. At some point, this *loss of production* forces the economic decision to pull the pump.

The hydraulic shock that occurs, as described above, may be severe enough to split the barrel or break the bottom of the barrel off. This same shock may also cause damage to the sucker rod string immediately above the pump, and if severe enough, may cause damage to the gear box on the pump jack.

History has shown that when the barrel tube if filled with liquid, most if not all, of the premature failures in the traveling assembly disappear. Both cage life plus ball & seat life are increased considerably. ***Much more is accomplished by treating the cause rather than the symptom.***

Pump Efficiency And Incomplete Pump Barrel Fillage

Pump efficiency is the *ratio of produced liquid measured at surface* compared to *theoretical pump displacement* based on down hole stroke length.

Plunger slippage, leaking pump valves, poor seal between the hold down and the pump seating nipple, a hole in the tubing or pump barrel, will all have a negative influence on production and pump efficiency.

However, the single major cause of low or poor pump efficiency is *incomplete barrel fillage* with liquid. Incomplete pump barrel fillage with liquid may occur as a result of any of the following conditions:

1) Pump displacement exceeds formation deliverability.

2) Liquid accompanied with gas out of solution is present at pump intake.

3) Gas comes out of solution as the liquid passes through the pump intake.

Considering <u>condition #1</u> above, pump efficiency will be increased by more closely matching the pump displacement with the deliverabilty capability of the well.

Condition #2 requires two considerations. Providing that the pump **is not** "gas locking", pump efficiency may not change significantly even though a wide range of pump displacement rates is employed. This employment make take place in the form of changes in pump bore diameters, stroke lengths and/or stroke speeds. Actual liquid production may increase or decrease parallel to changes in the pump displacement while the pump efficiency does not change simply because the pump is "seeing" the same ratio of gas to liquid in the fluid entering the pump. The only way to improve the pump efficiency in this category of applications is to employ some means of gas-liquid separation device that will allow a greater ratio of liquid to gas in the fluid at the pump intake.

Knowledge of the relationship between pump displacement and fluid entry flow rates must be employed when determining size and type of separator best suited for a given application. Consideration of pump volume and fluid flow rates must also be given when determining size and length of the dip tube.

The second consideration is if the pump **is** "gas locking".

Gas Locking And Pump Compression Ratio

Gas locking occurs when the pressure that builds up on the down stroke between the traveling valve and the standing valve is not sufficient to over come the hydrostatic load on the traveling ball. As a result, the traveling valve does not open or remains "locked", hence new fluid is not moved through the pump on the next up stroke. This condition allows the column of fluid above the plunger to move up and down as the plunger travels through the pumping cycle.

The pressure that can be built up on the down stroke of the pump is determined by two factors.

(a) *pump intake pressure* (Pump intake pressure is the producing down hole pressure at the pump intake)

(b) *the compression ratio of the pump*

The compression ratio of the pump is determined by two factors.

(1) *the stroke length or travel of the plunger,*

(2) *the proximity of the traveling ball and seat to the standing ball and seat, not merely the proximity of traveling assembly to standing assembly or traveling cage to standing cage.*

The use of recessed or hex plugs as seat retainers in the traveling cage will change this valve proximity. The standard seat retainer extends below the bottom of the traveling cage approximately 1". The recessed or hex plug is flush with the cage bottom. The employment of the recessed plug allows the traveling ball and seat to get 1" closer to the standing ball and seat. The recessed plug should be used in applications of gas interference.
However, standing cages have a much greater influence due to design differences which vary by size and type of cage. Overall length of same size cages differs considerably. For example, 2" bore (2¼") standing cage length varies from less than 4" to 9". Since the lower end of the cages are quite similar (about 1¼" from cage bottom to top of seat), the distance from the top of the standing seat to the top of the cage can vary by more than 5 inches. This design difference *greatly influences the compression* ratio since this space cannot be displaced by the plunger.

Consider a 2" bore pump with 100" stroke length. The cages mentioned above are employed on 2" bore pumps. The same size of cages are also used on 1½" and $1^3/_4$" externally thread heavy wall barrels, and in traveling and standing positions on 2¼" tubing pumps. If we are not able to get the traveling ball and seat, and the standing ball and seat closer than 10" apart, we have an approximate *pump compression* ratio of 10:1(100:10). If through a change in standing cage design, we can change that proximity to 4", the *compression ratio increases* to 25:1. If we are lifting fluid from 5000 feet, we may have 2000 PSI hydrostatic load on the traveling ball. Assume the pump intake pressure is 100 PSI, and the compression ratio is 10:1. We can build up only 1000 PSI on the down stroke, hence the pump is gas locked. If however, the compression ratio of the pump is 25:1, with the same pump intake pressure, the pressure that will build up on the down stroke will be 2500 PSI. This is more than enough to overcome the hydrostatic load, and will prevent gas locking from taking place.

The above illustration is used to demonstrate the influence standing cage design has on compression capability of a pump, and how compression capability affects gas locking.

Cage Flow Capacity And Gas/Vapor Break - Out

<u>Condition #3</u> occurs when fluid entering the pump is subjected to a flow restriction anywhere on the pump intake that creates a pressure drop sufficient to allow gas/vapor to break out of solution and occupy space in the pump barrel below the plunger.

This condition is influenced by the following factors which vary greatly:

- the proximity of bubble point pressure of the fluid compared to the pump intake pressure
- the viscosity of the fluid
- the available flow area or flow capacity of equipment on the pump intake.
- the pump displacement.
- the amount of gas in solution.
- thermal processes (fluids at close to saturation conditions) in combination with the above.

Since this paper is about pump cages, I will keep most of my comments to cages rather than all of the components that may be on the pump intake such as screens, strainers, filters, dip tubes, hold down mandrels, strainer bushings, and adapters to pressure and temperature sensing equipment to name a few. The *flow capacities* of different styles and manufacturers of standing cages *vary widely*. Once again, if we consider the standing cage on the 2" bore pump (which is a 2¼" cage size) we can see from the flow capacity comparison chart attached, that this size of cage varies in flow capacity from 6 US GPM to 34 US GPM. These flow results were obtained using water with approximately 18" of head.

These numbers by themselves do not tell us very much. However, when we recognize that **a** *27.2 US GPM flow rate is required to fill that same 2" bore pump with 100 stroke length and a stroke speed of 10 per minute,* we begin to understand how standing cage design influences the flow capacity and hence the pressure drop in that component on the pump intake.

When we have the *pump requiring a flow rate of 27.2 US GPM,* and a standing cage that will allow *only 6 US GPM of water* with a pressure drop of less that 1 PSI to pass through it, the *pressure drop created* with oil of higher viscosity at or near the required flow rate of 27 US GPM, is going to *increase* substantially. It is the *pressure drop*, created by the increased upstream pressure required to force the fluid through the *restrictive standing cage* at the flow rate required by the pump displacement, that allows the *gas/vapor to come out of solution.* Once out of solution, the *gas/vapor occupies space* in the compression chamber of the pump that would otherwise be occupied by liquid. The resulting decrease in liquid production negatively impacts the pumps efficiency.

If the pump was operated at a shorter stroke length or slower stroke speed, the required flow rate through the standing cage into the pump would be correspondingly less and may be low enough to prevent gas from breaking out. If the *stroke length or stroke speed were increased*, the required flow rate would be higher, the pressure drop would be greater, and *more gas/vapor would come out of solution.*

As flow capacity increases, the pressure required to force a given volume through the cage, in a certain time frame, decreases. As a result, the pressure drop across the cage also decreases. The lower pressure drop will allow a greater amount of gas/vapor to stay in solution. *This in turn means that a greater portion of the barrel tube will be filled with liquid.* What ever is in the barrel tube,below the plunger, when the plunger is at the top of the stroke is what will be produced on the next up stroke. If the barrel is half filled with liquid and half with gas/vapor, the pump will be 50% efficient.

Greatest pump efficiency is attained when the compression chamber of the pump is completely filled with liquid. It is the amount of liquid produced by the pump compared to the theoretical displacement that determines the pump efficiency.

The above illustration shows how the flow capacity of the standing cage influences the amount of liquid that enters the pump. It is the design of the cage, relative to the available flow area around the ball inside the cage, that determines this flow capacity.

An international oil producer commissioned the Alberta Research Council (ARC) in Edmonton, Alberta, Canada to evaluate HIVAC cages for production of high viscosity fluids. Four viscosity's of test fluid were utilized. At the same pressure drop, tests showed that the HIVAC cage will allow more than double the volume of fluid to pass through it than the standard same size cage it was compared with. This increased flow capacity appeared to be consistent over the full viscosity range applied. Please refer to "ARC TESTING" attached.

These laboratory test results coincide with actual field data from various well installations.

Summary

The two most important features of a high performance standing cage are,

(1) the ability to allow close proximity of the traveling ball & seat to the standing ball & seat. This will allow for greatest compression capability to combat gas locking.

(2) the ability to allow the greatest amount of fluid to pass through in the shortest possible time with the lowest possible pressure drop.

Not surprisingly, these two features are related to each other in an inverse manner.

In other words, as one is increased, the other is decreased. This is why cages that are designed for high compression usually exhibit poor or low flow capacity. Conversely cages that are designed for higher flow capacity commonly demonstrate poor or low compression capability.

As shown in the attached charts, the High Volume And Compression (HIVAC) cage design offers the "best of both worlds." Flow capacity in the 2¼" cage size is almost 500% more than some same size cages, and compression capability is 83% more than some same size cages. Please refer to the related attachments.

Actual production results obtained over a number of years, and from many hundreds of installations indicate that very remarkable production increases have resulted from the installation of the HIVAC cage. Greatest production increases have been obtained from wells that have had HIVAC cages installed that previously had cages with low compression capability and/or low flow capacity relative to pump displacement.

Wells with high viscosity liquids have also shown very significant production increases.

The author is not aware of any wells that have exhibited a decrease in production after the installation of the HIVAC cages. This would seem to indicate there is minimal or no "risk" involved with installation of this cage design.

Cost premiums are three to four times that of mild steel cages. However, when similar materials are compared, the premium decreases considerably. At time of writing, HIVAC cages were offered in 17-4 PH heat treated stainless steel.

Best performance has been achieved in the 2-1/4" cage when installed with the HIVAC (Hi-Volume ball and seat). This ball and seat combination exhibits much greater flow capacity than the standard API design with either the API or alternate size of ball.

Attention must be given to all pump components up stream from the HIVAC cage to assure that they all have flow capacity equal to or greater than the cage.

Acknowledgments

The author wishes to thank the employees of producing companies for sharing information regarding production before and after HIVAC cage installation. Special thanks to Chevron Canada Resources and to Don Patterson, Richard Raiwet, and Shauna Noonan for their joint efforts in preparing and sharing meaningful comparative production reports for a field in Northern Alberta. Special thanks to Imperial Oil Resources Limited and to Ivan Purdy for assistance provided.

References

1. "Laboratory Evaluation Of Hivac Rod Pump Valves For Production Of High Viscosity Fluids", closed file report, ARC, Roy Coates and Gerry Pierce, March 1996.

2. Toma, P., Coates, R., Nguyen, D., Ivey, R., Good, W. " Improved Design of Rod Pump Valves For Thermal and Horizontal Wells - Laboratory and Field Results", Paper presented at the 46th Annual Technical Meeting of The Petroleum Society of CIM May 1995.

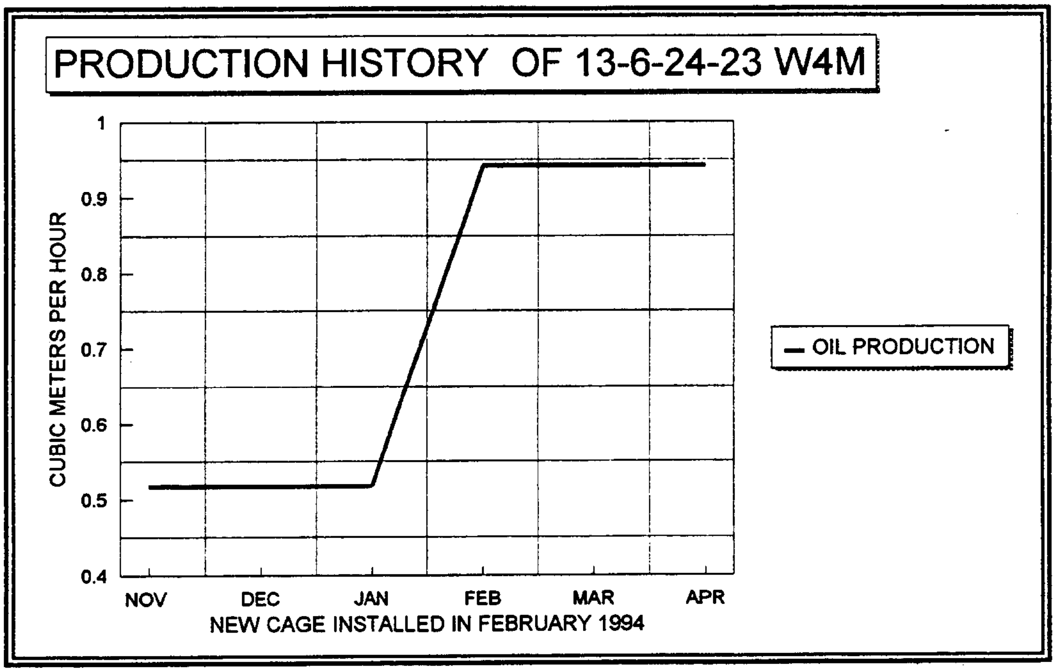
3. Toma, P., Korpany, G., Sudol, T., and Gonnie, K. "Hydrodynamic Characteristics and Comparative Study of Sucker-Rod Pump Valves," closed-file report ARC/AOSTRA-Industry - Thermal Well Pumping Committee, October 1992.

4. "The History of Canada's Oil & Gas Industry" by Ed Gould, Hancock House Publishers.

CASE HISTORY

THE FOLLOWING IS A CASE HISTORY OF 13-6-24-23 W4M BASED ON DATA OBTAINED FROM THE ENERGY RESOURCES CONSERVATION BOARD IN CALGARY, ALBERTA (ERCB) WHICH SHOWS THE PRODUCTION HISTORY IN CUBIC METERS PER HOUR BEFORE AND AFTER THE INSTALLATION OF T H E NEW STYLE (HIVAC) STANDING CAGE IN EARLY FEBRUARY OF 1994

	OIL M3	TOT HRS	OIL M3/HR	WATER	GAS 103M3X10
OCT	458.6	741	0.62	1.2	1.77
NOV	426.1	720	0.59	1	2.62
DEC	399.7	724	0.55	0.8	2.76
JAN	301.4	544	0.55	1.2	1.87
FEB	621.4	634	0.98	1.2	3.45
MAR	730.3	744	0.98	1.4	3.48
APR	624.8	720	0.87	1.2	4.13

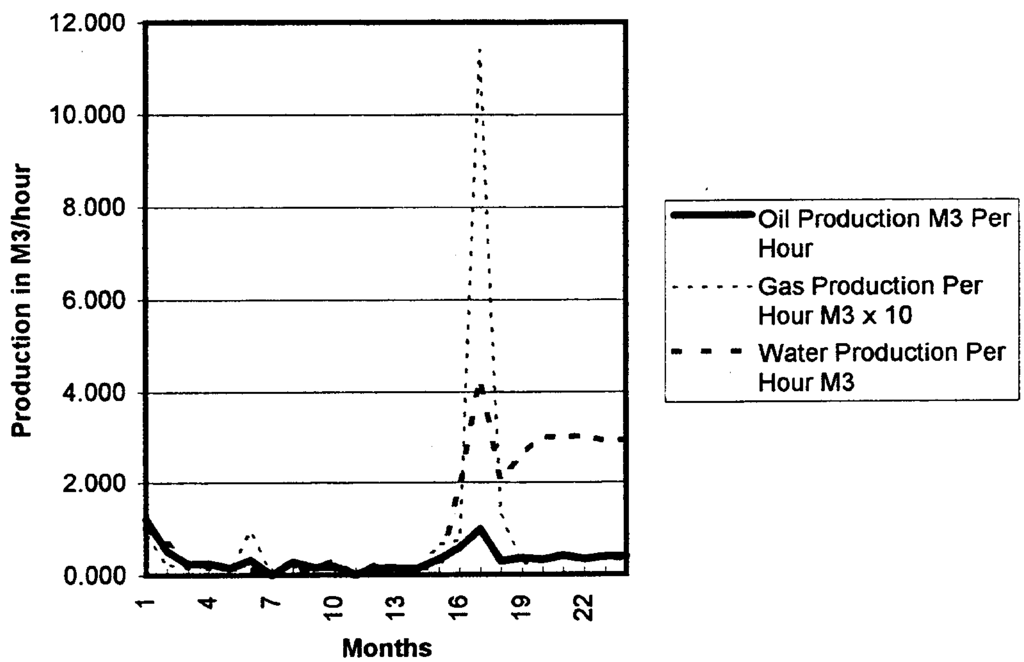


INCREASE COMPARING AVERAGE PRODUCTION IN 1993 OF .518 M3/HOUR TO FEBRUARY THROUGH APRIL 1994 OF .943 M3/HOUR IS **82%!!!**

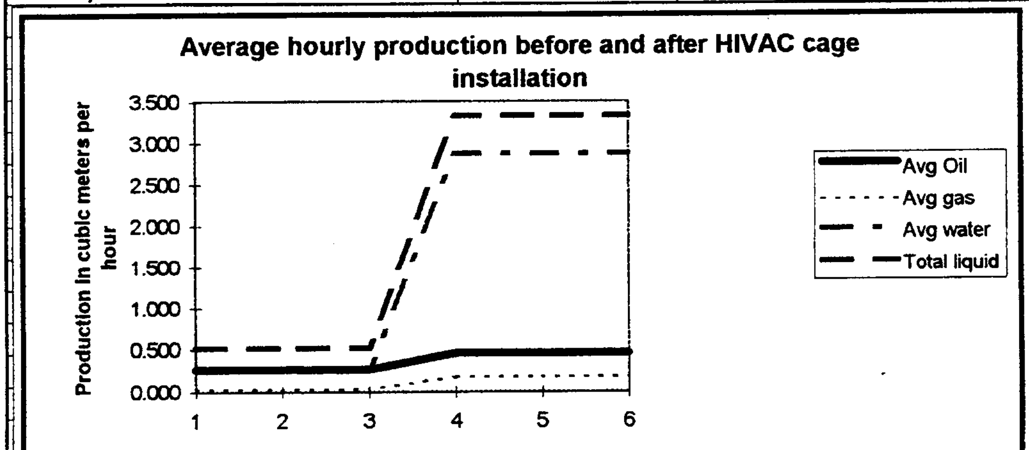
THE INCREASE OF .425 CUBIC METERS OF OIL PER HOUR GENERATES EXTRA REVENUE WHICH WILL EXCEED $115,000 U.S. EVERY 100 DAYS ASSUMING OIL TO BE VALUED AT $18.00 U.S. PER BARREL.

CASE HISTORY

Location				
00/10-35-109-07W6/0				
HIVAC standing cage installed April 1996 (month 16).				



Oil production increase	70.43%		
Gas production increase	571.95%		
Water production increase	1063.13%		



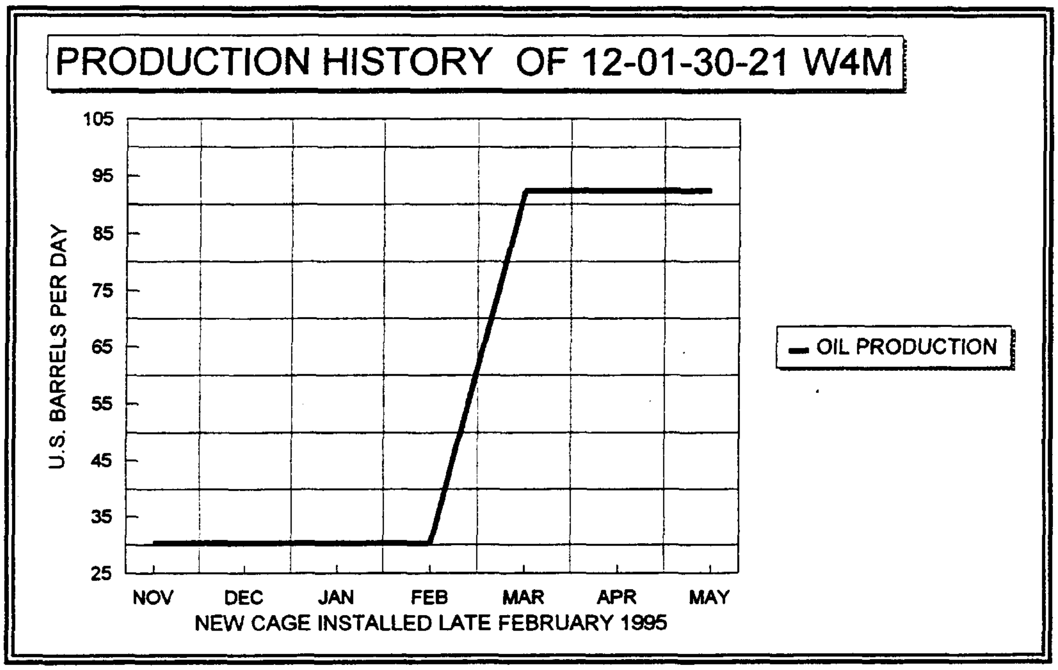
The **extra** oil production from April through December 1996 has generated	$192,400
gross revenue assuming oil valued at $25.00/bbl	

CASE HISTORY

THE FOLLOWING IS A CASE HISTORY OF 02/12-01-30-21 W4M BASED ON DATA OBTAINED FROM THE ENERGY RESOURCES CONSERVATION BOARD IN CALGARY, ALBERTA (ERCB) WHICH SHOWS THE PRODUCTION HISTORY IN U.S. BARRELS PER DAY BEFORE AND AFTER THE INSTALLATION OF THE NEW STYLE (HIVAC) STANDING CAGE IN LATE FEBRUARY OF 1995.

NOTE OIL PRODUCTION INCREASE (317%

	TOTAL OIL PROD	TOTAL DAYS	OIL BPD	WATER BPD	GAS Mcf/D
NOV	904.3	30	30.14	168.3	16.7
DEC	933.2	31	30.10	164.9	12.9
JAN	956.5	31	30.85	169.3	15.7
FEB	786	26	30.23	169.0	13.8
MAR	2755	29	95.00	359.4	34.3
APR	2887.2	30	96.24	366.3	35.3
MAY	2411.4	28	86.12	382.7	33.0



INCREASE COMPARING AVERAGE PRODUCTION IN 1994 OF 29.1 BOPD TO MARCH THROUGH MAY 1995 OF 92.5 BBLS PER DAY IS **317%!!!**

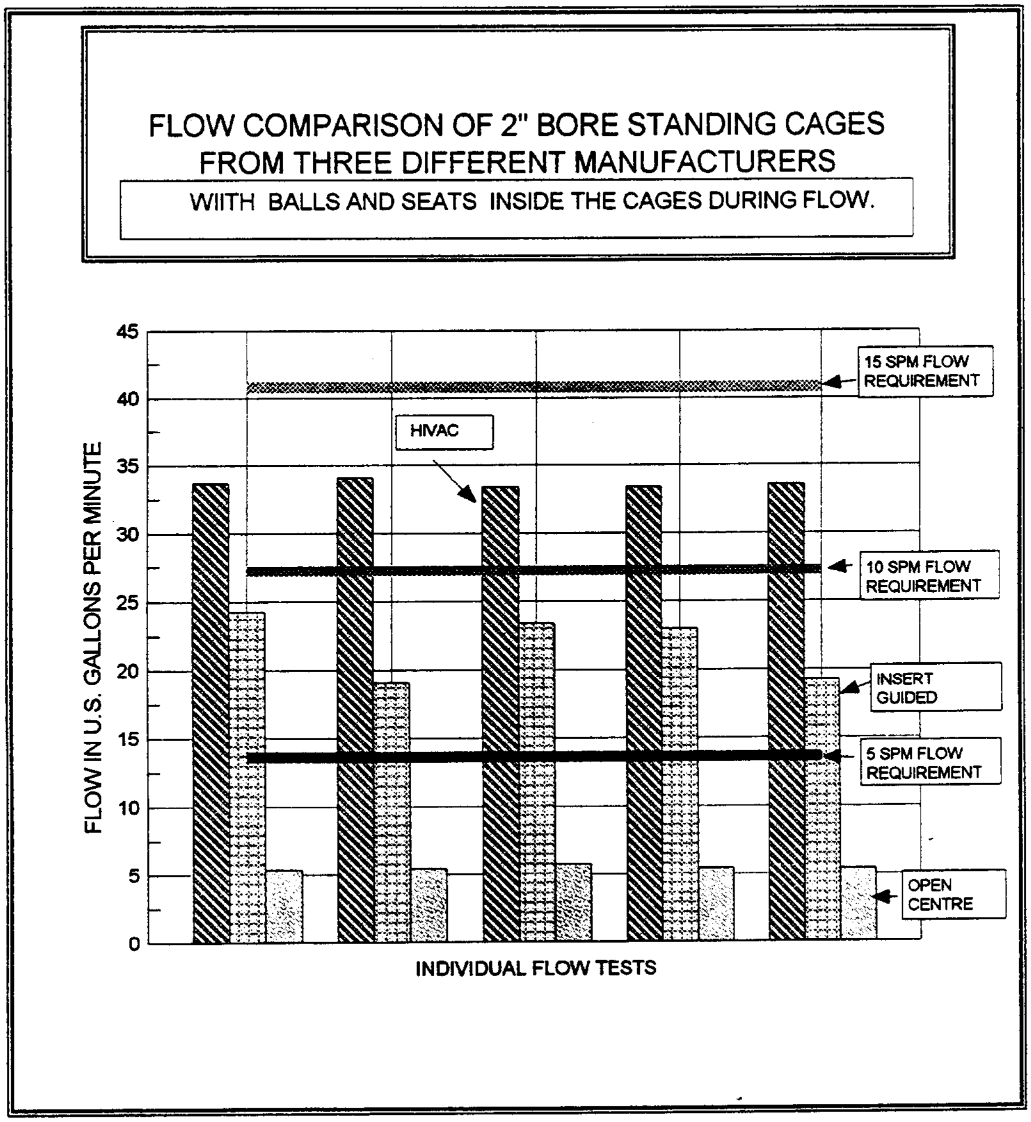
THE INCREASE OF 63.4 BARRELS OF OIL PER DAY GENERATES EXTRA REVENUE WHICH WILL EXCEED $114,000 U.S. EVERY 100 DAYS ASSUMING OIL TO BE VALUED AT $18.00 U.S. PER BARREL. THAT'S $1144 PER DAY EXTRA.

FLOW COMPARISON

THE GRAPH BELOW SHOWS FLOW RATE CAPACITY OF 2" PUMP BORE STANDING CAGES WITH BALLS AND SEATS INSIDE COMPARED TO THE FLOW RATE REQUIRED FOR PUMP DISPACEMENT ASSUMING A 100" STROKE LENGTH AT 5, 10, AND 15 STROKES PER MINUTE. WHEN STROKE LENGTH MULTIPLIED BY STROKES PER MINUTE EQUALS 1500 INDUSTRY CONSIDERS THIS TO BE MAXIMUM PUMPING SPEED. (IE; 70% OF ROD FREE FALL IN WATER)

CAGES WERE IN THE INVERTED POSITION WITH BALLS AND SEATS INSIDE. THE HIVAC CAGE HAD THE HI-VOLUME SEAT INSTALLED WHILE THE OTHER TWO HAD STANDARD API SEATS. ALL HAD 1-3/8" API BALLS.
WATER WAS USED AS THE TEST FLUID WITH APPROXIMATELY 18 INCHES OF HEAD.

BOTTOM HORIZONTAL BAR INDICATES FLOW RATE REQUIRED AT 5 SPM AND 100" SL
MIDDLE HORIZONTAL BAR INDICATES FLOW RATE REQUIRED AT 10 SPM AND 100" SL
TOP HORIZONTAL BAR INDICATES FLOW RATE REQUIRED AT 15 SPM AND 100" SL



THE GRAPH BELOW IS A RESULT OF DATA GATHERED FROM COMPRESSION TESTING 2" BORE RW STANDING CAGES FROM DIFFERENT MANUFACTURERS.

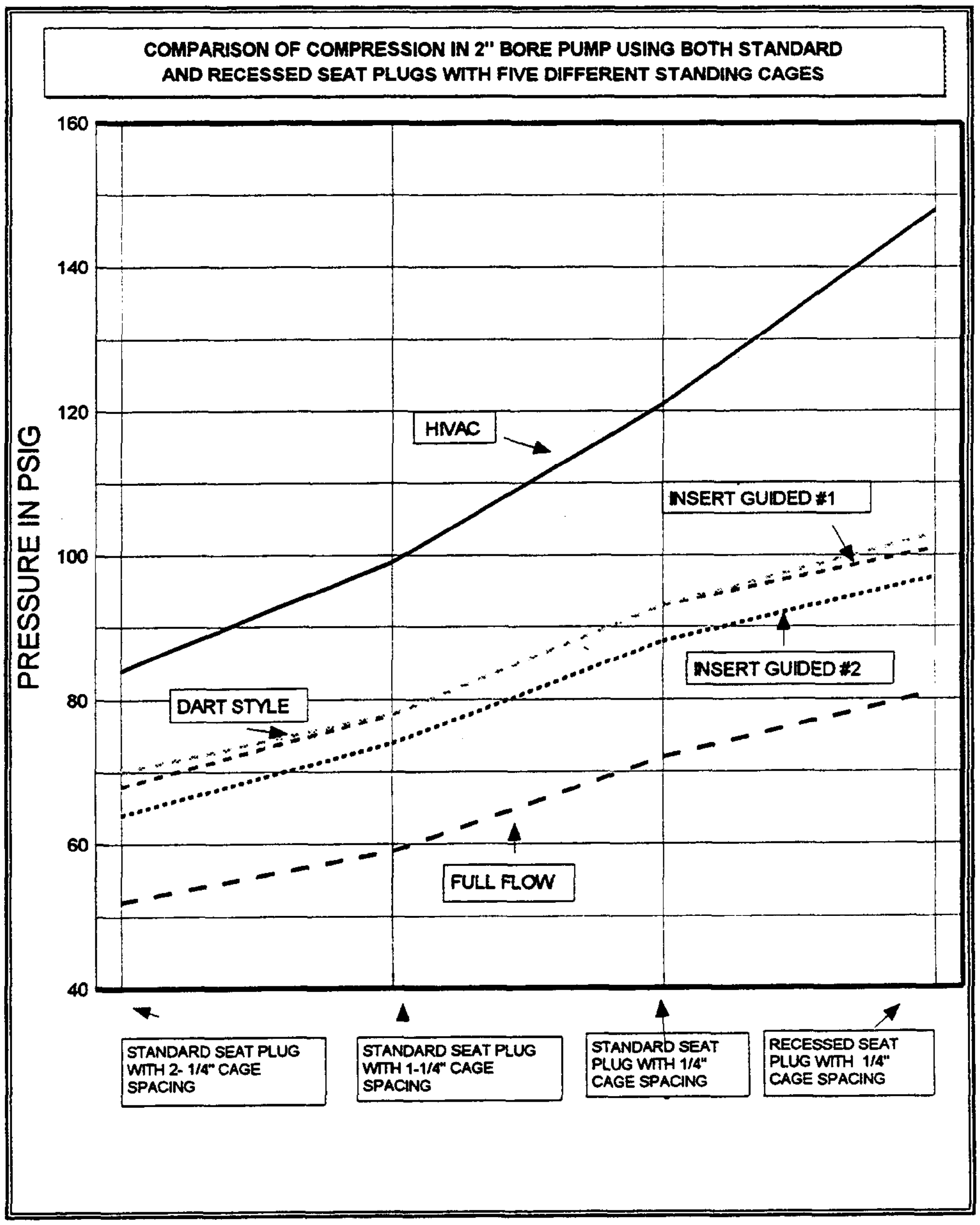
THE TESTS WERE DONE BY STROKING A ONE FOOT LONG PLUNGER INSIDE A 2" BORE RW BARREL TUBE WITH THE BOTTOM OF THE TRAVELLING CAGE COMING WITHIN 2-1/4"; 1-1/4"; AND 1/4" OF THE TOP OF THE STANDING CAGE.

THE TRAVELLING CAGE USED WAS OF API DESIGN WITH A BLANK DISC IN PLACE OF THE SEAT TO SIMULATE HYDROSTATIC LOAD ON THE TRAVELLING BALL WHICH WOULD KEEP THE BALL ON THE SEAT DURING THE DOWNSTROKE OF THE PLUNGER.

THE STANDARD API SEAT PLUG AND THE RECESSED SEAT PLUG WERE BOTH USED TO ILLUSTRATE THE IMPORTANCE OF CLOSE VALVE SPACING.

THE TESTS WERE DONE USING A 24" PLUNGER STROKE. ALL VALUES SHOWN IN THE TABLE AT RIGHT ARE IN PSIG.

	1 HIVAC C14-25	2 DART STYLE	3 INSERT GUIDED #1	4 INSERT GUIDED #2	5 FULL FLOW
STANDARD SEAT PLUG WITH 2-1/4" CAGE SPACING	84	70	68	64	52
STANDARD SEAT PLUG WITH 1-1/4" CAGE SPACING	99	78	78	74	59
STANDARD SEAT PLUG WITH 1/4" CAGE SPACING	121	93	93	88	72
RECESSED SEAT PLUG WITH 1/4" CAGE SPACING	148	103	101	97	81



CHEVRON CANADA RESOURCES NORTH KAYBOB

LOCATION	DATE INSTALLED OR CHANGED	SPM	AVERAGE TOTAL FLUID PER DAY BEFORE AND AFTER BHP CHANGE		AVG OIL PRODUCTION PER DAY BEFORE AND AFTER BHP CHANGE		AVG H2O PRODUCTION PER DAY BEFORE AND AFTER BHP CHANGE		AVG OIL AND H2O PRODUCTION INCREASE AFTER BHP CHANGED TO HIVAC VALVES		DATE SPM CHANGED	COMMENTS
			BEFORE	AFTER	BEFORE	AFTER	BEFORE	AFTER	OIL	WATER		
			M3	M3	M3	M3	M3	M3	M3	M3		
12-27	96/07/22	8.20	67.17	85.88	6.60	6.86	50.00	79.01	0.26	26.01		
02-04	96/07/31	9.50	14.86	25.87	6.60	9.58	8.49	16.20	2.98	7.71		
10-04	96/07/16	7.50	10.30	10.90	5.32	6.36	4.98	4.06	1.04	-0.92		
02-09	96/07/19	5.40	17.00	19.20	8.83	9.23	9.40	9.05	0.40	0.35		
04-09	96/10/25											WELL WORKED OVER. NO PREVIOUS INFORMATION
12-09	96/07/16	6.60	49.51	48.23	8.56	9.94	40.95	39.19	1.38	-1.76		
12-09		7.90	48.23	67.58	9.94	13.96	39.19	42.32	4.02	3.13	96/10/07	INCREASED SPM BY 1.3
04-11	96/12/20	6.60	24.73	36.00	4.50	8.00	20.00	28.00	3.50	8.00		
10-17	96/07/23	7.30	46.65	56.06	6.58	7.04	40.00	42.60	0.46	2.60		
12-21	96/05/14	6.60	9.90	37.67	3.80	13.30	6.44	24.50	9.50	18.06		
04-26	96/07/13	6.00	14.80	18.70	6.10	9.50	8.80	9.20	3.40	0.40		
10-28	96/07/13	8.50	81.00	67.00	15.40	18.50	57.00	50.80	3.10	-6.20		
15-28	96/10/26	4.00	13.20	37.20	6.50	7.30	8.40	12.00	0.80	3.60		
02-11	97/01/29	5.40	17.45	10.40	8.66	10.40	8.79	9.29	1.74	0.50		
04-11	96/12/20	6.80										WELL SLUGS..TOO SOON AFTER PUMP CHANGE
09-33	97/02/29	6.00	?	?	?	?	?	?	?	?		PUMP USED TO UNLOAD GAS WELL
14-10	96/10/16	7.50	2.41	3.63	2.38	3.63	0.03	0.04	1.25	0.01		
04-14	96/11/10	4.00	1.52	2.06	1.37	1.85	0.19	0.21	0.48	0.02		
03-15	96/08/08	6.00	2.90	3.15	2.89	3.12	0.03	0.03	0.23	-0.01		
09-15	97/02/27											WELL NOT YET TESTED
	TOTALS		**421.63**	**529.53**	**104.03**	**138.57**	**302.69**	**366.49**	34.54	**61.50**		
NET CHANGE M3				**107.90**		**34.54**		**63.80**				

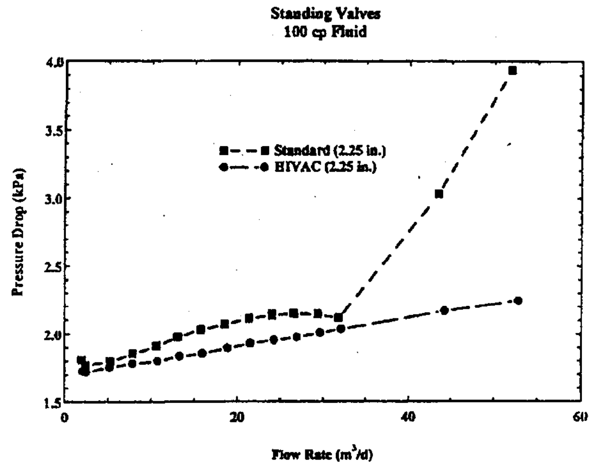
REVENUE FROM INCREMENTAL OIL INCREASE OF 34.54 M3 EQUALS $4,562.38 US ON A DAILY BASIS, ASSUMING OIL @ $21.00 PER BARREL.

ON A MONTHLY BASIS CONVERTED INTO CANADIAN DOLLARS, THIS EQUALS **$ 188,882**

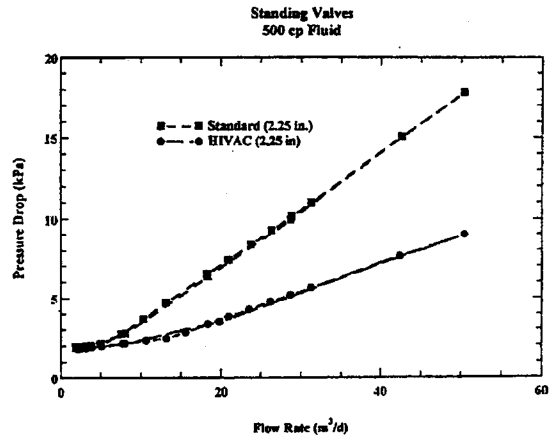
The above case history represents data from a field in Northwestern Alberta with the following operating conditions:

Contact person and telephone numbers are available to producers who require further information.

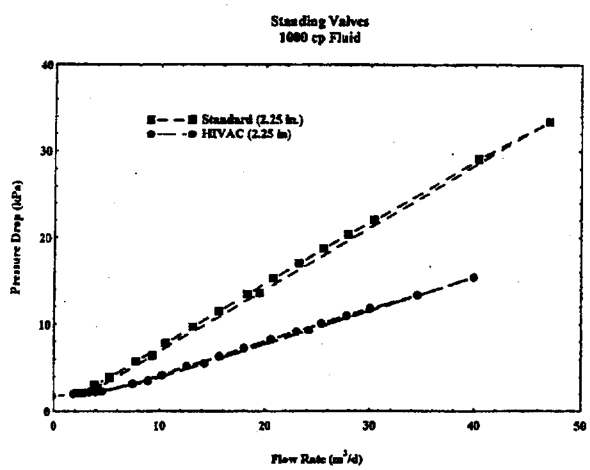
Depth	10,000 feet	3048 M
Temperature	240° F	115° C
GOR	1000:1 SCF/B	$180M^3/M^3$
GFR	100:1 SCF/B	$18M^3/M^3$
Water Sp. Gr.	1.06	
Water cut	75 - 95%	
Oil API	42	
H_2S	Trace	



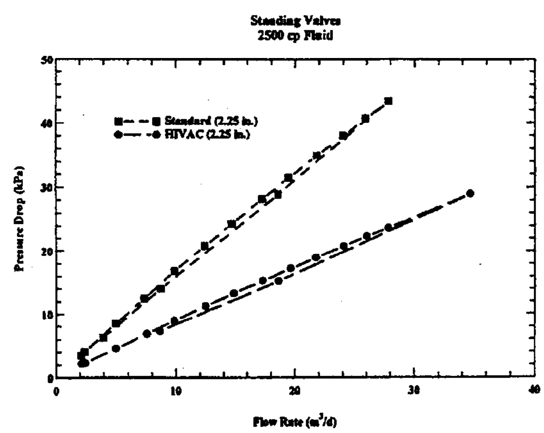
Pressure drop across standing valves with 100 cp fluid



Pressure drop across standing valves with 500 cp fluid



Pressure drop across standing valves with 1000 cp fluid



Pressure drop across standing valves with 2500 cp fluid