BEAM PUMP ROD BUCKLING AND PUMP LEAKAGE CONSIDERATIONS

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INTRODUCTION

Pump slippage occurs primarily on the upstroke, leaks through the plunger-barrel interface, and serves to lubricate the plunger-barrel action. Usual industry minimum requirements are quoted to be in the range of 2-5% of production to provide lubrication. This figure could come under scrutiny as well as slippage becomes better defined. If the slippage is too large, then the system becomes inefficient. This can be due to a worn plunger-barrel, and/or traveling valve, or to sizing the pump with an excessive clearance.

Recent tests[1-3] indicate that older equations (i.e., References 4-5) used by industry for years may have over-predicted the slippage. If verified, then pumps can be sized with larger clearances, reducing plunger-barrel friction, and possibly eliminating some of the compression at the bottom of the rod string yet still not allowing excessive leakage. In this paper, a short derivation of the slippage equation is developed which also provides an indication of the contribution to slippage due to plunger upward velocity.

Since buckling considerations may arise from pump clearance and other factors, rod buckling equations are presented and reviewed. Rods buckle due to outside forces acting to compress the bottom of the rod string on the downstroke. It is well known[6] that buoyancy forces do not contribute to buckling. Thus only negative "effective" forces excluding buoyancy contribute to buckling and "true forces", that include buoyancy, should not be used in buckling equations. An example of how to calculate the rod projected area and buoyancy induced pressure forces is presented and examples of calculating "true forces" and "effective forces" are presented.

Contributing factors to rod buckling include the force to slide the plunger in the barrel and also the pressure drop across the traveling valve as the pump travels downward. These forces are calculated and examples are presented. The compressive forces predicted to initiate buckling on the bottom of the rod string range from about 20-150 lbfs for rod sizes from sizes 5-10. Also fluid pound is discussed relative to rod buckling.

PLUNGER- BARREL SLIPPAGE

Relatively recent work[1-3] has shown that traditional equations for slippage[4-5] greatly over predict the plunger-barrel leakage, especially at larger diameter clearances of over 0.01".

Appendix A contains a short derivation of the traditional version of the leakage expression contrasted against the new leakage expression derived from recent test work. Also shown in Appendix A is the traditional leakage term but also with an additional term which accounts for the plunger velocity on the upstroke. This effect is small but shows plunger velocity increases the slippage.

Slippage is approximately constant regardless of the pump speed. Another way of saying this is that slippage is a bigger percentage of the production at low rates than it is at high pumping rates. Additional testing on plunger slippage considering a large number of variables is under way at the Texas Tech test well in a project sponsored by several interested operators including use of some donated equipment and services from a number of companies.

ROD BUCKLING IN RODS OVER THE PUMP

When studying pump slippage due to pump clearances, it is clear that tighter clearances have an effect on what compression the rods experience on the downstroke. Appendix B lists the equations showing what compressive forces are required for buckling rods over the pump. Rod buckling is not necessarily catastrophic. When rods do buckle, side forces between the rods and tubing occur that wears both components.

Forces That Cause Buckling

Appendix B shows the magnitude of compressive forces that initiate buckling in sucker rods. Forces at the bottom of the rod string are generated by some pumping conditions at the pump. These forces are considered as outside forces and would contribute to buckling.

However forces due to fluid buoyancy would not contribute to buckling[6]. Appendix C shows how the projected areas of rods can be calculated to account for the rod upsets.

A formula that allows one to determine what forces will cause buckling and what forces will not at any point in the rod string is $Teff = T_{true} + P_oA_o$. T_{true} is the actual axial tension force seen by the rod string and includes buoyant forces. This is the force that would relate to strain gage measurements on a sucker rod string. Rod stresses are calculated with the T_{true} force.
Teff is a mathematical artifice that accounts for the force of the hydrostatic pressure P_o acting on the rod string. Side forces from P_o oppose buckling and act to restore the rod to its initial unbuckled condition.

This equation allows one to determine if a compressive force on a rod string is sufficient to cause buckling. Appendix D shows how the true and effective forces are distributed in a sucker rod string hanging statically in the wellbore. The total forces on the upstroke and the downstroke are calculated by adding the dynamic forces to the static forces and these are forces that are usually reported in various computer program results.

If there is a negative Teff at the bottom of the bottom rod, then this negative Teff is the force to compare with buckling criteria (Appendix B) to determine if the bottom region of the bottom rod will buckle. Actually if a negative Teff occurs at any other place in the rod string other than at the bottom, then the same criterion can be used to determine if buckling might be occurring elsewhere in the rod string.

PLUNGER VISCOUS DRAG FORCE

One force that contributes to buckling is the force required to push the plunger through the barrel on the downstroke. The viscous drag force required to do this is calculated in Appendix E. It is a viscous concentric calculation. It shows that the force to push the plunger through the barrel with small values of viscosity is small. This force would be considerably larger if a high viscosity fluid is being pumped or if sand/scale/trash are present.

FLOW THROUGH THE TRAVELING VALVE

Another force that contributes to buckling is the force due to the pressure drop of fluids passing through the TV (traveling valve) on the downstroke. This force is calculated in Appendix F. The calculations show that for faster speeds of the plunger on the down stroke (velocity$>\approx$ 5-6 fps) leads to forces that are predicted to be in the range that would buckle the bottom rods in the sucker rod string. This points to a good operational practice of using the highest flow through TV assembly or the TV assembly area that generates the least pressure drop. Especially when pumping at approximately 6-7 SPM and higher.

FLUID POUND

Most discussions on beam pumping usually indicate that the rods buckle when fluid pound occurs. Many times this discussion is accompanied by a drawing showing the rods buckling when the plunger hits the gas-liquid interface in the barrel. However schematics of the bottom hole dynamometer (calculated or measured) show a card that shows no unusual compression occurring due to fluid pound. One measured card that does show compression in the rods on the down stroke is presented in Appendix G but this does not seem to be the norm. The conclusion is that often either that fluid pound induces a shock load in the rods but does not cause compression, or that compression occurs but so quickly, the results of this force are not recorded by the dynamometer equipment.

CONCLUSIONS

The effects of plunger speed on plunger-barrel leakage are developed and calculated. The overall expression for slippage is compared to recent test data that shows much less leakage or slippage than traditional expression have shown. More testing on slippage is under way.

Discussion is presented to re-iterate that forces from fluid buoyancy have zero effect on possible rod buckling. The distinction between true forces and the artifice of effective force is discussed. Effective forces should be examined to determine if forces are contributing to buckling. True forces should be used to calculate rod stresses. See Reference 6 for acomplete derivation and discussion of these effects.

Rod buckling due to forces at the pump is a result of plunger-barrel clearance and also due to other forces. An analysis of the force to push the plunger in the barrel shows the force to be small. It would be large with solids present. An analysis of the force required to push the plunger down with fluids flowing through the TV is presented. It shows that this force can be large enough to buckle the rods above the pump. The TV assembly flow-through area should be as large as possible, especially when pumping at higher speeds.

Fluid pound is discussed and it is observed that most calculated or measured downhole pump dynamometer cards do not show forces that would tend to buckle the rods as a result of fluid pound. This seems contrary to most discussion on the subject, but not contrary to most dynamometer evidence usually presented for the fluid pound effect.

REFERENCES

1. Patterson, J.C. and Williams, B.J., A Progress Report on "Fluid Slippage in down-Hole Rod-Drawn Oil Well Pumps", presented at the Southwestern Petroleum Short Course, April, 1998.
2. Patterson, J.C. et al, Progress Report #2 on "Fluid Slippage in Down-Hole Rod-Drawn Oil Well Pumps, presented at the Southwestern Petroleum Short Course, April, 1999.
3. Progress Report #3 on "Fluid Slippage in Down-Hole Rod-Drawn Oil Well Pumps" John Patterson, Jim Curfew, Mike Brock, Dennis Braaten, Jeff Dittman, ARCO, Benny Williams, Harbison-Fischer, presented at the Southwestern Petroleum Short Course, April, 2000.
4. Glenn M. Steams a progress report, "An Evaluation of the Rate of Slippage of Oil Past Oil-Well Pump Plungers," Drilling and Production Practice, 25-33 (1944)
5. "Minimizing Slippage in Subsurface Pumps," Petroleum Engineer, April 1960, by R.W. Reekstin.
6. Newman, K., Bhalla, K., "The Effective Force", CTES L.C., Conroe, TX. Technical Note, Jan 13, 1999
7. Eickmeier, J. R., "Applications of the Delta II Dynamometer Technique", 17th Annual Technical Meeting. The Petroleum Society of C.I.M., Calgary, May, 1994

APPENDIX A

Leakage Through the Barrel-Plunger Interface on Upstroke

Leakage is calculated assuming the plunger to be centered in the barrel and that both have uniform diameters. The leakage flows through the passage between the concentric plunger and barrel. The annular area of flow is approximated as flow between two flat plates. This is valid when the clearance is small compared to the inner diameter.The equations for flow of a viscous fluid between two parallel flat plates are taken from Schlichting (6th Edition, page 77) where the top plate (plunger) has velocity U and the plates are separated by distance h (the clearance).

$$u = \frac{y}{h}U - \frac{h^2}{2\mu}(\frac{dp}{dx})\frac{y}{h}(1-\frac{y}{h}) \qquad \text{(A1)}$$

$$\left.\frac{du}{dy}\right|_{y=h} = \frac{U}{h} + \frac{h}{2\mu}(\frac{dp}{dx}) \qquad \text{(A2)}$$

$$\tau_{y=h} = -\mu\frac{du}{dy} = -\mu[\frac{U}{h} + \frac{h}{2\mu}(\frac{dp}{dx})] \qquad \text{(A3)}$$

Where:

u	=	the local velocity at any y, ft/sec
U	=	the velocity of the top plate, ft/sec
h	=	the distance between plates, ft
x	=	the distance along streamline between plates, ft
μ	=	fluid viscosity, cp x .0000209 lbf-sec/ft^2
dp/dx	=	pressure gradient, lbf/ft^3
τ	=	shear stress, lbf/ft^2

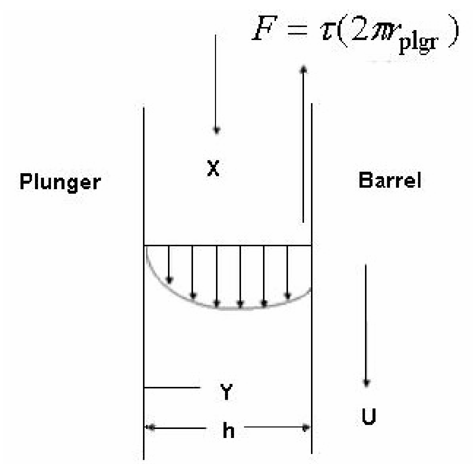


Figure A.1

In Figure A.1, the drawing is relative to being on the Plunger. Then on the upstroke as the plunger moves upward, the barrel appears and is moving downward relative to the plunger as above. The leakage is relative to the plunger, or what passes beneath the plunger is leakage from above the plunger to below the plunger.

Upstroke: Leakage Analysis:

$$\bar{u} = \frac{\int_0^h u(dy)}{h} = \frac{\int_0^h \frac{y}{h}U - \frac{h^2}{2\mu}(\frac{dp}{dx})\frac{y}{h}(1-\frac{y}{h})dy}{h} = \frac{U}{2} - \frac{h^2}{12\mu}(\frac{dp}{dx}) \ , \quad \text{ft/sec}$$

$$\bar{\mathrm{u}} = \text{average velocity through plunger} - \text{barrel interface, ft/sec}$$

$$\mathrm{A}_{\text{flow}} = \pi(do^2 - di^2)/(144x4) \quad , \mathrm{ft}^2$$

$$\text{BPD, leakage (rate on upstroke)} = \bar{\mathrm{u}} \text{ x } \mathrm{A}_{\text{flow}} \text{ x } 24 \text{ x } 3600 / 5.615$$

$$\text{BPD , leakage} \cong \text{above / 2 to account for only upstroke}$$

The pressure decreases in the direction of the positive U and X, so the pressure gradient is a negative value. Therefore the sign of the second term becomes a positive term when $\Delta p/L$ is inserted below for dp/dx:

Final form of leakage formula:

$$\text{const} = \pi / (144 \text{ x } 8) \text{ x } 24 \text{ x } 3600 / 5.615 = 41.96$$

$$\bar{u} = \frac{U}{2} + \frac{h^2}{12\mu}(\frac{dp}{dx}) = \frac{U}{2} + \frac{144(\mathrm{do}-\mathrm{di})^2 \Delta \mathrm{p}}{144. \text{ x } 12 \text{ x } .0000209 \text{ x } \mu \text{ x L}} = \frac{U}{2} + \frac{3991.67(do-di)^2 \Delta p}{4\mu L} \ , \ \text{ft/sec}$$

$$BPD, leakage = \mathrm{A}_{\text{flow}} \text{ x } \bar{\mathrm{u}} = \text{const x } (do^2 - di^2)\{\frac{U}{2} + \frac{3991.67(do-di)^2 \Delta p}{4\mu L}\}$$

$$\text{or : BPD, leakage} = 41.96 \text{ U D C } + 83745 \text{D C}^3 \text{ P} / \{\mu/\mathrm{L}\}$$

$$\text{With U in ft/sec, D in inches, C in inches, P in psi, L in ft, } \mu \text{ in cp :}$$

The first term above is for the leakage due to plunger movement and the second is for viscous leakage under pressure between the plunger and barrel.

SUMMARY

BPD, leakage = (41.96)UDC + (83745) $DC^3P/\mu L$ (A5)

where the first term is due to plunger velocity and the second term is for the viscous leakage due to the pressure across the plunger-barrel interface.

U	=	*velocity of plunger, up, ft/sec*
D	=	*average diameter, in, (do + di)/2 = D*
C	=	*diameter clearance of barrel ID – plunger OD, in, C= (do-di)*
P	=	*pressure difference across pump, psi (Δp)*
μ	=	*viscosity of fluid, cp*

L = *length of plunger, ft*

Leakage Example Calculations

D = 2.00 inches *Δp = P = 2000 psi* *μ = 3 cp*
C=2.008 – 2.00 = .008” *L = 4 ft* *U = 5 ft/sec*

Below is a velocity-at-pump plot from Qrod, supplied by Echometer for 5000’ 86 rod string, 2” pump, 144” stroke at 8 SPM showing 5-6 fps is not unusual.

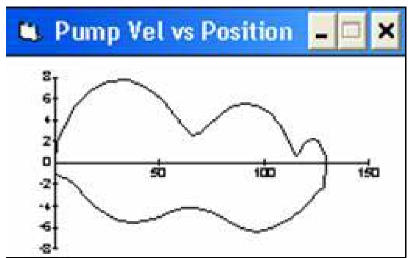


Figure A.2 - Example From Program Qrod Available From Echometer

The velocity U is the instantaneous pump velocity during the upstroke. An approximation might be the average velocity on the upstroke at the pump but better would be to break up any leakage calculation into time increments, each with the correct instantaneous pump velocity.

BPD,leakage= (41.96) UDC + (83745) $DC^3P/\mu l$
= (41.96)(5)(2.00)(0.008 + (83745)(2.00)($.008^3$)(2000)) / (3 x 4) = 3.36 + 14.3
= 17.65 bpd (note: plunger velocity contributes 19% of leakage for 5 ft/sec)
Qarco = 870 (2.00 inches) (2000 psi) $(.008)^{1.52}$/ (48 x 3 cp) = 15.7 bpd
(note: Qarco does not determine leakage from plunger velocity)
<u>For C = 0.01 inches clearance:</u>

C = .01 inches *Δp = P = 2000 psi* *μ = 3 cp*
D = 2.05 inches *L = 4 ft* *U = 5 ft/sec*

BPD, leakage= (41.96) UDC + (83745) $DC^3P/\mu l$
= (41.96)(5)(2.00)(0.01 + (83745)(2.00)($.01^3$)(2000)) / (3 x 4) = 4.19 + 27.9
= 32.1 bpd (note: plunger velocity contributes 13% of leakage for 5 ft/sec)
Qarco = (870)(2.00 inches)(2000 psi) $(0.01)^{1.52}$/ (48 x 3 cp) = 22.0 bpd

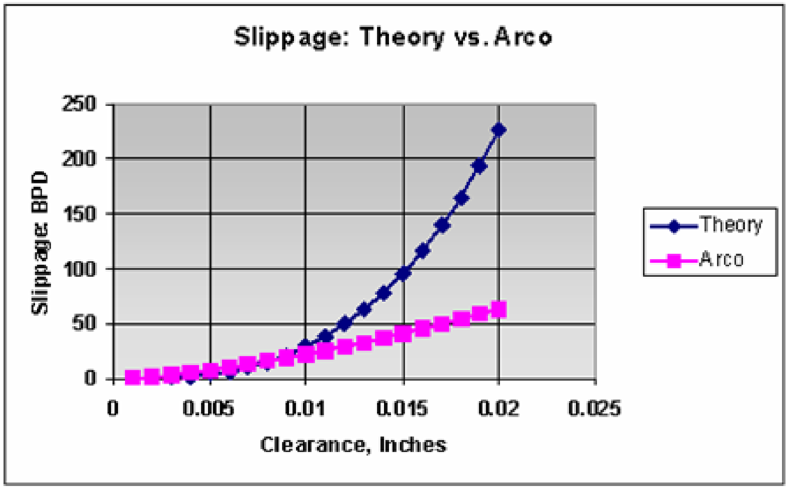


Figure A.4 - Plunger Leakage with Velocity =1 ft/sec

The plots change slightly with the velocity.

CONCLUSIONS

The flat plate theory is a good fit to the Arco equation up to about .01 in clearance. The theory here shows idea of what portion of leakage is due to plunger velocity using viscous data. However this classical model predicts that the faster the plunger velocity on the upstroke, then the more leakage.

Leakage is predicted to vary slightly with plunger velocity; however it is practically constant for different production rates. So in field trials, a high production rate would be predicted to have about the same leakage rate as a low production rate using this theory. Or in other words, leakage would be a much bigger percentage of a low production rate (given the same P and C and viscosity, and plunger length) than it would be of a high production rate.

Previous field trials seemed to indicate that slippage decreased as the rate increased, however could it be that for larger production rates, the slippage was just a smaller percent of the whole and not in experimental accuracy? Also there could be a large entrance loss as fluids enter the plunger-barrel interface from above and this could affect the results, but this is not modeled here.

APPENDIX B

Equations for Rod Buckling

Example Calculation for One Inch Rods :

$$w = w_{air}(1 - .128\gamma) = 2.9(1. - .128) = 2.5288 \text{ lb/f}t$$

$$I = \pi d^4 / 64 = \pi / 64 = .04908 in^4 E = 30x10^6 \; psi$$

$$Feff = -\{.795 EI(w\pi)^2\}^{1/3} = \qquad \text{(B-1)}$$

$$-\{\frac{.795 \;\; \pi^2 \;\; 30\text{x}10^6 \;\; 0.04908 \;\; 2.5288^2}{144 in^2 / ft^2}\}^{1/3} = -80.05 \text{ lbf}$$

$$L = \{.795\pi^2 EI / w\}^{1/3} \qquad \text{(B-2)}$$

$$= \{0.795\pi^2 \frac{30\text{x}10^6 \; .04908}{2.5288 \;\; 144}\}^{1/3} = 31.65 \text{ f}t$$

w is weight per foot in liquid E, steel, = 30 x 10^6 psi, I = moment of inertia of cross section, $\pi d^4/64$

Table B-1
Critical Force and Length for Rod Buckling

Size	W_{air}	W_{fluid}	Area	Lc	Fc
	Lb/ft	Lb/ft	in^2	Ft	(-)lbf
5	1.13	.9854	.307	23.16	22.82
6	1.63	1.42	.442	26.1	37.15
7	2.22	1.936	.601	28.96	56.07
8	2.9	2.528	.785	31.65	80.06
9	3.67	3.200	.994	34.26	109.6
10	4.53	3.950	1.227	36.74	145.1

Rourk, R. J., and Young, W. C., *Formulas for Stress and Strain*, McGraw-Hill, NY,NY, 1982, P. 539

APPENDIX C

Fluid Pressure Forces, Archimedes Principle, True/Effective Forces

Rod Area Correction Factor Ac

Assume a 1" rod string with upsets to depth L ft
Wra = Rod weight = 2.90 lb/ft in air
Ar = Area of 1" section = 0.7854 in^2 = .0054542 ft^2
ρ = density of steel = 487.6 lbm/ft^3

The Area Correction Factor Ac is the equivalent rod area required for the rod weight and accounts for the additional weight of the couplings.
(Ac)(ρ)(Ar)(L) = 2.904 L
Ac = (2.904 x L)/(ρ x Ar x L) = 2.904/(ρ x Ar) = 2.904/(487.6 x .0054542)
= 1.0919
In general,
Ac = Wra / (ρ x Ar) = Wra / (487.6 x .7854 x d^2/144) = Wra / (2.65945 x d^2)
Ac (7/8's) = 2.223 / (2.65945 x $.875^2$) = 1.0917
Ac (3/4's) = 1.633 / (2.65945 x $.75^2$) = 1.0916
Ac (5/8's) = 1.13 / (2.65945 x $.625^2$) = 1.0877
Where Ac multiplies the area of the body to correct projected area to account for the rod upsets.

Archimedes Principle
Depth: 25 feet
Wra = 25 feet x 2.904 for one inch rod = 72.6 lbf
Wrf = 72.6 (1.-.128γ) = 72.6 x .872 = <u>63.31 lbf</u>
or:
P@depth = .4333 x 25 = 10.8325 psi
A, one inch = .7854 sq in
Ac (to account for area of upsets) = 1.0919

F@depth = P x A x Ac = 10.8325 x .7854 x 1.0919
= 9.2897 lbf
Sum of forces = Wra – F = 72.6 – 9.2897 = <u>63.31 lbf</u>
<u>This says "Weight in air – sum of vertically projected pressure x area forces = Weight in fluid"</u>
<u>Also same as Wrf = Wra - wt of fluid displaced</u>
= 72.6 – 25 x.7854x1.0919x62.4/144 = 63.31 lbf

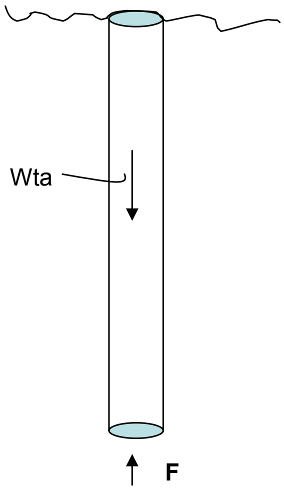


Figure C.1 - Rod in Fluid

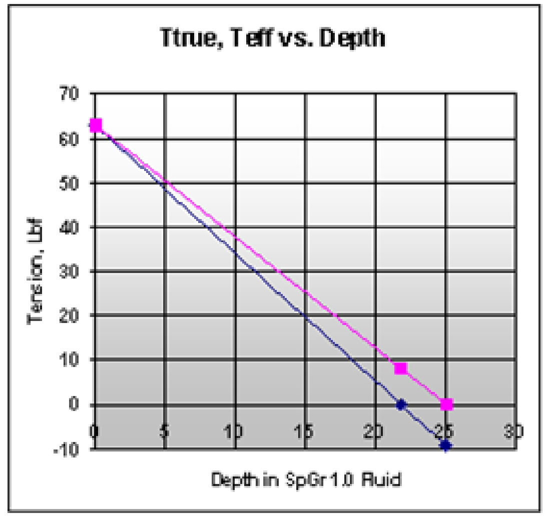


Figure C.2 -True and Effective Force

Distributions

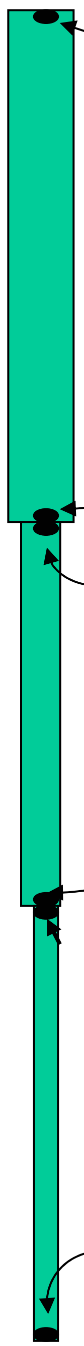
APPENDIX D

Static $_{True}$/Effective Loads on Submerged Tapered Rod String

Static True/Effective Loads on Tapered Rod String

Ptbg = 100 psi

Wrf = Wra(1.-.128γ) - PoAo = 11062.7 (1.-.128) -100*.7854*1.0906 = 9646.6 -85.65 = 9560.95 lbf



Ttrue = 5091 + 1539*2.904 = 9560.3 lbf

Teff = True + PoAo

Teff = 9560 + 100*.7854*1.0906 = 9645 lbf

Size 8
.7584 in^2
Ac=1.0906
2.904 #/ft
L=1539'

If PR 1/8 larger then

PRL = 9560 –100*(.994-.7854)1.09=9537 lbf

Not adding wt/ft x length up to load cell

ΔA = .7854*1.0906 - .601*1.0902 = .201 in^2
PoΔA = 766.387*.201 = 154 lbf
Ttrue = 5245.3 - 154 = 5091 lbf

Teff = True + PoAo
Teff = 5091 + 766.387*.7854*1.0906 = 5747.4 lbf

P = 100 + .433*1589 = 766.387 psi
Ttrue = 1697.4 + 1596*2.223 = 5245.3 lbf

Teff = True + PoAo
Teff = 5245.3 + 766.387*.601*1.0902 = 5747.4 lbf

Size 7
.601 in^2
Ac=1.0902
2.223 #/ft
L=1596'

ΔA = .601*1.092 - .442*1.095 = .1723 in^2
PoΔA =1457.5*1723 = 251.13 lbf
Ttrue = 1948.54 -251.13 = 1697 lbf

Teff = Ttrue + PoAo
Teff = 1697 +1457.5*.601*1.0902 = 2652.4 lbf

Size 6
.442 in^2
Ac=1.095
1.633 #/ft
L=1865'

Po = 100 + .433*3135 = 1457.5 psi
Ttrue = 1865*.442 – 1097 = 1948.54 lbf

Teff = True + PoAo
Teff = 1948.54 +1457 *.442*1.095 = 2654 lbf

At bottom, no external loads:
Po = 100 + .433*5000 = 2266.5 psi
Ao = .442 sq in, Ac = 1.095
PoAo =2266.65*.442*1.095 = 1097 psi
Ttrue = -1097 lbf

Teff = True + PoAo = -1097 + 1097 = 0.00 lbfs

Figure D.1 - Example of Calculating Static Effective and True Forces (Static Distribution)

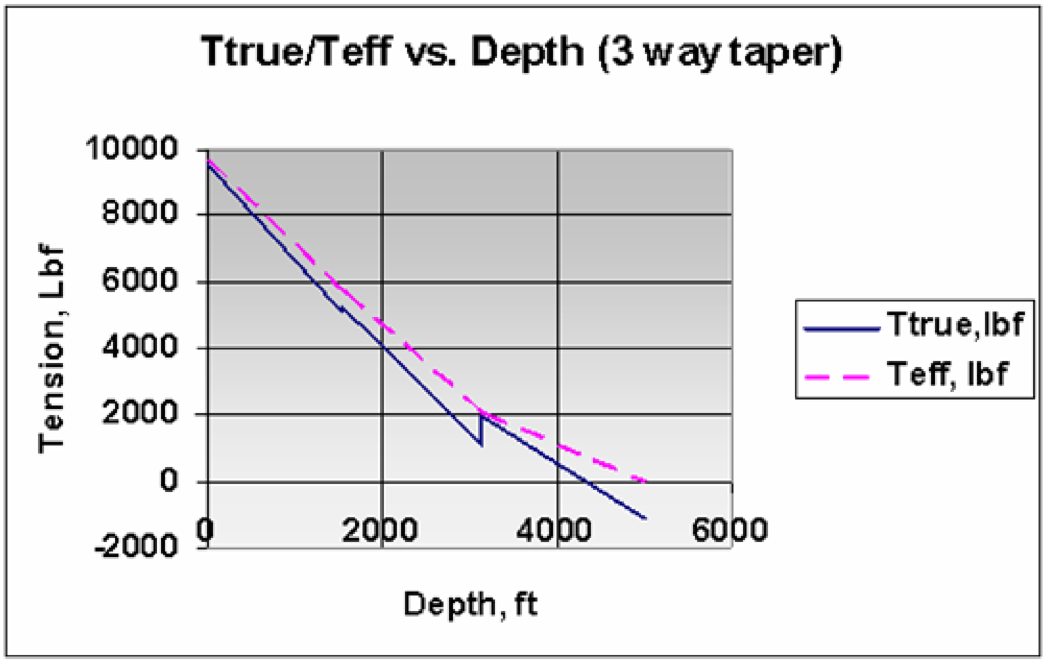


Figure D.2 - Plot of Effective and True Forces with Depth for 3-Way Taper

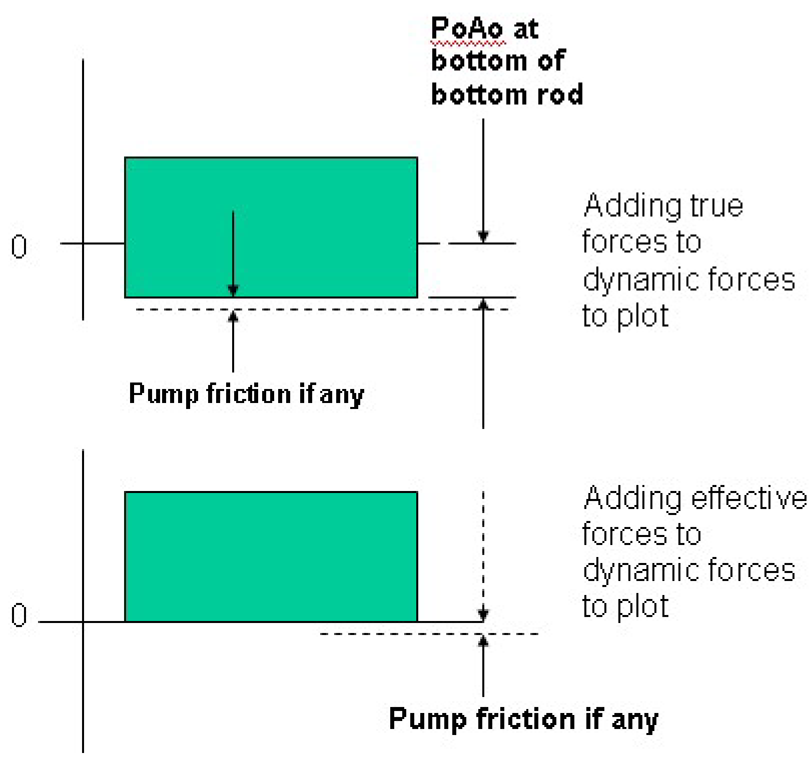


Figure D.3 - Two ways of plotting the bottom hole Dynagraph. Stress should be calculated by dividing true load by the actual body area.

Computer Model Output:

5000 deep, 2" pump, 86 rod string, 2" pump, 8 SPM, 100 psi surf pressure, ~1sp.gr.,

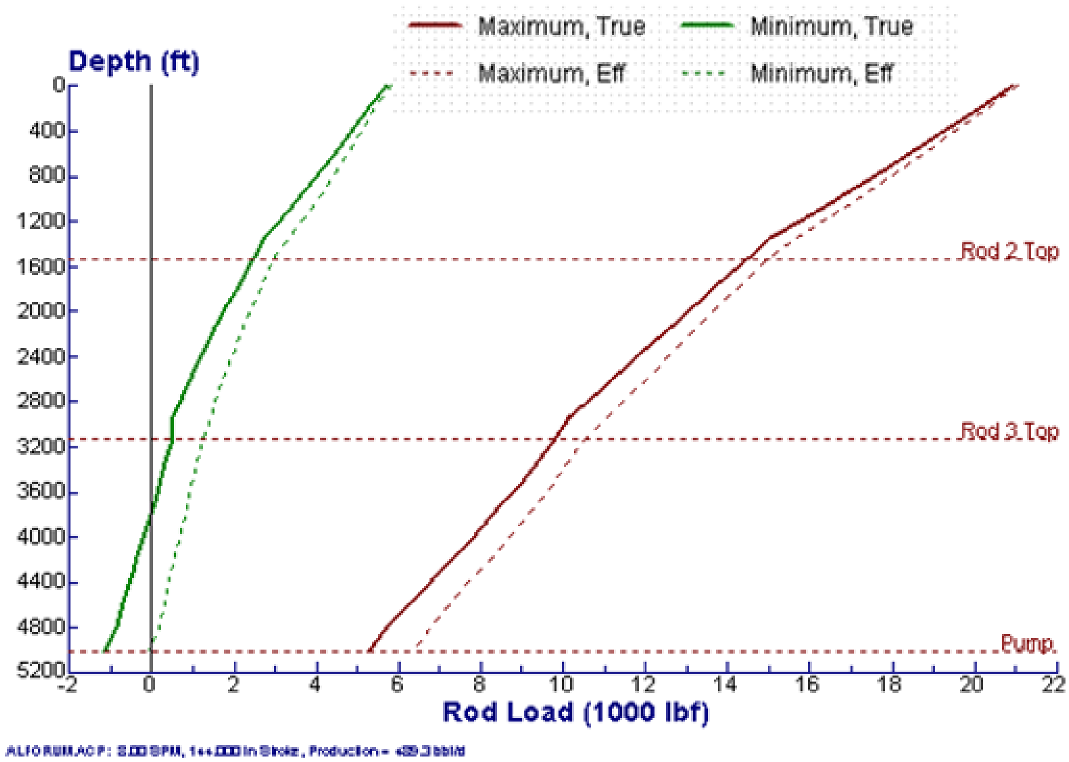


Figure D.4 - Computer Output of Maximum, Minimum True and Effective Forces vs. Rod String Depth

APPENDIX E

Plunger – Barrel Viscous Drag Analysis

See equations in Appendix A and Figure A.1.

Plunger Viscous Drag Analysis on Downstroke:

$Adrag = 2\pi(di/2)L$ is the area that fluid viscous drag acts on, ft^2

From above figure, U is positive downward, dp/dx is position, and x is positive downward.

Where:

U = velocity the plate (plunger surface), ft/sec

L = plunger length, ft

do = ID of barrel, ft

di = OD of plunger, ft

μ = Viscosity, cp x .0000209 lbf-sec/ft^2

F = viscous force along plunger that acts on bottom rod in upward, negative direction, on the downstroke

$$F = (Adrag)(\tau_{y=h}), lbfs \qquad \text{(E1)}$$

Example of viscous drag across the plunger on the downstroke.

<u>Example calculation:</u>

L= 4 ft

di =2.000 inches

do=2.008 inches

dp/dx = 62.4 psf/ft for the case of γ=1.00

μ = 3 cp x .0000209 = 0.0000627 lbf-sec/ ft^2

h= (do-di)/2. = (2.008-2.0000)/(2.x12.) = .000333 ft

U = 5 ft/sec down stroke velocity of the plunger

Adrag = 2πdiL/2 = 2 x 3.14 x 1.00 x 4.00 / 12 = 2.093 ft^2

$$\tau_{y=h} = -\mu[\frac{U}{h} + \frac{h}{2\mu}(\frac{dp}{dx})] \qquad \text{(E2)}$$

For do = 2.008" and h=.000333 ft, then:

τ = -.0000627{ 5/.000333 +.000333(62.4)/(2 x .0000627)} = -(0.941 + 0.01) = - 0.95 lbf/ft^2

Force = 2.093 ft^2 x (-.950 lbf/ft^2 = 1.99 lbfs drag acting up on down moving rods

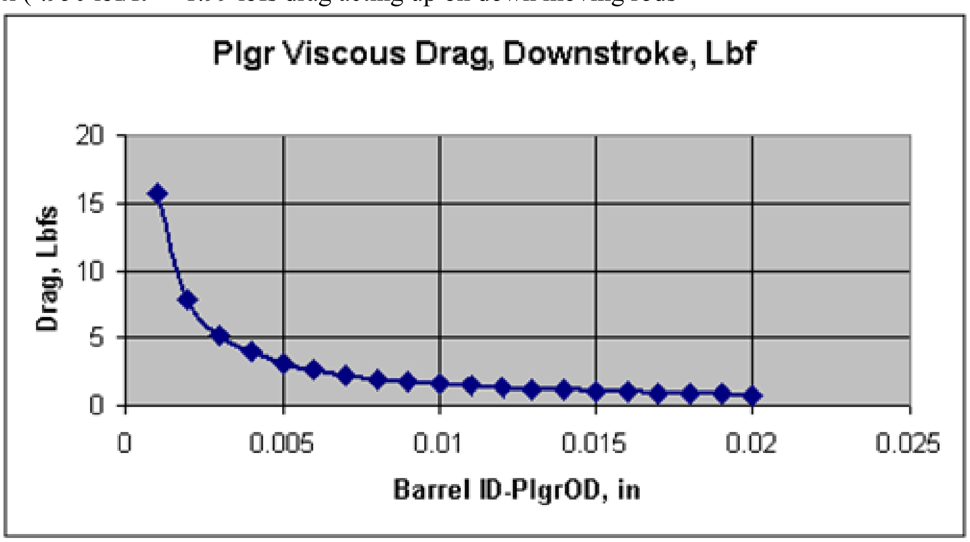


Figure E.1 - Plunger-Barrel Viscous Drag vs. Clearance for Velocity =5 ft/sec

Conclusion: From this analysis, there is little calculated viscous drag between the plunger and the barrel for the nominal conditions shown above.

APPENDIX F

Force Due to Fluids through the TV on the Downstroke

Pressure Drop across the Traveling Valve on the Down-Stroke:
A typical orifice equation that could predict this pressure drop is:

$$Q = CAo, ft^2 \sqrt{\frac{2\Delta p, \frac{lbf}{ft^2} \; g_c, \frac{lbm - ft}{lbf - \sec^2}}{\rho, \frac{lbm}{ft^3}}}, \frac{ft^3}{\sec} \qquad \text{(F1)}$$

Where:

Q, ft^3/sec
Ap = π (Dplgr)^2/ (4 x 144), ft^2
Dplgr = 2.0 inches
ρ =density of fluid, lbm/ft^3
gc = 32.2 lbm-ft/(lbf-sec^2)

Δp = pressure drop across TV, psf
C = discharge coefficient, use C $\approx$ 0.7 for calculations
Vel = velocity of the plunger, ft/sec (from wave equation)

Solving for a calculated pressure drop:

$$Q = (Vel)(Ao), ft^3 / \sec \qquad \text{(F2)}$$

$$\Delta p = (\frac{Q}{AoC})^2 \frac{\rho}{(2g_c)}, lbf / in^2 \qquad \text{(F3)}$$

Example calculation:
Instantaneous plunger velocity = 6.0 ft/sec
Dorifice = 0.84" ρ = 62.4 lbm/ ft^3 C = .7
Ao = π x .84 x .84 / (144 x 4) = 0.00384 ft^2
Q = Vel x Aplgr = 6 x 3.14 x 2 x 2 /(144 x 4) = 0.1308 ft^3/sec
Δp, psf = $(0.1308/(0.7 \text{ x } .00384)^2$ 62.4 /(2 x 32.2) = 2294 lbf/ft^2 <u>or 15.9 psi</u>
Force across TV on plunger area = π 2^2/4 in^2 * 15.9 lbf/ in^2 = 49.93 lbfs
This would be enough to buckle a size "6" rod.

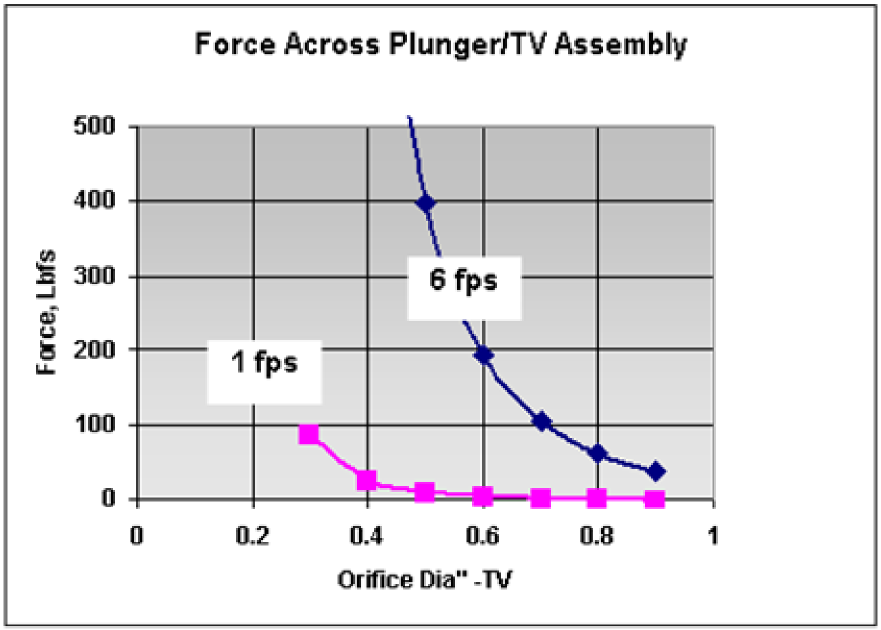


Figure F.1 - Force Due To Flow of Fluids Through TV for 1 & 6 fps

Below are the inside diameters of the seats for the traveling valves The smallest flow area in the cage is around the ball and it is slightly less (maybe 20% less) than the flow area through the seat. (*Courtesy: Bennie Williams, HF*)

Typical seat inside diameters for traveling valves:

1.5 " pump	*.660" diameter*
1.75" pump	*.850" diameter*
2.00" pump	*.940" diameter*
2.25 " pump	*1.070" diameter*

<u>SUMMARY:</u>
Above if the pump was a 2.00" pump, and the flow area is 20% less than 0.850 diameter would calculate, then the effective diameter would be 0.84". In the above plot, the upward force would be about 52 lbfs, for 6 fps. Conclusion: This could be of concern for buckling for higher plunger velocities on the downstroke and small orifices. The flow through the TV can create enough pressure drop across the plunger area to be of concern for buckling.

APPENDIX G

Fluid Pound vs. Rod Compression

" The dynamic loads during fluid pounding can cause several detrimental effects on the downhole equipment.: The rod string can experience buckling that leads to rod breaks,; rod-to-tubing wear is increased; shock loads contribute to coupling failure due to unscrewing; and pump parts can be damaged (as well as tubing), if unanchored. On the surface , shock loads can damage pumping unit bearings, and can lead to instantaneous torque that overloads the speed reducer." (Takacs)

Does fluid pound lead to compression or buckling in the lower rods?

Even in fluid pound, there is gas compression before the plunger hits fluid. Assume 100" bottom hole stroke. Assume ½ of the bottom hole stroke is filled with fluid and the other half with gas at 100 psi. Assume 2000 psi over the TV. Then:

$P_1/P_2 = 100/2000 = V_2/V_1 = (X \text{ x } A) / (50" \text{ x } A) =$

X = 2.5" or the TV will open when the TV is 2 ½ " above the fluid.

Fluid pound is usually shown in "cartoon" fashion as showing no unusual compression as a result of fluid pound. Usually in text describing fluid pound, it will indicate rod compression is a result of fluid pound. If fact it is nearly always shown as below and usually calculated or measured downhole cards actually look similarly.

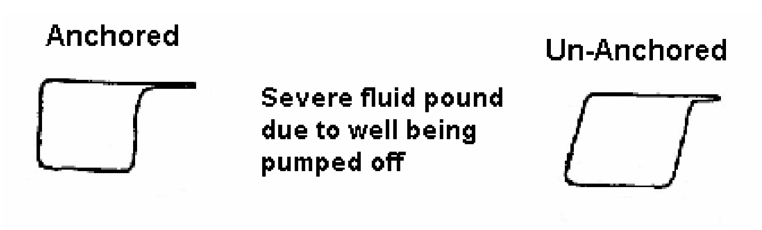


Figure G.1 - Cartoon of Fluid Pound Forces at Pump

However Figure G.27 indicates a compression spike due to fluid pound. This type of bottom hole card is usually not shown with a fluid pound situation. Usually the downward spike, indicating compression to the rods over the pump, is not present.

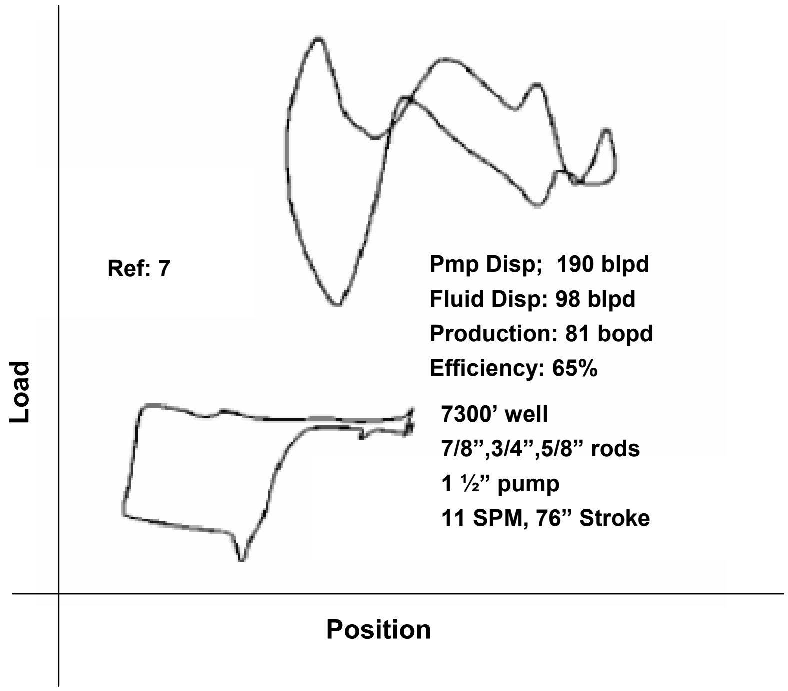


Figure G.2 - Fluid Pound Showing Compression Spike at the Pump

The conclusion might be that fluid pound may or may not cause rod compression. In most bottom hole dynagraphs (measured or calculated), no excess compression due to fluid pound is shown. Is the duration of such a spike so short that it is not recorded? This seems doubtful since spikes due to hitting up or down due to poor pump spacing are usually recorded.