QUANTITATIVE DETERMINATION OF ROD PUMP LEAKAGE

USING DYNAMOMETER TECHNIQUES

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ABSTRACT

New methods for quantifying pump leakage are described. These methods extend the effectiveness of dynamometer analysis and can lessen the dependence on well tests in determining the degree of pump leakage. Actual examples are included to illustrate the techniques. A detailed description of traveling and standing valve leakage checks is also included.

INTRODUCTION

The polished rod dynamometer has many important uses in determining surface equipment loads, calculation of downhole pump dynamometer cards and visual diagnosis of downhole pump conditions. It is also useful in determining the mechanical condition of the downhole pump from simple and inexpensive measurements taken at the surface.

All downhole rod pumps employ barrels (cylinders), plungers (pistons) and traveling and standing valves (normally balls and seats). As the names suggest, the traveling valve "travels" up and down with the rods and the standing valve "stands still" with respect to the tubing. Figure 1-a illustrates the primary parts of the two types of API pumps, the rod pump and the tubing pump. With the tubing pump, the barrel is an integral part of the tubing string allowing the use of a larger plunger when compared with rod pumps. With the rod pump, the entire mechanism (including the barrel) is run inside the tubing on the rods. There are three basic rod pump configurations which are (1) top anchor, (2) bottom anchor with traveling barrel and (3) bottom anchor with stationary barrel. Regardless of the type or configuration, basic pump operation is the same.

This paper presents the theory of traveling and standing valve checks with a dynamometer and documents three quantitative methods for deriving pump slippage rates in barrels per day.

TRAVELING VALVE CHECK

The traveling valve check tests the integrity of the plunger/barrel fit and traveling valve ball and seat. According to Stearns[1], leakage past a plunger can be calculated and depends on the physical dimensions of the pump, viscosity and density of the fluid and differential pressure across the plunger. Because the fit is known only when the pump is installed, the leakage rate is unknown as

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the barrel and plunger wear and the fit becomes poorer. Also, leakage can occur through the valves as the balls and seats wear. Thus, as wear progresses the need for quantifying pump leakage becomes important. Well tests can indicate leakage but tests are often inaccurate and infrequent. Changes in the well's productivity can also cloud the issue.

The traveling valve check is usually made high in the upstroke as shown in Figure 1. The traveling valve is closed and the standing valve is considered as open although the weight of the ball will cause it to rest lightly on seat as long as traveling valve leakage is insignificant. The dynamometer at the surface is sensing the traveling valve load Q_t which is comprised of buoyant rod weight Q_r, fluid load Q_f and upstroke friction Q_u. Analytically this is expressed as

$$Q_t = Q_r + Q_f + Q_u \tag{1}$$

The fluid load is defined as

$$Q_f = A_p\ (P_a - P_b) \tag{2}$$

The pressure above the plunger P_a is caused by the hydrostatic head of fluid in the tubing over the pump. Pressure below the plunger P_b is initially that of pump intake pressure minus the (usually) small pressure drop across the standing valve suffered as fluids enter the pump (Figure 1-a). The fluid load has stretched the rods by an amount δ_r. Rod/tubing drag friction acts downward because the traveling valve test is made by stopping the rods on the upstroke and friction acts in the direction opposite to the motion. Fluid frictional effects vanish when the rods stop but residual drag friction effects are normally locked-in even with the rods at rest. Downward acting drag friction increases the load sensed by the dynamometer (note the positive Q_u term in equation 1).

The pressure difference across the plunger forces fluids to migrate between the plunger and barrel and between the traveling valve ball and seat. When enough fluid has migrated from above to below the plunger to equalize the pressure difference (render $P_a = P_b$), the fluid load Q_f vanishes (Figure 1-b). The rods contract by the amount δ_r and the dynamometer registers the load

$$Q_t = Q_r + Q_u \tag{3}$$

With all of the fluid load removed from the rods, the standing valve is held tightly on its seat by the pressure difference $P_a - P_i$. All of the fluid load has been transferred from the traveling valve (rods) to the standing valve (tubing).

Modern digital-electronic dynamometers record both position and load versus time. Figure 2 shows a valve check with such an instrument wherein the unit has been brought to a halt about 90% into the upstroke. The horizontal position trace beginning at point S signifies that rod position is not changing (the unit is held motionless with the brake). Load trace a-b suggests significant leakage. In Figure 3 the load trace c-d suggests negligible traveling valve leakage. The analyst seeks to relate the severity of pump leakage with the rapidity of load loss. Later a quantitative relationship is developed to determine leakage rate (barrels/day) in terms of traveling valve load loss rate (pounds per second). A quantitative relationship is needed because the same load loss rate can mean severe leakage in one well, yet insignificant leakage in another.

STANDING VALVE CHECK

This check is made by stopping the unit in the downstroke. The standing valve is held tightly closed by the pressure difference $P_a - P_i$. The traveling valve is considered as open although the ball is resting lightly on seat because of its own weight (as long as standing valve leakage is negligible).

The unit is shown stopped deep in the downstroke (Figure 4-a). The surface dynamometer is registering the standing valve load Q_s

$$Q_s = Q_r - Q_d \tag{4}$$

All of the fluid load is being supported by the standing valve (tubing). The dynamometer only feels the buoyant weight of the rods and the effects of downstroke rod/tubing drag friction Q_d. Note the negative Q_d term in Equation 4 which indicates that rod friction is acting upward, making the rods seem lighter.

If the standing valve is defective, the pressure difference across the standing valve will eject fluid from the tubing. This creates a pressure difference across the traveling valve and causes the fluid load to be transferred onto the traveling valve (rods). This stretches the rods and eventually the dynamometer will read

$$Q_s = Q_r + Q_f - Q_d \tag{5}$$

Please refer to Figure 4-b. The rate at which fluid load is transferred from the standing valve (tubing) to the traveling valve (rods) is taken to be a measure of severity of standing valve leakage.

Figure 5 illustrates a standing valve check made with a digital-electronic dynamometer which records position and load versus time. The standing valve check suggests a rapid leak. As shown by the load trace a-b, the full fluid load has been transferred from the standing valve onto the rods in less than 4 seconds.

VALVE CHECK DATA GATHERING TECHNIQUES

Good data gathering techniques in the field are required to obtain valid valve checks. The unit must be stopped gently to minimize dynamic effects. Figure 6 shows the effects of stopping the unit too rapidly such that standing valve checks SV1 and SV2 and traveling valve checks TV1 and TV2 are invalid. The standing valve checks are invalid because some of the fluid load is still applied. The traveling valve checks are improper because some of the fluid load has been lost. Because of more gentle stops (see position traces), SV3 and TV3 are good valve checks. Two or three gentle stops during the upstroke (traveling valve checks) and two or three gentle stops during the downstroke (standing valve checks) are preferred. When making traveling valve checks, the first stop should be at least halfway into the upstroke to insure all rod stretch has occurred and the full fluid load is on the rods. By making several stops on the upstroke, the plunger/barrel fit can be investigated at several points along the barrel which can identify uneven barrel wear or a split barrel.

When making standing valve checks, the first stop should be at least halfway into the downstroke. For pumps with poor liquid fillage, the final stop should be near bottom of the stroke to insure all of the fluid load is transferred to the standing valve.

It is very important that all of the fluid load be initially registered on the traveling valve check and all of the fluid load be initially removed on the standing valve check. This leads to maximum pressure differences which are needed in the quantitative theory.

STATIC STRETCH PROPERTIES OF ROD STRINGS

Rod stretch plays a vital role in the quantitative theory of pump leakage derived from valve checks. Static type formulas for rod stretch are used for simplicity. This requires that the valve checks be taken gently to minimize dynamic effects.

For tapered strings (including the case of combination strings of fiberglass and steel), the formula for stretch is

$$\delta_r = Q_f \sum_{i=1}^{m} \frac{X_i}{E_i A_i} \tag{6}$$

An elastic constant E_r is defined in practical units

$$\delta_r = Q_f \, L_r \, E_r \tag{7}$$

where

$$L_r = \sum_{i=1}^{m} X_i ; \quad E_r = \frac{12}{L_r} \sum_{i=1}^{m} \frac{X_i}{E_i A_i} \tag{8}$$

For steel rod designs using API rod tapers; E_r can be obtained from API RP11L[2].

METHOD I

Pump Leakage From Initial Load Loss Rate

In this method, reference is made to the traveling valve check although the method is equally applicable to determination of standing valve leakage rate. Refer to Figure 1 and note that rod stretch caused by the fluid load completely disappears when enough fluid migrates from above to below the plunger to equalize the pressures. The volume of this amount of fluid is exactly

$$\text{Volume} = \frac{\pi}{4} d^2 (\delta_r + \delta_t) \tag{9}$$

wherein the most general case of unanchored tubing is considered. This volume is leaked in time t so that

$$\bar{R} = \frac{\text{Volume}}{\Delta t}$$

If the volume leaked per unit of time $\bar{R}$ is expressed in barrels per day, we have in practical units

$$\bar{R} = \frac{6.99 \; d^2 \; (\delta_r + \delta_t)}{\Delta t}$$

Substituting the relations for stretch results in

$$\bar{R} = 6.99 \; d^2 \; \frac{Q_f}{\Delta t} \; (L_r E_r + L_t E_t) \tag{10}$$

Note that the pump leakage rate $\bar{R}$ defined above is an average rate because the instantaneous rate decreases as the pressure gradient across the plunger diminishes. Figure 2 shows how the load loss rate from a-b decreases with time during the traveling valve check. The initial loss rate is maximum. The loss rate diminishes to zero when the fluid load is fully transferred to the standing valve (tubing). Traveling valve/plunger leakage occurs only while the traveling valve is closed and a pressure difference across it is applied. To account for this fact, a pump slippage coefficient C_p is defined which represents a weighted average of applied pressure differences and durations of application. The final leakage formula then results

$$R = 6.99 \; d^2 \; C_p (L_r E_r + L_t E_t) \left(\dot{Q}_t\right)_{max} \tag{11}$$

wherein $(\dot{Q}_t)_{max}$ is the maximum rate of traveling valve load loss. Since Q_f and Q_t only differ by a constant in the static case, Q_t can be substituted for Q_f in the load loss rate expression. The upward motion of the pump when fluid is being lifted tends to maximize pressure difference across the traveling valve. Thus it is important that maximum load loss rates are determined from carefully measured valve checks.

For full liquid fillage, a pressure difference is applied to the traveling valve for about 50 percent of the cycle time. Thus C_p is approximately 0.5 which leads to the practical formula

$$R = 3.5\, d^2\, (L_r E_r + L_t E_t)\left(\dot{Q}_t\right)_{max} \tag{12}$$

It is difficult to objectively determine (by eye) the initial load loss rate. The following procedure removes some of the subjectivity. Refer to Figure 7 and define 3 points on the load trace as follows:

Point	Time (sec)	Load (lbf)
1	0	Q_1
2	t_2	Q_2
3	t_3	Q_3

Pass a second degree polynomial through these points obtaining

$$Q(t) = a_o + a_1\, t + a_2\, t^2$$

in which Q(t) is polished rod load during the traveling valve check, t is time and the a_i's are coefficients to be determined.

The maximum rate of change of Q(t) occurs at point 1 (t=0) and is

$$\left(\dot{Q}_t\right)_{max} = \frac{dQ(t)}{dt} = a_1$$

where

$$a_1 = \frac{(t_2 + t_3)(Q_2 - Q_1)}{t_2 t_3} + \frac{t_2(Q_2 - Q_3)}{t_3(t_3 - t_2)} \tag{13}$$

The technique is illustrated with the data of Figure 7. An API 76 rod design is installed and a 1-1/4 inch pump is set at 6909 feet. From API RP11L, E_r is found to be 0.812×10^{-6} in/lbf-ft. The tubing is anchored so that $E_t = 0$.

Point	Time (sec)	Load (lbf)
1	$t_1 = 0$	$Q_1 = 14276$
2	$t_2 = 2$	$Q_2 = 12961$
3	$t_3 = 4$	$Q_3 = 12210$

$$a_1 = \frac{(2+4)(12961-14276)}{2(4)} + \frac{2(12961-12210)}{4(4-2)}$$

$$= -798 \ lb_f/sec$$

$$\left(\dot{Q}_t\right)_{max} = -798 \ lb_f/sec$$

Thus from equation 12

$$R = 3.5 \ (1.25)^2 \ (6909)(.812 \times 10^{-6})(-798)$$

$$\cong 25 \ bpd$$

Normal leakage according to Stearns[1] for these conditions is about 10 bpd. Thus an additional 15 bpd of production are being lost because of abnormal pump leakage. The traveling valve assembly and/or plunger-barrel fit is shown to be worn. In the actual well a downhole dynamometer card analysis indicates a liquid handling capacity of 148 bpd at stock tank conditions. This implies that 123 bpd (148-25) should be reaching the tanks. This result is in good agreement with measured production of 125 bpd. In summary, pump leakage has been inferred independently of the well test which gives the analyst yet another method of evaluating pump mechanical condition.

Two useful facts should be noted at this point.

1. In shallow wells with stiff rod strings and full liquid fillage, the traveling valve check can mislead the analyst into believing the traveling valve assembly or plunger fit relationship is defective when such may not be the case.

2. In deep wells with flexible rod strings, the traveling valve check can mislead the analyst into believing the pump is in good condition when the opposite is true.

These special cases (shallow and deep well examples) can be explained by studying the form of equation 11 which relates pump leakage to pump diameter, pump depth, rod stiffness and traveling valve load loss rate. As shown, different leakage rates would be calculated with the same load loss rate depending upon the stiffness of the rods. In the shallow well example, a large load loss rate would not necessarily imply a large leakage rate because the elastic constant is small (the rods are stiff). In the deep well case, a small load loss rate could imply a large leakage rate because the elastic constant is large (the rods are flexible). Figure 8 shows a nomograph for pump leakage based on equation 12. The tubing is assumed to be anchored at the pump.

The condition of the standing valve can be evaluated with the same method. Instead of using traveling valve load loss rates, standing valve load increase rates are employed.

METHOD II

Pump Leakage from Polished Rod Velocity

In order for the standing valve to open on the upstroke, the pump must be lifting fluid out of the tubing faster than a leak is allowing fluid to slip downward past the plunger and traveling valve assembly. The displacement rate depends on pump velocity and pump diameter according to the equation

$$D = 6.99\ d^2\ V(14) \tag{14}$$

If the traveling valve check is made gently and care is taken to make the unit ascend uniformly without undue jerking, polished rod velocity will approximate pump velocity. Figure 9 shows an actual traveling valve test made with a time based dynamometer. The unit is rising at a fairly constant rate and upward velocity of the unit is maintaining polished rod load fairly constant (except for small dynamic effects). Slightly later, a small decrease in upward velocity causes polished rod load to decrease. This is designated as point 1 and shows the time at which upward velocity is no longer sufficient to keep the standing valve open. This is the critical velocity V_c at which fluid lifting rate equals slippage rate. As before a second degree polynomial is applied to the position history with point 1 being the critical point as described above and points 2 and 3 being successively later position points. The critical velocity V_c can be calculated with equations similar to those derived previously

$$V_c = b_1$$

in which

$$b_1 = \frac{(t_2 + t_3)(Y_2 - Y_1)}{t_2\ t_3} + \frac{t_2\ (Y_2 - Y_3)}{t_3\ (t_3 - t_2)} \tag{15}$$

and in which the Y_i terms are polished rod positions corresponding to the various points shown on Figure 9.

The final form of the slippage formula based on polished rod velocity then results

$$R = 6.99\ d^2\ C_p\ V_c \tag{16}$$

As before, C_p is approximately 0.5 for full fillage and anchored tubing so that a simpler formula is

$$R = 3.5\ d^2\ V_c \tag{17}$$

An example will help clarify the procedure. From the data of Figure 9

$$b_1 = \frac{(1.21 + 2.42)(81.4 - 76.2)}{1.21(2.42)}$$

$$+ \frac{1.21(81.4 - 84.1)}{2.42(2.42 - 1.21)} = 5.3 \text{ in/sec}$$

hence

$$V_C = 5.3 \text{ in/sec}$$

$$R = 3.5\ (1.25)^2(5.3) = 29 \text{ bpd}$$

from equation 17.

This approach is useful when the load leaks off very rapidly, such as in a shallow well with a stiff rod string. Under these rapid leak-off conditions, Method I may not be applicable. Successful application of Method II depends on very delicate handling of the pumping unit during the valve check measurements.

The procedure has been described in terms of the traveling valve check. The technique can also be used to estimate standing valve leakage. The analyst carefully operates the unit to determine the polished rod velocity which yields a downward displacement rate equal to leakage rate past the standing valve. This is noted on the record when load begins to increase from the minimum value. This occurs when the traveling valve closes.

Method II should only be used when the tubing is anchored at the pump.

METHOD III

Pump Leakage From the Downhole Pump Dynamometer Card and Pump Velocity

A variation on the polished rod velocity method uses the pump card and pump velocity to determine the critical point at which upward displacement rate equals leakage rate. This method of determining pump slippage is applied when card shape shows abnormal pump leakage. Also, the first method discussed may not be applicable because initial load loss rate is too rapid to record on surface valve checks.

Pump velocity V is derived by differentiating the formula for pump position developed in pump card theory (Refs. 3, 4),

$$V = \frac{\partial u}{\partial t}(L_r,t) = \sum_{i=1}^{n} -i\ O_i(L_r) \sin i\omega t$$

$$+\ i\ P_i(L_r) \cos i\omega t \qquad (18)$$

When a downhole pump card is available, the pump slippage coefficient (usually taken to be $C_p = 0.5$) can be estimated from

$$C_p = K \sum_{i=1}^{q} \left[0.5 \left(Q_i^P + Q_{i+1}^P \right) - Q_{min}^P \right] (t_{i+1} - t_i)$$

in which (19)

$$K = \frac{1}{\left(Q_{max}^P - Q_{min}^P \right) \theta}$$

Figure 10 shows a calculated downhole pump card and a pump velocity card in a well with a leaking plunger/barrel or traveling valve assembly. The characteristic card shape indicative of this type of leakage is evident, i.e. slow application and premature release of pump load on the upstroke. Equation 16 can be used to estimate leakage from pump velocity. The horizontal (position) scales in Figure 10 are aligned so that pump velocity corresponding to standing valve opening can be determined. The critical pump velocity is 26.6 in/sec. For a 1.25 inch pump, the leakage rate is

$$R = 6.99\ (1.25)^2 (0.47)(26.6) = 137 \text{ bpd}$$

The pump slippage coefficient $C_p = 0.47$ has been derived from the pump card using equation 19.

Gross pump displacement for this well was 204 bpd at stock tank conditions. Accounting for leakage of 137 bpd, production is estimated at 67 bpd (204-137) which agrees favorably with the reported test of 65 bpd.

The pump velocity method can also be used to infer standing valve leakage. In this case, critical velocities are at the moment of traveling valve opening and closing. Inasmuch as the diagnostic equations compute pump velocity with respect to the casing (not tubing), Method III should be used only when the tubing is anchored at the pump.

ASSUMPTIONS AND CONDITIONS OF APPLICABILITY

Pertinent assumptions and conditions need to be noted when employing these techniques. One assumption is that wear is reasonably uniform along the barrel so that critical measurements are representative. Since this assumption is not always borne out, a source of error exists. Another condition is that the tubing should be anchored at or near the pump in Methods II and III. Further work is needed to generalize these methods to handle unanchored tubing. As in all methods discussed, it is convenient to presume that leakage occurs about 50% of the time during the stroke. This assumption is not fulfilled in wells with incomplete liquid fillage (fluid pound or gas interference). The effect of rod/tubing drag on rod stretch is ignored. This is also a source of error.

Leakage rate in shallow wells with significant fluid acceleration effect is greater than calculated with any of these methods. Leakage measurements discussed are made under near static conditions. Fluid acceleration effects add to fluid loads and pressure differences and hence increase leakage rates.

It is also necessary that the standing valve must be holding in order to calculate traveling valve leakage, and vice versa. If both are leaking, measurements would assign the defect to the valve leaking at the greater rate. Actual leakage would be more than calculated.

Experience has shown that most of the leakage indicated by the traveling valve check is caused by slippage between the plunger and barrel, and that traveling valve checks can not differentiate between plunger/barrel leakage and traveling valve leakage. Although not studied explicitly in this paper, traveling valve checks can not identify tubing leaks. If the tubing were leaking enough to be seen with a traveling valve check (which is made within a matter of seconds), the leak would greatly exceed pump displacement and no fluids could be produced to the surface.

CONCLUSIONS

To expand the use of dynamometer acquired data, three methods have been developed to quantify pump leakage. The measurements and calculations are quick and simple. Implementation requires time based polished rod load and position data which are readily available with digital-electronic dynamometer equipment. By quantifying pump leakage, the operator can be more objective as to when to change the pump, thus avoiding needless repairs or losses in production.

NOMENCLATURE

A_p = pump area, in^2

A_i = rod areas in tapered string, in^2 ($i = 1,2, \ldots m$)

a_i = polynomial coefficients for load ($i = 1,2,3$)

b_1 = polynomial coefficient for position

C_p = pump slippage coefficient

D = instantaneous pump displacement, bbl/day

d = pump diameter, in

δ_r = static rod stretch caused by fluid load, in

δ_t = static tubing stretch caused by fluid load, in

E_i = rod moduli in tapered string, psi (i = 1, 2, ...m)

E_r = elastic constant for rods, in/lbf-ft

E_t = elastic constant for unanchored tubing, in/lbf-ft

i = summation index

L_r = combined length of rods, ft

L_t = combined length of unanchored tubing, ft

m = number of intervals in tapered string

n = number of terms in series for pump position

$O_i(L_r)$ = diagnostic functions evaluated at pump, in

$P_i(L_r)$ = diagnostic functions evaluated at pump, in

P_a = pressure above traveling valve, psi

P_b = pressure below traveling valve, psi

P_i = pump intake pressure, psi

Q(t) = dynamic polished rod load during traveling valve test, lb_f

Q_i = polished rod load at specific times, lb_f (i = 1,2,3)

Q_i^P = pump loads used to construct pump card, lb_f (i = 1,2,...q)

Q_f = fluid load on pump, lb_f

Q_d = downstroke rod drag, lb_f

Q_r = buoyant rod weight, lb_f

Q_s = standing valve load, lb_f

Q_t = traveling valve load, lb_f

Q_u = upstroke rod drag, lb_f

Q_{max}^P = maximum pump load, lb_f

Q_{min}^P = minimum pump load, lb_f

$\left(\dot{Q}_t\right)_{max}$ = maximum rate of traveling valve load loss, lb_f/sec

q = number of points used to construct pump card

R = pump leakage rate, bbl/day

$\bar{R}$ = average leakage rate, bbl/day

t = time,sec

Δt = time required to lose fluid load, sec

t_i = specific times, sec

$u(L_r,t)$ = pump position at arbitrary time, in

V = pump velocity, in/sec

V_c = critical pump velocity at valve actuation, in/sec

ω = angular frequency, rad/sec

X_i = interval length in tapered string, ft ($i = 1, 2,...m$)

Y_i = specific polished rod positions, in ($i = 1,2,3$)

Θ = period of pumping cycle, sec

ACKNOWLEDGMENTS

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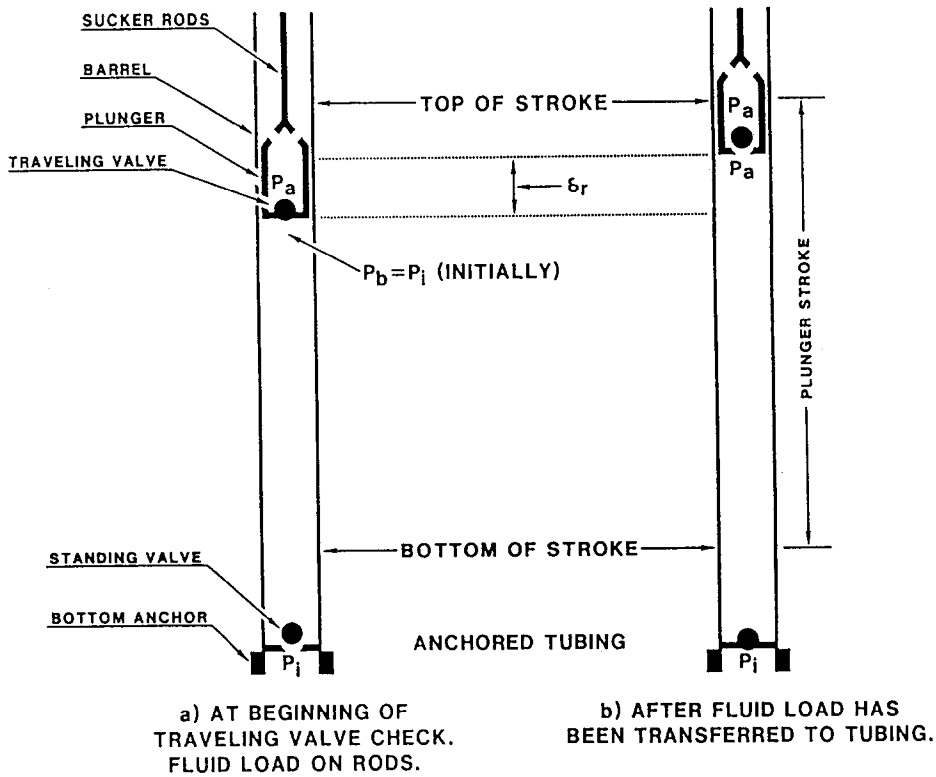

Figure 1 — Pump nomenclature and pressures during traveling valve check

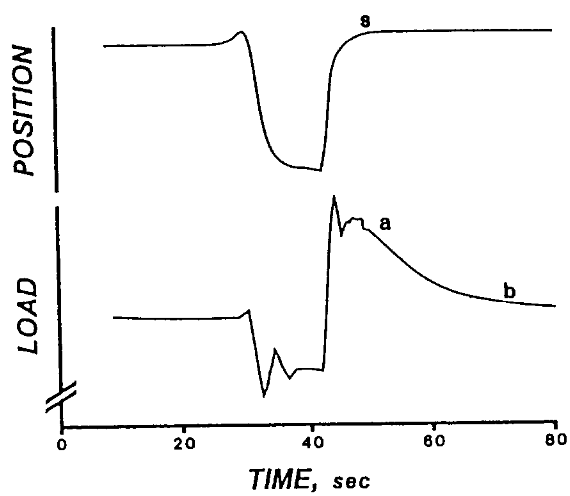

Figure 2 — Traveling valve check showing rapid load loss

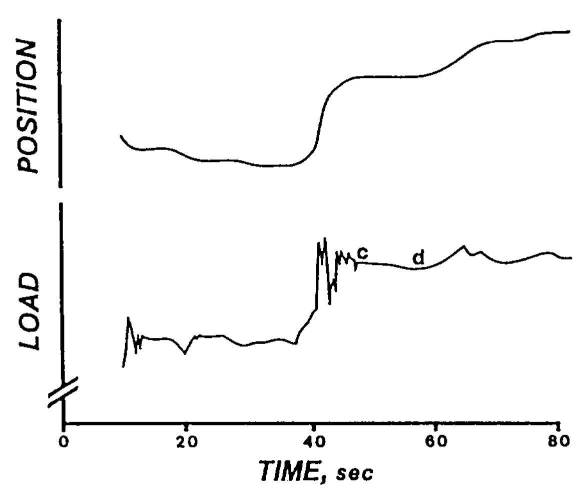

Figure 3 — Traveling valve check showing slow loss of load

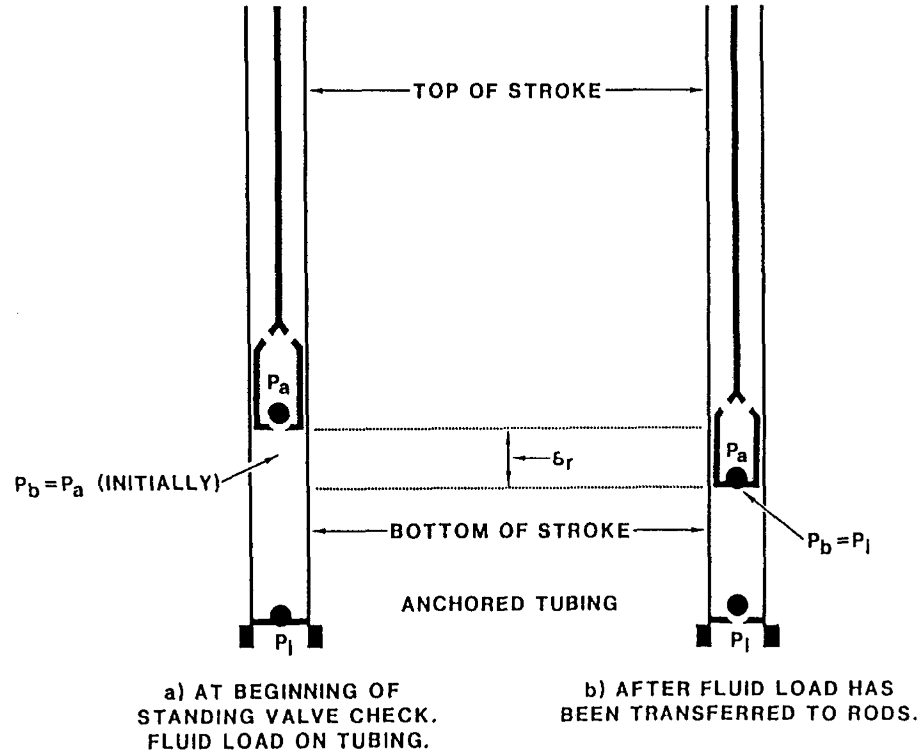

Figure 4 — Pressures during standing valve check

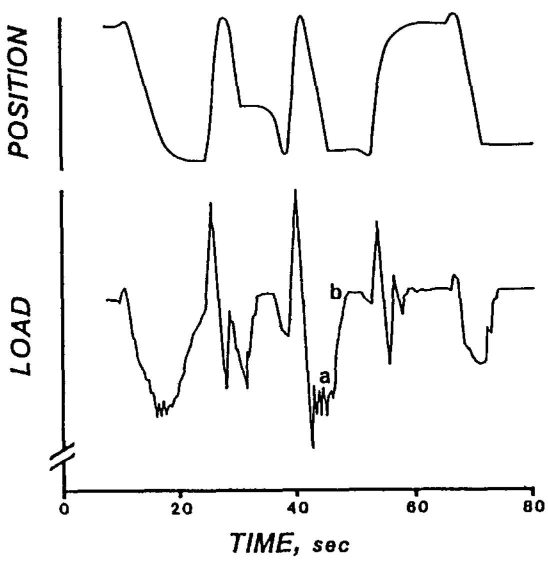

Figure 5 — Standing valve check showing rapid load increase

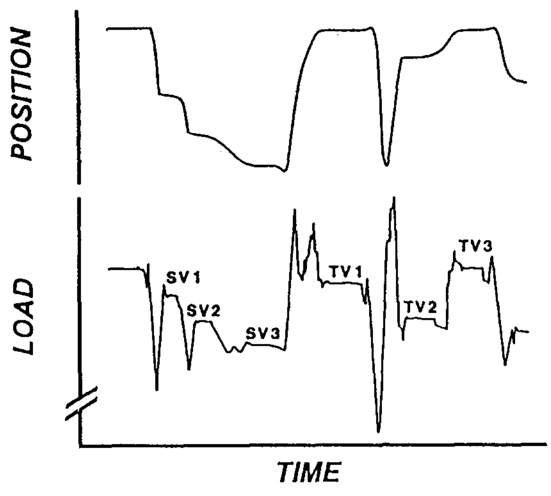

Figure 6 — Adverse effect of abrupt valve check stops

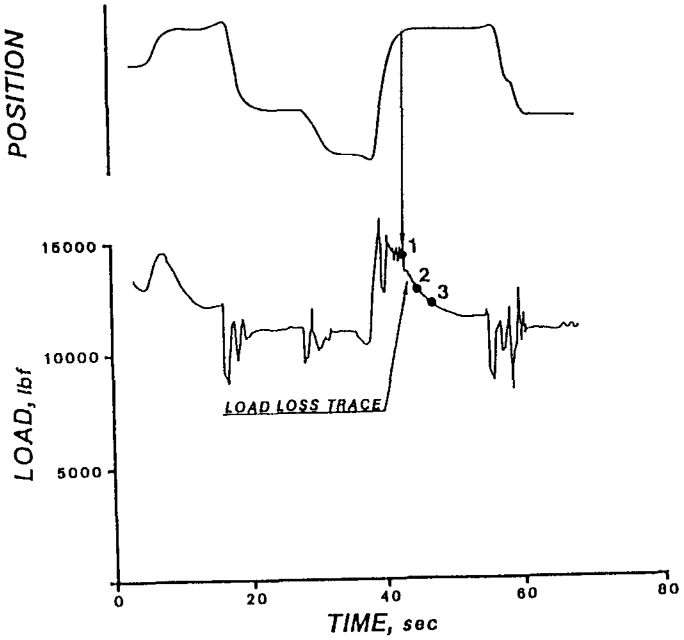

Figure 7 — Traveling valve check showing points for determining initial load loss rate

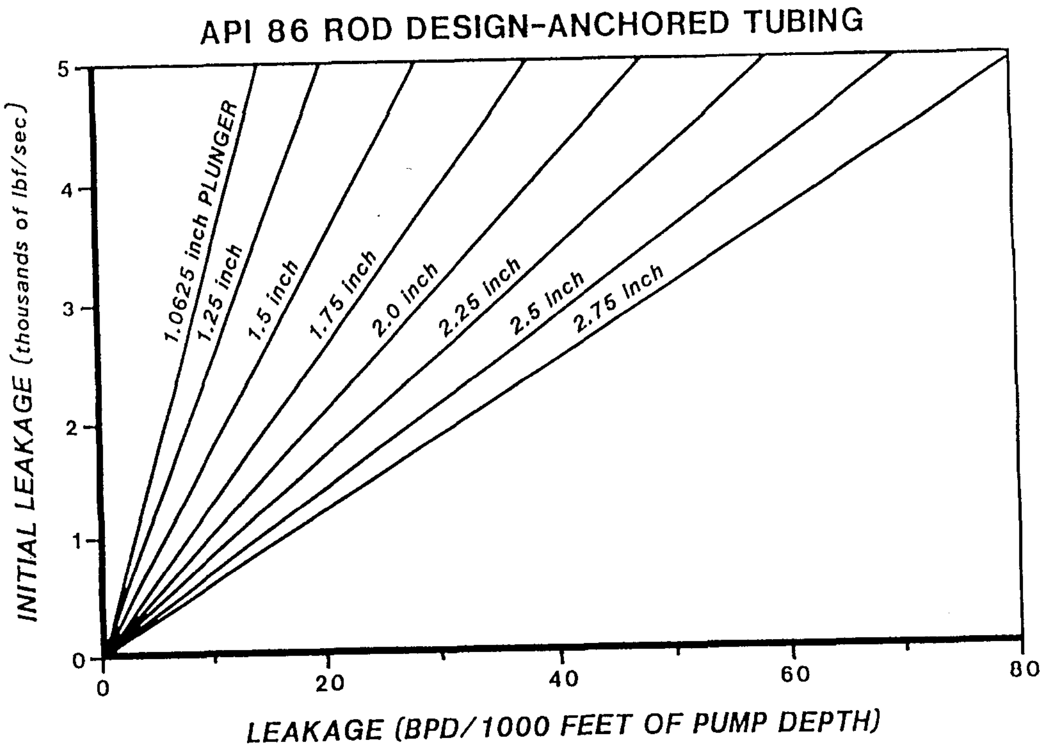

Figure 8 — Pump leakage nomograph

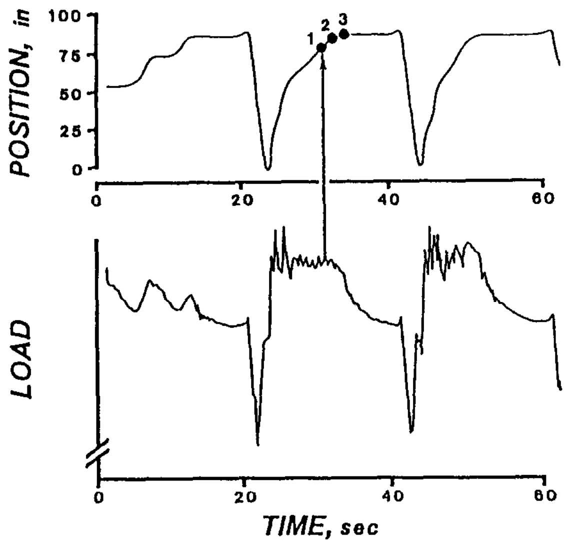

Figure 9 — Traveling valve check showing points for determining critical upward velocity

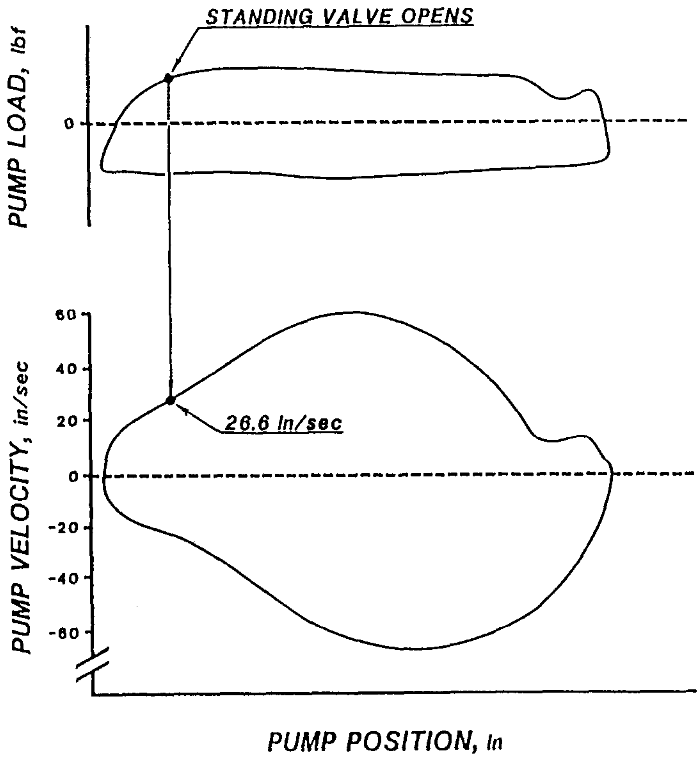

Figure 10 — Critical upward velocity determined from pump card