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VALVE EFFICIENCY AT DEPTH PUMPS

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ABSTRACT

Oil extraction by deep pumping uses bottom hole pumps whose efficiency depends on several parameters like: surface stroke length, stroke rate, pump depth, pump clearance. Pump valves play an important role in the efficiency of deep pumping. They wear out quite frequently, especially when the pumped oil contains mechanical suspensions. Choosing the right valves increases the pumping efficiency and prolongs the service life of the pump.

Keywords: Pump, Oil extraction, Pump valves, Pumping efficiency

1. THE SYSTEM OF FORCES ACTING ON THE VALVE PISTON AND PUMP

The efficiency of the depth pump depends on how the valve sits and rises from the seat. The valve stroke depends on the forces acting on the upward and downward stroke of the pumping unit. Specialized literature presents this phenomenon from qualitatively point of view, considering that successive collisions occur when the valve is positioned on the seat. This paper is based on the fact that the system of forces acting on the valve leads to its closing and opening, without denying the collision phenomenon.

It is considered that the forces acting on the valve are the following: its own weight (G), the running resistance force, (C) , Archimedes' force, (A), and the inertial force due to the valve acceleration (F). If the pump shaft is not vertical, a frictional force occurs between the valve and the pump body. In the following, this force is being neglected.

The valve weight is calculated with the following formula:

$$G = \frac{4\pi R^3}{3} g \rho_m \tag{1}$$

where: R is the ball radius

ρ_m - the valve material density

The force due to the movement of the fluid over the valve ball, as called the running resistance, appears as a result of the relative speed between the oil and the ball.

This force therefore appears only when there is a movement of the oil over the ball. For this reason, it is expected that both where the valve falls on the seat and during its opening, such resistance forces will appear because in these situations there is a relative displacement between the fluid and the valve.

The running resistance force is calculated with the following formula:

$$C = \rho \frac{v^2}{2} \pi R^2 C_R \tag{2}$$

where: ρ is the oil density

C_R – running resistance coefficient

C_R is determined using one of the following formulas

$$\text{if Re} \leq 1, C_R = \frac{24}{Re} \tag{3'}$$

$$\text{if } 1 < \text{Re} \leq 10^3,\; C_R = \frac{18{,}5}{Re^{0{,}6}} \tag{3''}$$

$$\text{if Re} > 10^3,\; C_R = 0{,}44 \tag{3'''}$$

Re is the Reynolds'number defined using the ball diameter and the relative speed of the oil against the valve.

$$Re = \frac{(2R)v}{\nu} \tag{4}$$

υ is the oil cinematic viscosity

It is obvious that, during each stroke of the piston, the resistance force varies due to the variation of the relative speed between the oil and the ball as well as the different values of the resistance force.

Archimedes' force must be calculated in two different situations, namely:

- the valve is moving up or down from the seat (there is no contact between the two pieces)
- the valve is seated

At the ascending stroke end, the valve speed becomes zero and it begins to fall. The period between the two strokes is extremely short because on the valve acts its own weight and the running resistance force of the fluid at that moment, both forces being oriented downwards.

When the valve is moving up or down from the seat, Archimedes' force is as follows

$$A = \frac{4\pi R^3}{3} g\rho \tag{5}$$

When the valve is seated, there is a contact between these pieces on a certain surface (figure 1). If H is the fluid column measured from the seat to the surface, the relative pressure at the seat is

$$p = \rho g H \tag{6}$$

If p_s is the pressure below the valve, it may be the equivalent of a fluid column h_s:

$$h_s = \frac{p_s}{\rho g} \tag{7}$$

It results that on the arc AB, the pressure varies continuously from p to p_s.

Considering the fluid flow on the AB arc as a laminar movement between two circular discs, the variation of the pressure expressed in the oil column can be written, depending on the radii r_1, r_2 and r:

$$\bar{H}(r) = \bar{H} - \frac{H-h_s}{ln\frac{r_2}{r_1}} ln\frac{r_2}{r} \tag{8}$$

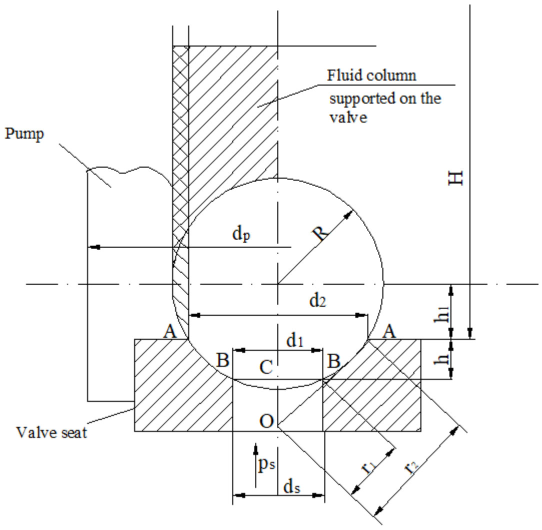

Figure 1. *Depth pump valve geometry*

Some approximations are needed in the above equation.

It is known that for a small argument x ,

$$ln(1+x) \cong x \tag{9}$$

Taken into account the relationship (9), the approximations below can be written

$$ln\frac{r_2}{r} = -ln\frac{r}{r_2} \cong -\left(\frac{r}{r_2} - 1\right) = \frac{r_2 - r}{r_2} \qquad (10')$$

$$ln\frac{r_2}{r_1} = -ln\frac{r_1}{r_2} \cong -\left(\frac{r_1}{r_2} - 1\right) = \frac{r_2 - r_1}{r_2} \qquad (10'')$$

So the relation (8) becomes

$$\bar{H}(r) = H - (H - h_s)\frac{r_2 - r}{r_2 - r_1} \tag{11}$$

As (8), (11) satisfies the boundary conditions

at $r = r_1, \bar{H}(r) = h_s$ (12)

at $r = r_2, \bar{H}(r) = H$

The pressure on the arc AB is calculated with the relation

$$p = \rho g \bar{H}(r) \tag{13}$$

where $\bar{H}(r)$ is according to (11)

Because the pressure is variable, the elemental force on a surface element is therefore calculated. Due to the valve symmetry with respect to its vertical axis, this force has only the vertical component (figure2) and it is calculated with the following relation:

$$dF_v = 2\pi rp\ dr\ sin\theta = 2\pi \rho g r \overline{H}(r) sin\theta\ dr \tag{14}$$

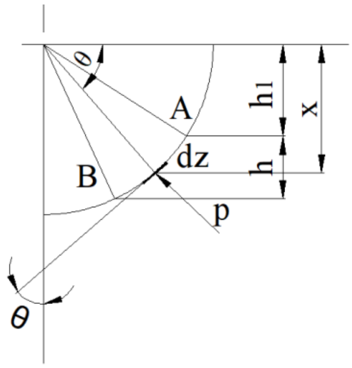

Figure 2. *Depth pump valve geometry - detail*

Because

$$sin\theta = \frac{r_2 - r_1}{h} = \frac{d_2 - d_1}{2h}$$

the vertical force F_v is

$$F_v = \int_{r_1}^{r_2} dF_v = \frac{\pi \rho g (d_2 - d_1)}{h} \int_{r_1}^{r_2} r \left[H - (H - h_s)\frac{r_2 - r}{r_2 - r_1}\right] dr \tag{15}$$

where d_1 and d_2 – diameters of circles in figure 1

Eq. (15) becomes

$$F_v = \frac{\pi \rho g}{6h}(d_2 - d_1)[H(2r_2^2 - r_1 r_2 - r_1^2) + h_s(r_2^2 + r_1 r_2 - 2r_1^2)] \tag{16}$$

The pressure force on the spherical cap (dome) above the valve seat is added to the above force that appears on the arc AB. The pressure force is calculated as follows

$$F_s = \left[\frac{\pi}{3}(2R^3 + 3R^2 h_1 - h_1^3) - \frac{\pi d_2^2}{4} H\right] \rho g \tag{17}$$

So, when the valve is placed on its seat, the total force F_t is acting on it, as the sum between F_v and F_s.

$$\frac{F_t}{\rho g} = \frac{d_2 - d_1}{6h}[H(2r_2^2 - r_1 r_2 - r_1^2) + h_s(r_2^2 + r_1 r_2 - 2r_1^2)] + \frac{2R^3 + 3R^2 h_1 - h_1^3}{3} - \frac{d_2^2}{4} H \tag{18}$$

The positive components of forces are directed upwards vertically and the negative ones are directed downwards.

2. PISTON EFFICIENCY AT THE ASCENDING STROKE OF THE PUMPING UNIT

At the upward stroke of the piston, its valve falls on the seat which causes the oil to be driven to the surface. If the valve is closed exactly at the beginning of the ascending stroke of the piston, the closing efficiency of the pump is very good, i.e. equal to the unit.

$$\eta_i = 1 \tag{19}$$

It is possible to exist a period of time t_a from the ascending stroke start until the complete closure of the hole between the valve and the piston. At that time, a part of the oil above the pump will be drained. The amount of the drained oil depends on the height of the column to the surface, the flow coefficient of the drain section, the size of the cross section of the drain and the physical properties of the oil.

Some researches have shown the dependence of the flow coefficient on the shape and Reynolds number of the drain section.

At the ascending stroke of the pumping unit, the piston valve is located at the top of the cage whose length is l_c.

Considering the valve as a material point, the motion equation is written as follows

$$\ddot{x}\frac{4\pi R^3}{3}\rho_m g = G + C - A \tag{20}$$

or

$$\ddot{x} = g\left(1-\frac{\rho}{\rho_m}\right)+\frac{3}{4}C_R\varphi_v^2\frac{H}{R}\frac{\rho}{\rho_m}g = gB \tag{21}$$

where: H - pump set depth

φ_v – speed factor of the valve hole

$$B = 1-\frac{\rho}{\rho_m}+\frac{3}{4}C_R\varphi_v^2\frac{H}{R}\frac{\rho}{\rho_m}$$

$\varphi_v < 1$ and it depends on the local resistances as well as on the valve opening system geometry.

The boundary conditions for Eq. (21) are

at $t = 0$, $x = l_c$

at $t = t_a$, $x = 0$ (22)

Considering B as a constant, equation (21) is easily integrated and the result is

$$x = l_c - \frac{1}{2}gBt^2$$

Time from the upward stroke begins until the valve reaches its seat is calculated providing that the valve distance x is equal to the bottom stroke of the pumping unit, S, which has the expression

$$S = Kkr\left(1-cos\varphi+\frac{r}{2l}sin^2\varphi\right) \tag{24}$$

Where: K – overstroke factor

ratio r/l depends on the chosen pumping unit

φ is the angle made by the crank

k- is the ratio between a and b (fig 3)

The accepted calculation hypothesis leads to equality

$$l_c - \frac{1}{2}gBt_a^2 = Kkr\left(1-cos\varphi_a+\frac{r}{2l}sin^2\varphi_a\right) \tag{25}$$

φ_a is the angle made by the crank in t_a

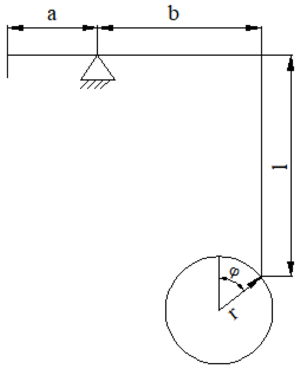

Figure 3. *Illustrative geometry of pump unit*

If it is considered that

$$cos\varphi_a \approx 1$$
$$sin\varphi_a \approx \varphi_a \qquad (26)$$
$$\frac{r}{l} = ct \approx 0{,}4$$

Eq. (25) becomes

$$l_c - 0{,}5gBt_a^2 = 0{,}2\varphi_a^2 Kkr \qquad (27)$$

The relationship between φ_a and the angular velocity ω of the pumping unit is as follows

$$\varphi_a = \omega t_a \qquad (28)$$

Due to the above considerations

$$l_c = \left(0{,}2Kkr + \frac{0{,}5gB}{\omega^2}\right)\varphi_a^2 \qquad (29)$$

so

$$\varphi_a = \omega\sqrt{\frac{l_c}{0{,}2Kkr\omega^2 + 0{,}5gB}} \qquad (30)$$

It is obvious that, as the angle φ_a is higher the valve closes more slowly, which leads to a decrease of closing efficiency of the pump, η_i.

Closing efficiency of the pump, η_i, is defined according to the following formula

$$\eta_i = \frac{S|_{\varphi_a}^{\pi}}{S|\mathrm{I}_0^{\pi}} \qquad (31)$$

where: $S|_0^{\pi}$ is the pumping unit bottom stroke;

$S|_{\varphi_a}^{\pi}$ is the effective stroke where the oil is pumped to the surface.

The stroke ‘consumed’ at the pump level $S\mathrm{I}_0^{\pi}$ is easily determined making the difference between the expression of S given by the relation (24) for φ=π and S also calculated with Eq. (24) for the angle φ = 0. We obtain

$$S|_0^{\pi} = 2kr \qquad (32)$$

The 'useful' stroke is determined as the difference between the value of S for φ=π and S for φ= φ_a.

So,

$$S|_{\varphi_a}^{\pi} = Kkr\left(1 + \cos\varphi_a - \frac{r}{2l} sin^2\varphi_a\right) \tag{33}$$

And the efficiency of the pump, η_i, is

$$\eta_i = \frac{1 + \cos\varphi_a - \frac{r}{2l} sin^2\varphi_a}{2} \tag{34}$$

Also considering the approximations (26), η_i is as follows

$$\eta_i = 1 - 0{,}1\varphi_a^2 \tag{35}$$

where angle φ_a is according to Eq. (30).

The above calculation of η_i was made not considering the fluid leakage through the annular space between the piston and the cylinder.

3. EFFICIENCY AT THE VALVE OPENING

The valve should be opened when the descendent stroke of the pumping unit begins. Theoretically, this would be achieved when the angle of the crank is equal to π, the valve being opened until φ is equal to 2π.

Actually, there is a delay in the valve opening due to the compressibility of the fluid under the pump. It can be admitted that the valve opens when

$$F_c + \frac{\pi d_s^2}{4} p_d \geq F_t \tag{36}$$

The force due to the fluid compressibility, Fc, is obtained from the definition of the compressibility coefficient of the fluid under the pump.

Thus, it is known that

$$\beta = \frac{1}{V}\frac{\Delta V}{\Delta p} \tag{37}$$

Where: V is the volume of the compressed fluid

Δp - the increase in pressure leading V to decrease with ΔV

If it is considered that V is calculated with the formula

$$V = \frac{\pi d_p^2}{4}(KS - \lambda) \tag{38}$$

then

$$\beta = \frac{\Delta h}{KS - \lambda}\frac{1}{\Delta p} \tag{39}$$

Regarding the above relations, d_p is the piston diameter, λ is the elongation of the system of pumping rods - extraction pipes column, Δh is the piston movement until the valve opens.

So, the force due to the fluid compressibility, Fc, is as follows

$$F_c = \frac{\pi d_s^2}{4}\Delta p = \frac{\pi d_s^2}{4}\frac{\Delta h}{\beta(KS-\lambda)} \quad (40)$$

It can be admitted that Δh is the difference between the piston stroke when $\varphi=\pi$ (the moment when the valve should open), and the same stroke for $\varphi=\varphi_d$ (φ_d is the angle of crank when the valve is raised),

$$\Delta h = Kkr\left(1+\cos\varphi_d - \frac{r}{2l}sin^2\varphi_d\right) \quad (41)$$

Taking into account formula (39), we obtain

$$Kkr\left(1+\cos\varphi_d - \frac{r}{2l}sin^2\varphi_d\right) \geq \left(\frac{4F_d}{\pi d_s^2} - p_s\right)\beta(2Kkr-\lambda) \quad (42)$$

Admitting $\lambda \ll 2Kkr$ and noting

$$p_c = 2\left(\frac{4F_t}{\pi d_s^2} - p_s\right)$$

Eq. (36) can be written

$$1+\cos\varphi_d - \frac{r}{2l}sin^2\varphi_d \geq p_c\beta \quad (43)$$

For the case when $\frac{r}{l} \approx 0{,}4$ – a common value for pumping units, Eq. (43) becomes

$$2cos^2\frac{\varphi_d}{2} - 0{,}8\left(1-cos^2\frac{\varphi_d}{2}\right)cos^2\frac{\varphi_d}{2} \geq \beta p_c \quad (44)$$

Noting $y = cos^2\frac{\varphi_d}{2}$, Eq. (44) becomes

$$0{,}8y^2 + 1{,}2y - \beta p_c = 0 \quad (45)$$

This equation has the following solutions

$$y_1 = \frac{-0{,}6+\sqrt{0{,}36+0{,}8\beta p_c}}{0{,}8};\ y_2 = \frac{-0.6-\sqrt{0.36+0.8\beta p_c}}{0.8} \quad (46)$$

y_1 is the only solution having sense from physically point of view.

For a small argument x, we can apply the following relation in order to simplify the radical in y_1 expression

$$\sqrt{1+x} \cong 1+\frac{x}{2} \quad (47)$$

So,

$$y_1 = 0{,}5\beta p_c \quad (48)$$

which means

$$cos\frac{\varphi_d}{2} = \pm\sqrt{0{,}5\beta p_c} \quad (49)$$

$$\varphi_d = -2arccos\sqrt{0{,}5\beta p_c} \quad (50)$$

β=0 for an incompressible oil, so $\varphi_d = \pi$, which means the valve raising is realised as soon as the pumping unit descendent stroke begins.

When gas is under the pump, it can be considered as having either an isothermal evolution, so

$\beta p_c = 1$ (51)

or an adiabatic evolution, so

$\chi\beta p_c = 1$ (52)

Where

χ is the gas adiabatic exponent and has the following values: 1,32 for methane; 1,18 for ethane; 1,12 for propane

For an isothermal evolution of the gas,

$\varphi_d = -2\ arccos\sqrt{0,5} = \frac{3}{2}\pi$ (53)

So the valve opening efficiency is

$\eta_d = \frac{SI_{2\pi}^{\varphi_d}}{SI_{2\pi}^{\pi}}$ (54)

$S|_{2\pi}^{\pi}$ is the difference between the piston stroke when φ=π and φ=2π, respectively.

$S|_{2\pi}^{\varphi_d}$ is the difference between the piston stroke when φ=φ$_d$ and φ=2π, respectively.

In case of $\varphi_d = \frac{3}{2}\pi$,

$S|_{2\pi}^{\varphi_d} = kr\left[\left(1 - \cos\frac{3}{2}\pi + \frac{r}{2l}sin^2\frac{3}{2}\pi\right) - \left(1 - \cos 2\pi + \frac{r}{2l}sin^2 2\pi\right)\right] = kr\left(1 + \frac{r}{2l}\right)$ (55)

Similarly, we obtain

$S|_{2\pi}^{\pi} = 2kr$ (56)

Considering r/l = 0,4, the valve opening efficiency for an isothermal behavior of the gas is

$\eta_d = \frac{kr\left(1+\frac{r}{2l}\right)}{2kr} = 0,6$ (57)

If gas has an adiabatic evolution and admitting that there is methane under the valve,

$\beta p_c = \frac{1}{x} = \frac{1}{1,32} = 0,7575$ (58)

So,

$\varphi_d = -2arccos\sqrt{0,75 \cdot 0,7575} = 255,96°$ (59)

Using Eq. (59),

$S|_{2\pi}^{\varphi_d} = kr\left[\left(1 - cos\varphi_d + \frac{r}{2l}sin^2\varphi_d\right) - \left(1 - \cos 2\pi + \frac{r}{2l}sin^2 2\pi\right)\right] = kr\left(1,2426 + 0,4706\frac{r}{l}\right)$ (60)

Also considering r/l = 0,4, the valve opening efficiency for an adiabatic behavior of the gas is

$\eta_d = \frac{1,4308kr}{2kr} = 0,7154$ (61)

This value is close enough to that obtained for an isothermal evolution of the gas.

4. CONCLUSIONS

The efficiency of the valves, at closing and opening, depends on several parameters such as: the length of the stroke performed by the valve, the Pump Unit characteristics, the buoyancy of the valve, the fixing depth of the pump, the size of the valve and the forward resistance coefficient; the efficiency of opening the valve depends largely on the compressibility of the oil under the pump. The existence of solid particles in crude oil can substantially change the valve efficiency at depth pumps.

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