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A WEAR STUDY CASE OF CERAMIC BALL SEAT VALVE

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Summary

This paper describes an erosion- corrosion wear model of carbide materials in crude petroleum fluid flow. The wear model is based by elastically or plastically fatigues, when the corrosion fluid has solid particles. The velocities of solid particles are evaluated by the Reynolds equation. Mathematical model for solid particle has been combined with those for crude petroleum corrosion. This has been addressed through the ball valve of crude petroleum extraction pump. The paper describes some new theoretical developments on the above work.

The synergic effect of erosion and corrosion rate was observed for carbide and ceramic composites. Theoretical erosive-corrosive wear model has been used to define the possibility to increase the durability of ball and seat valve. This model is able to solve the contact problem under flow of contaminated corrosive fluid with abrasive particle. It is based on Newton fluid low in the angular gap between the ball and the seat valve.

Keywords: *Erosion - corrosion; Analytical wear model; Case study; Ball valve.*

1. INTRODUCTION

Solid particle erosion in fluid flow is an important material degradation mechanism wear encountered in a number of engineering systems such as aircraft gas turbine engines, liquid transports, coal slurry pipe lines or crude petroleum extraction pumps. Over the last few decades, almost all aspects of solid particle erosion have been investigated in great detail [1]. The action of erosive crude petroleum component may cause removal of a passive corrosion film, thereby removing the ability of the material to withstand corrosion. On the other hand, the effect of the chemical crude petroleum may reduce the ability of the material to resist the mechanical attack [2-4]. Typical ball and valve seat materials of crude petroleum extraction pumps, whether they are stainless martensitic steels, do not have adequate corrosion – erosion resistance. An other our paper describes the service life of ball–valves, which are manufactured from composite materials (sintered metallic carbide for valve seats and balls and ceramic sintered composite for balls) [5]. The primary objective of this paper is to assess the petroleum fluid flow velocities between ball and valve seat, to rationalize the observed erosion behavior on the basis of an fatigue erosion wear model. The secondly objective of this paper is to define the impact surface area damaged by corrosion and erosion and to proposed a wear model for the ball – valve seat. The experimental "in situ"results will be analyzed with the theoretical wear model.

2. EROSIVE FLUID FLOW MODEL

The crude petroleum liquid is defined as a Newtonian fluid and the entire abrasive solid particle are fine dispersed in mass of fluid. The geometry of the ball–walve seat and the theoretical model of fluid flow is shown in Fig.1. The ball of valve is moved in conical or spherical seat by the hydrostatic pressure of fluid. An exterior pump has created the pressure of fluid. In theoretical model is appreciated that ball is moved only to axial direction and the charge is axial.

In these conditions, the Reynolds pressure equation, in spherical coordinates, is

$$\frac{\partial}{\partial\theta}\left(h^{3}\sin\theta\frac{\partial p}{\partial\theta}\right)=0 \tag{1}$$

where h is the thickness of the erosive – corrosive fluid.

From Fig. 1, this thickness can be write as a

$$h_{a}=\frac{h}{R}=\frac{1+h_{pa}}{sin(\alpha+\theta)}-1 \tag{2}$$

The boundary conditions for to integrate Reynolds eq. (1) are

$$\begin{aligned} p=p_{i} \quad & if \quad \theta=\theta_{i}\\ p=p_{e} \quad & if \quad \theta=\theta e \end{aligned} \tag{3}$$

where p_i and p_e are the pressure in a front of the valve and respectively at exit of valve.

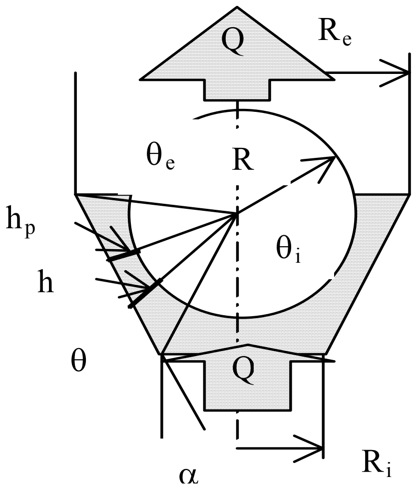

Fig.1. Geometry gape for the flow model of crude petroleum liquid.

The pressure in contact between ball and valve seat will be write

$$p=\frac{c_{1}}{R^{3}}\int\frac{d\theta}{\left(\frac{1+h_{pa}}{sin(\alpha+\theta)}-1\right)^{3}\sin\theta} \tag{3}$$

where c_1 is a constant which be defined by the flow rate.

The mechanical equilibrium condition of the valve in the fluid film is

$$F=\pi R_{i}^{2}\left(p_{i}\sin^{2}\theta_{i}-p_{e}\sin^{2}\theta_{e}+2\int_{\theta_{i}}^{\theta_{e}}p\sin\theta d\theta\right) \tag{4}$$

and the fluid pressure in a front to valve

$$p_{m}=F/\left(\pi R_{i}^{2}\right),$$

where F is the charge of ball, which can be evaluate as a weight of ball.

The Poiseuille flow velocity in the gap

$$u_{\theta}=\frac{-1}{2\eta R}\frac{\partial p}{\partial\theta}g(h-g) \tag{5}$$

and the mean velocity

$$u_{\theta m}=\frac{1}{h}\int_{0}^{h}u_{\theta}=\frac{-h^{2}}{12\eta}\frac{\partial p}{\partial\theta} \tag{6}$$

The flow rate of crude petroleum in the gap

$$Q_{\theta}=u_{\theta m}\pi R^{2}\left(\frac{h}{R}\right)\left(2+\frac{h}{R}\right)=\frac{-c_{1}}{12\eta}\frac{1}{\sin\theta}\left(2+\frac{h}{R}\right) \tag{7}$$

But, the flow rate is constant (Q), thus the constant c_1 can be evaluated

$$c_{1}=\frac{-12\eta\cos\alpha}{2+h_{p}/R} \tag{8}$$

The solid abrasive particles in crude petroleum are characterized by the volume concentration (c).

The following dimensionless parameters will be used

$$\begin{aligned} &p_{ai}=p_{i}/p_{m},\quad p_{ae}=p_{e}/p_{m},\quad p_{a}=p\ /p_{m},\\ &h_{pa}=h_{p}/R,\quad u_{a\theta}=u_{\theta m}/u_{i}. \end{aligned} \tag{9}$$

The equilibrium position of the ball in conical gape (the dimensionless parameter h_{pa}) will be obtained numerically, by using the eq. (4).

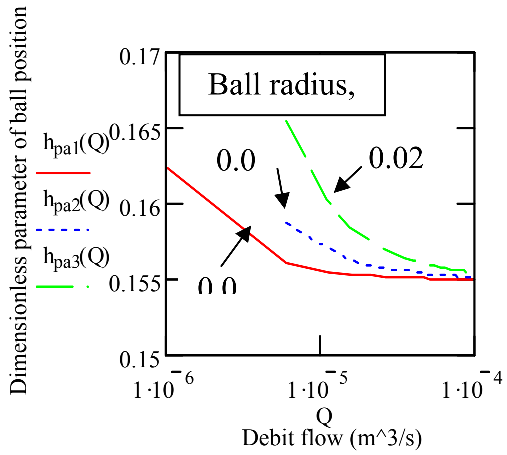

Fig. 2. Position ball in conical gape

Figure 2 shows the equilibrium position of ball when the crude petroleum flows in the valve.

For this position, the pressure of fluid between ball and valve seat can be solving by eq. 3. The effect of ball radius and fluid debit about the fluid pressure is shown in Fig. 3 and Fig. 4.

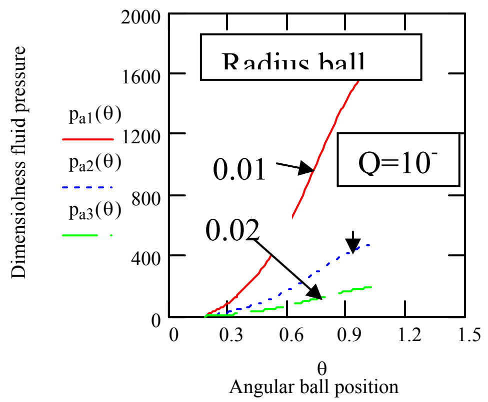

Fig.3 Fluid pressure in gap

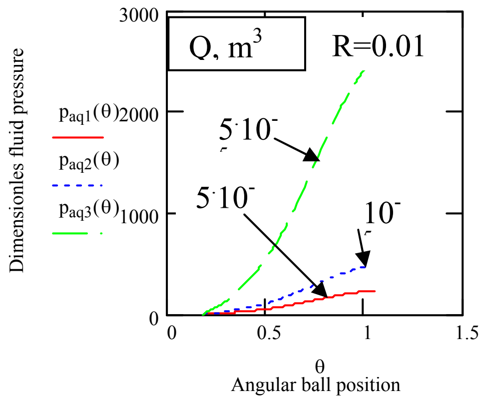

Fig.4. Fluid pressure in gap for diverse flow rates

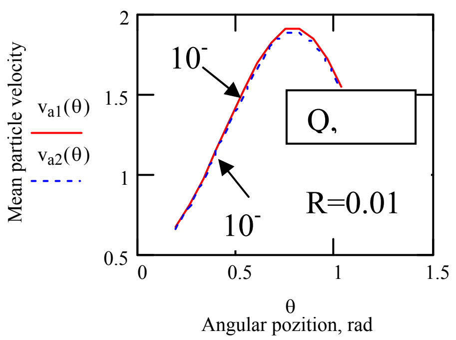

Fig.5. Mean particle velocity in gap

The influence of fluid debit about pressure is similarly as well as radius ball. Note that all abrasive particles have the same dependence on fluid parameters. Thus, the velocities (u_θ) are defined by the Poiseuille flow and the tangential fluid strengths (τ) are evaluated by Newton's fluid friction low. The local friction coefficient (μ_f) between a solid particle and the ball can be calculate

$$\mu_f = \frac{\tau|_{g=0}}{p} = -\frac{h}{2Rp}\frac{dp}{d\theta} \tag{10}$$

In Fig. 5, the theoretically mean velocity between particle and ball is given as a function of angular ball position and flow rate.

The effect of flow rate about dimensional mean velocity is very small, only 1.5% at 10 times increasing of flow rate.

3. EROSION WEAR MODEL

The influence of eroding particle related parameters (e.g. particle hardness, size, shape, density and friability) and particle flow related parameters (e.g. particle velocity, impact angle, particle flux rate and particle phase density) on erosion behavior has been investigated [6-9].

The theory of quasi-static indentation can be used for solid particle impact, which are in crude petroleum, because the impact velocities are much smaller than the velocity of elastic and plastic deformation metallic materials.

On impact the deceleration of solid particle generates the indentation force on the substrate. The impact angle of solid particles in petroleum fluid for ball and seat is variable. The dimensionless erosion rate (I_{er}) is defined as mass of material removed (ball or seat) per mass of eroding (solid particles in crude petroleum).

We accept the equation of motion of single abrasive particle interacting with the surface (Finnie's models) [4]. The erosion of ductile or brittle metals comprise two wear mechanisms occurring simultaneously: one caused by cutting action of free moving particles in fluid with impact angle grater than the critical impact angle; other caused by repeated elastic or plastic deformation during collision with friction (Manson-Miner's rule) [5],[9].

The critical impact angle is defined as the angle of particle, at which a microchip appears for only one impact. The motion equation of the particle and the mechanical properties of target materials can be calculated this angle [9]:

$$\alpha_{ecr} = a\sin\left[4.3\left(\frac{\sigma_c\theta_{el}}{k_e\mu_f}\right)^{5/2} .(v_o\rho_r\theta_{el})^{-1/2}\right] \tag{11}$$

When the impact angle of particle is smaller than the critical angle, the dimensionless erosion wear rate can be evaluated for three-limit positions of the collision particles:

$\mu_f \tan(\alpha_o) \geq 1$; 2) $0.5 \leq \mu_f \tan(\alpha_o) \leq 1$; 3) $\mu_f \tan(\alpha_o) \leq 0.5$.

We defined this parameter as a friction impact factor for ball in erosive – corrosive fluid (C_{ef}). Figure 6 illustrates the change from friction factor impact in angular gap between ball and conical valve seat.

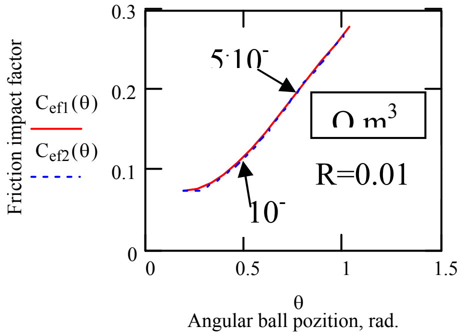

Fig. 6. Friction impact factor for ball

It is observed that for all angles and flow rate, the friction factor corresponds to three-limit condition and the effect of fluid flow rate is very small (maximum 3 %).

The dimensionless erosion rate expression is as follows:

$$I_{er3} = k_{er} I_3 \tag{12}$$

where k_{er} is the dimensionless wear parameter:

$$k_{er} = \frac{3}{4}\left(\frac{5\pi}{4}\right)^{(t+5)/5} \frac{\rho_s}{\rho_r}\left[\frac{4k_e\mu_f}{3\pi\sigma_c\theta_{el}}\right]^t \left[\rho_r\theta_{el}v_o^2 \sin^2\alpha_o\right]^{\frac{t+5}{5}} \tag{13}$$

The integral function I_3 can be evaluated by numerical methods. In Fig. 7 and 8, the variation of the calculated erosion rate with angular position of ball is illustrated for various flow rate and ball radius.

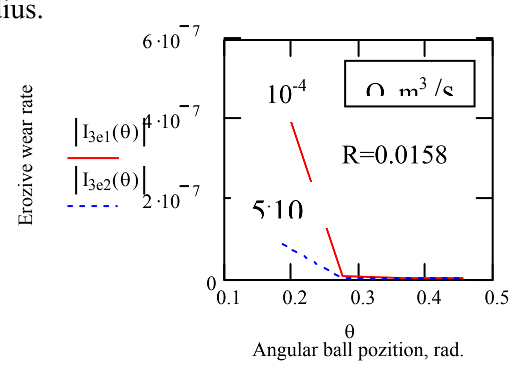

Fig.7. Erosion wear rate vs. angular ball position

The values of the ball radius and the fluid flow rate are used on the "situ" experiments.

Figures 7 and 8 show that the maximum wear rate appears on the ball and the entrance of conical seat valve.

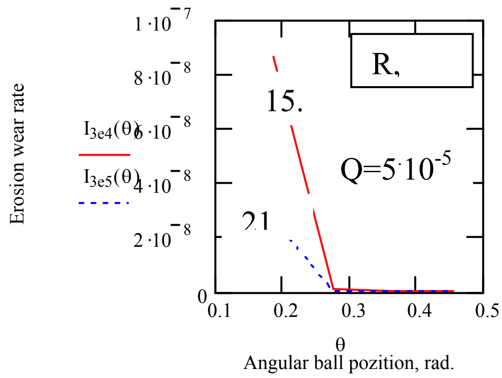

Fig.8. Erosion wear rate vs. angular ball position and ball radius

4. CORROSIVE EFFECT

The petroleum fluid corrosion of ball and seat valve in the passive region is taken to be entirely due to erosive events. The corrosive rate is assumed to be rate of repassivation over crater surfaces formed owing to erosion. The repassivation process is completely (100%) and no dissolution occurs during film formation.

The crater diameter d_c can be determined by equating the trajectory of the incident particle in penetration time. Thus

$$d_c = x_a h_{\max} \tag{14}$$

where $x_a = \left[1 - \left(\frac{1}{\mu_f \tan\left(\frac{\pi}{2} - \theta\right)} - 1\right)^2\right]^{2/5}$ and

$$h_{max} = r_p\left[1.25\pi\rho_r\theta_{el}v_o^2\cos^2(\theta)\right]^{2/5}$$

The crater depth h is significantly less than the radius of particle, then d_c and h are

$$h = d_c^2 / (8r_p). \tag{15}$$

The lateral area of the crater A will be subjected to repassivation and the mass of ball or seat corroded material (M_f) is given by

$$M_f = A h_f \rho_f = 2\pi r_p h h_f \rho_f \tag{16}$$

where h_f is the thickness of the passive film, and ρ_f is the density of passive corroded metal; these parameters will be determined by experiments.

The corrosive wear rate I_c is defined as the ratio between the passive film mass and mass of impacting particle,

$$I_c = M_f / \left(4\pi r_p^3 \rho_r / 3\right) \tag{17}$$

Fig. 9 shows the corrosive wear rate for M1 and M2 ball materials. The passive film thickness was evaluated by the corrosion test.

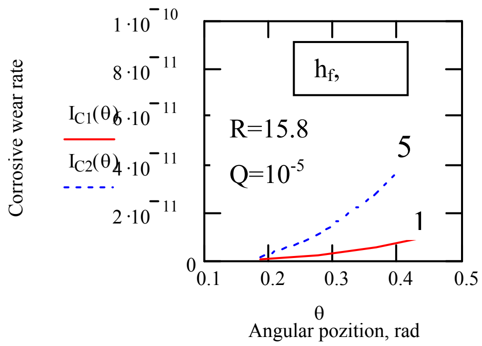

Fig.9. Corrosive wear rate versus impact angle

The equation (12) and (17) can be used to define erosion and corrosive wear map for ball and seat pumps.

5. EXPERIMENTAL RESULTS

The valve of crude petroleum extraction pump has two main components the ball and valve seat and it is shown in the Fig. 10.

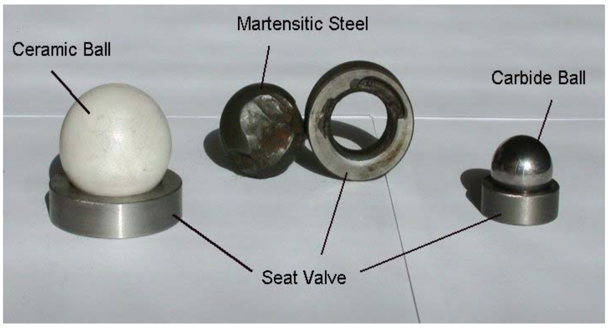

Fig. 10. Picture of ball

The experimental ball - valves were used in situ conditions for extraction of same crude petroleum in ten similar well. The experimental conditions: flow rate of crude petroleum - 41.5 m^3/ day; depth of pump in extraction tube-2380 m; temperature – 41 ^oC; dynamic viscosity of crude petroleum – 6.3 cP. The limit of working pump was considered when the crude petroleum lost in the ball valve is larger than 50 ml/day at 200-bar pressure.

The durability of ball valves is defined as the time until the crude petroleum lost has the limit value (50 ml/day).

The second experiments were used to define the corrosive wear rate. The crude petroleum fluid was used for testing corrosive wear of some materials. The corrosive wear was evaluated by mass method.

Taking into account the corrosion rate, were selected five materials for balls (M1-sintered ceramic composite: 98.5% Al_2O_3, 1% CrO, 0.5% MgO; M2- sintered ceramic composite: 91.5% Al_2O_3, 1.1% MgO, 5.55% SiO_2, 1.75% CaO, 0.06% K_2O, 0.04% Na_2O; M3- carbide: 84% WC, 8.75% TiC, 3% Ni, 4.25% Mo; M4-carbide: 91.15% WC, 8.85% Co); M5- sintered ceramic composite: 99.00% Al_2O_3, 1% Cr and two materials for valve set (G40-carbide: 80%WC, 20% Co; NG40-carbide: 80.61% WC, 19.39% Ni). The ball-valve durability of pump, in situ conditions, is shown in Fig.11 in the form of bar graphs.

The values shown are mean values of four experimental ball-valve pumps. The maximum statistical variation coefficient of durability for all experimental pumps was 0.09.

Figure 11 clearly indicates that while the ceramic composite ball M1 with the carbide G40 seat exhibits the lowest durability, the ceramic composite ball M2 with the carbide NG40 seat has the highest durability.

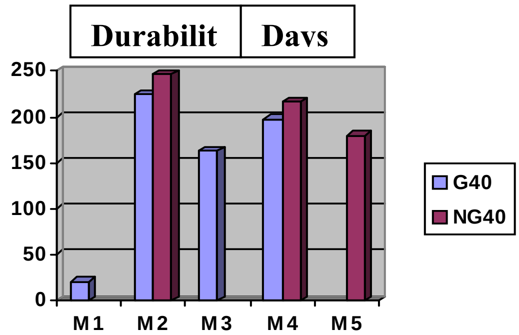

Fig. 11. Durability of ball valve pump

In the present paper had been selected six composite sintered materials, as a function of corrosive rate (standard test) of crude petroleum (Fig.12)

Figure 12 shows corrosion rate as a function of time (720 hours, 1440 hours and 2160 hours) for the ceramic sintered composite materials (M1, M2) and the carbide sintered materials (M3, M4, G40 and NG40).

It is observed that the corrosion rate is variable in time for all materials. Thus, for ceramic composites, the corrosive rate has maximum values in time of 1440 hours.

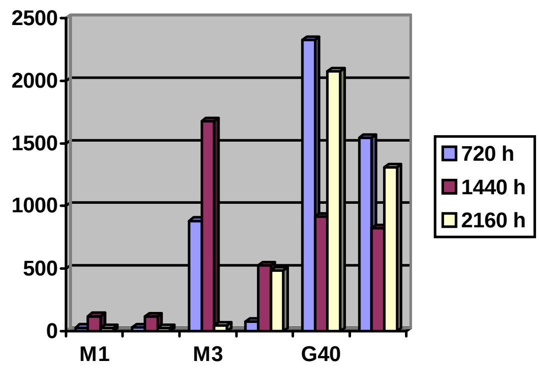

Fig. 12. Corrosion rate of ball and seat materials

5. CONCLUSIONS

1.Theoretical erosive-corrosive wear model has been used to define the possibility to increase the durability of ball and seat valve. This model is able to solve the contact problem under flow of contaminated corrosive fluid with abrasive particle. It is based on Newton fluid low in the angular gap between the ball and the seat valve.

2. Impact angle effects have been evaluated by the ball position in the gap.

3. Erosion wear rate of the ball and the seat valve depends on the fluid rate of abrasive particle, which flows in gap.

4. Corrosive effect on the erosion wear rate will modify the fatigue strength and the wear rate was increased.

5. In situ experiments have been developed for the erosion and corrosion wear of carbide and ceramic composites for ball-valve pump. The balls

6. The corrosion rate of crude petroleum fluid has been used for selection of ball and seat valve pump material.

Nomenclature

d_c	*Crater diameter*
h	*Thickness fluid film*
h_a	*Dimensionless thickness*
h_f	*Passive film thickness*
h_{max}	*Maximum crater diameter*
k_e	*Equivalent stress coefficient*
p	*Pressure of fluid*
p_m	*Pressure in a front to valve*
r_p	*Radius of particle*
R	*Ball radius*
t	*Fatigue coefficient*
v_o	*Impact velocity*
α	*Seat valve angle*
α_{ecr}	*Critic particle impact angle*
α_o	*Particle impact angle*
η	*Dynamic viscosity*
μ_f	*Friction coefficient*
θ_{el}	*Elasticity parameter*
ρ_f	*Passive corroded material density*
ρ_r	*Density of particle*
ρ_s	*Density of material target*
σ_c	*Shear strength*

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