The reduction of valve wear in proppant pumps

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1 Introduction

The problem concerns the two–stroke positive displacement pump pictured in figure 1. During the suction stroke, the plunger is withdrawn, the inlet valve opens and fluid rushes through it into the chamber. During the discharge stroke, the plunger is pushed into the chamber, causing pressures of around 10^7 to 10^8 Nm^{-2} which close the inlet valve and force the fluid out through the outlet valve.

A typical valve is shown in more detail in figure 2. Notice the shaded urethane seal which stands slightly proud of the metal face to which it is bonded; a close–up is shown in figure 3. It is the contact between this urethane annulus and the metal of the valve seat which seals the valve and prevents fluid from leaking through when the valve is closed. These seals, however, have an average lifespan of only 10 hours. Failure of the seal means that the valve "washes out" rapidly and the pump has to be shut down. The annual cost to the company runs into millions of dollars, so they are anxious to determine the mechanism whereby failure occurs, in order to design a longer-life valve.

The fluid passing through the pump is a slurry, consisting of a carrier gel (of uncertain rheology, but similar viscosity to water), and sand particles (35% by volume).

Examination of a broken seal reveals the damage to be erosion of the urethane annulus in a neighbourhood of the metal-urethane join. The "pitting" due to sand particles extends some 4 or 5 mm radially inwards, and is of a similar maximum depth. Final failure is caused by a complete break in the urethane, extending radially outwards (see figure 4).

In this report the following aspects of the problem are considered:

- Fluid Dynamics
- Particle Dynamics
- Stresses in the Seal
- Seal Failure

2 Fluid dynamics

As the plunger in the valve closes, the width of the gap through which the fluid is passing becomes small, and we might expect to be able to use squeeze film theory (see figure 5). To this end, we consider the relevant scalings and physical parameters in the flow.

$\mu = 1.5 \times 10^{-2}$ kg m^{-1} s^{-1}	(viscosity of slurry)
$\rho = 2{,}000$ kg m^{-3}	(density of slurry)
$V = 0.5$ ms^{-1}	(vertical velocity scale)
$L = 2.5 \times 10^{-2}$ m	(horizontal lengthscale)
$a = 0.4 \times 10^{-3}$ m	(radius of a sand particle)
$\delta :=$ aspect ratio of film	
$U = V/\delta$.	(horizontal velocity scale)

These values enable us to calculate a reduced Reynolds number for the film,

$$\delta^2 \mathrm{Re} \sim \delta\, 10^3\,,$$

where $\mathrm{Re} = \rho U L/\mu$. For this to be small, the gap thickness would have to be very much smaller than the particle size, which will not be the case in any regime of interest. Hence the reduced Reynolds number will be large, and we conclude that we have an *inviscid* squeeze film. We may estimate the Bernoulli pressure within this film,

$$\text{Bernoulli pressure} \sim \rho U^2 \sim 500/\delta^2 \text{ Pa}.$$

When we compare this to the Young's modulus of urethane, E, which is approximately 10^9 Pa, we see that for no realistic value of δ can the fluid pressure have any detectable effect on the urethane. We are thus led to conclude that fluid dynamics has little direct bearing on the problem. In the following section we focus on the transport of the proppant particles by the fluid.

3 Particle transport

The Reynolds number on the scale of a typical proppant particle radius is found to be

$$\mathrm{Re}_{particle} \sim 50\, \delta^{-1} \gg 1.$$

The particles are therefore transported according to an inviscid drag law, whereby the drag is proportional to the product of the fluid density, the particle cross-sectional area and the square of the velocity difference between the particle and the fluid. A simple calculation reveals that a typical timescale for a proppant particle to "catch up" with the fluid is given by

$$\frac{\rho_{particle}}{\rho_{fluid}} \frac{a}{U} \sim 10^{-3} \delta \text{ s}.$$

Since the timescale for the closing of the valve is between 10^{-3} s and 1 s, we conclude that the particles are effectively "carried along" by the fluid.

The upshot of this is that it is extremely unlikely to be possible to redesign the valve in such a way that proppant particles are expelled preferentially over the fluid. When the valve closes, particles are certain to be trapped and crushed between the urethane and the metal. The only remaining possibility for reducing wear on the seal is to reduce the stresses imposed on the urethane by the trapped particles. We discuss this aspect in the following section.

4 Stresses in the seal

Having determined that the proppant particles will be trapped in the seal it is necessary to determine how this affects the seal properties. When the valve is closed the large pressure difference across the valve causes a large load on the valve seat which is taken by the metal/metal contact on the inner annulus of the valve seat. In the absence of particles the deformation of the urethane is therefore determined by its shape, the shape of the valve seat, its elastic properties and any bonding to the valve. The subsequent deformations due to the particles can be considered as a localized variation to the stresses created by this deformation.

The design of the urethane is expected to ensure the following criteria are met.

- The normal stress of the urethane against the valve seat exceeds the working pressure drop across the valve to generate a seal
- The urethane can deform sufficiently to ensure that there is a metal/urethane seal around any particles that remain in the gap after closing (those particles left in the metal/metal part of the seal gap are crushed and create a sandstone layer that is known to allow leaking)

When a proppant particle is trapped beneath the urethane, as depicted is figure 6, high stresses will be generated in the neighbourhood of the particle. If these stresses exceed the yield stress of the urethane, plastic deformation (*i.e.* the "pitting" observed on worn seals) will result. We can expect the induced stresses in the urethane to be greater if the particle is close to the bond between the seal and the metal shown in figure 3, as there the urethane will be (in some sense) less compliant. This is a possible explanation for the concentration of wear on the seal in a neighbourhood of the urethane–metal bond.

Ideally, a finite element package could be employed to analyse the stresses in the seal in some detail. We restrict our attention to a few very simple model problems, whose solutions, though clearly not truly representative of the situation in practice, will allow us to propose some possible improvements to the seal design.

The first models will attempt to give insight into the stresses in the urethane when there are no particles and the second set of models consider the local deformation due to a particle.

4.1 Models of the seal deformation

Start by ignoring the proppant and simply consider the stress caused by closing the seal shown in figure 7. A simple model for this is two–dimensional linear elasticity in plane-strain. To generate simple analytical expressions for the the normal stress on the urethane/metal boundary consider the urethane to be a quarter space with the urethane/metal boundary at $y = 0$ and where the tangential stress is assumed to be zero and the displacement is specified.

4.1.1 Square seal with bonding

A first model is to take the displacement to be uniform, 1, corresponding approximately to the existing seal neglecting the local chamfer. The urethane is bonded to the metal valve and therefore the displacement is zero on the bond $x = 0$. This simple problem admits a similarity solution, in which the normal stress on $y = 0$ is given by

$$\sigma_{yy}(x,0) = -\frac{4E(1-\nu)}{\pi(1+\nu)(3-4\nu)}\frac{1}{x}, \tag{1}$$

where ν is Poisson's ratio. As may have been anticipated, there is a stress singularity at the origin. Because the current seal design (see figure 3) is similar to this model, this implies that the bulk of the normal stress (which, in fact is what makes the seal effective) is generated near to the urethane–metal bond; the rest of the urethane actually does little work in terms of sealing the valve. This opens up the possibility of redesigning the seal shape in order to distribute the stress more evenly over the seal while ensuring that the normal stress remains sufficiently high to retain good sealing properties.

4.2 Local deformation due to a particle

Having determined the overall stress field, now consider the local deformation of the urethane due to a particle trapped in the seal gap. Take the particle to be isolated from other particles, for the urethane to be a half space with the stresses at infinity given by the stresses on the boundary as determined from the no particle problem. The fluid has adequate time to drain away and there are therefore three parts of the urethane surface to consider. i) the part in contact with the particle, ii) the part in contact with fluid and iii) the part in contact with the metal valve seat. Where contact with a solid surface occurs it is necessary to give an additional condition about the shear stress and in the first instance it would be appropriate to consider a sliding contact with no tangential stress. To give some insight into the resulting local stresses the case of a given particle shape, with no region ii), can be considered and an example in three dimensions is given here. The complete problem is then discussed briefly.

4.2.1 A parabaloid of revolution

As a simple "first stab" at the full contact problem depicted in figure 6, consider the problem of finding the displacement of an elastic half-space into which a paraboloid of revolution is pushed. There is a well-known exact solution for this problem (see, for example, "*Contact Mechanics*" by K.L.Johnson, page 61), wherein the vertical displacement is given by

$$u_z = \begin{cases} (2a^2 - r^2)/a, & r \leq a, \\ \frac{2}{\pi a}\left((2a^2 - r^2)\sin^{-1}(a/r) + a(r^2 - a^2)^{1/2}\right), & r > a, \end{cases} \tag{2}$$

while the pressure applied by the paraboloid is

$$p = \frac{4E}{\pi a(1-\nu^2)}(a^2 - r^2)^{1/2}, \quad r \leq a. \tag{3}$$

This displacement is plotted in figure 9.

From this solution we can estimate the distance r_ϵ from the particle at which the displacement will have fallen to some small multiple, ϵ, of the displacement at $r = 0$:

$$r_\epsilon \sim \frac{4a}{3\pi\epsilon}\left[1 + \frac{1}{10}\left(\frac{3\pi\epsilon}{4}\right)^2 + O(\epsilon^4)\right]. \tag{4}$$

A simple argument for the concentration of pitting near the metal–urethane bond based on this simple solution might run as follows. We can expect the proximity of the bond to have little effect on the stress so long as it is far enough away from the particle that the displacement predicted by (2) is small. Alternatively, we might conjecture that the increased stress due to the proximity of the bond will be sufficient to cause pitting at distances r_ϵ away from the bond where ϵ exceeds some critical value. Rigorous justification of such an approach and determination of the appropriate critical value of ϵ clearly requires some more subtle analysis.

4.2.2 A sphere

Though the sand particles are of various shapes they are in general relatively spherical so we shall use this as an idealized particle. Consider the contact problem, depicted in figure 6, between an elastic half–space and a sphere on an infinite plane. This can clearly be expressed as a variational problem, and hence should be amenable to finite element methods. We require the relationship between the net applied stress (the difference between the normal stress created by deforming the urethane and the fluid pressure) and the contact distance d; we anticipate that this relationship will resemble that shown in figure 10. Note that the deformation depends on the pressure in the fluid and for a given normal stress will be maximum when the pressure is at a minimum. This deformation will not create pitting if the maximum tensile stress remains below the material yield stress $Y = ME$ (here the material parameter M is typically in the range 10^{-2}–10^{-1} for urethane).

5 Seal failure

It is likely that when the seal fails by splitting along a radius, it does so very rapidly. The evidence for this is firstly that no "half–split" seals have been observed and secondly that the seal always splits at exactly one place. We therefore conjecture that the seal failure is the result of a fracture in the urethane which propagates rapidly due to the high hoop stresses caused by the pressure drop across the valve. This obviously needs to be more carefully examined as the largest hoop stresses occur at maximum working pressures and are compressive for both inlet and outlet valves but does give a mechanism for sudden failure. As the urethane becomes pitted (via mechanisms discussed in the previous section), the pressure drop is concentrated on a diminishing effective sealing area; when this area becomes too small, rapid failure of the seal will follow.

6 Suggestions for experiments

In this section we suggest some experiments which could be carried out to verify some of the conjectures we have made.

1. "Sand–blasting"

Our assertion that the fluid mechanics of the problem has little direct effect on the failure of the seal could be tested by cutting a small radial groove in the seal. Simple calculations suggest that fluid will flow faster down such a groove, which provides a possible cause of instability: groove → faster flow → more erosion → deeper groove. According to our analysis, such effects should be negligible.

2. "Artificial wearing"

We suggest laboratory pressure–testing of seals from which chunks similar to those which are worn away during pitting have been excised. We anticipate, as described in the previous section, that when the amount of urethane cut out exceeds some critical value, rapid radial fracture will result when the valve is put under pressure.

3. Statistics of failures

More could be deduced about the probable mechanisms for seal failure if more was known about how typical failure times are distributed about the estimated average of 10 hours.

7 Possible design modifications

Given the understanding of the mechanism occurring within the seal we suggest some alternative seal designs which may reduce seal wearing, though each has its own drawbacks.

1. Remove metal–urethane bonding

The stress singularity at the interface between the urethane and the metal could be removed by simply removing the bonding along part of the join. This idea has the disadvantage that particles may force their way into the join and cause wear that way.

2. Make the exposed part of the urethane flat

The sand–blasting effect described briefly in the previous section could be substantially eliminated by making the urethane flush with the metal. The seal could then be effected by making the valve seat have the necessary height variations as shown in figure 11 and the metal would be less prone to erosion by the particles. The disadvantage with this design is that there will still be high stresses at the urethane–metal bond, and also that sand–blasting is probably not an important effect anyway !

3. Use alternative seal profile

The design shown in figure 8 eliminates the stress singularity at the urethane–metal bond. In addition the normal stress in the urethane at the valve seat interface can be made more uniform by suitable shaping of the urethane (particularly making it linear from the inner radius and then superlinear as the outer radius is approached). By suitable choice of the urethane material used it should then be possible to ensure good sealing properties while not creating the high local stresses that cause pitting. If the stresses remain too high it may be possible to accommodate the particles by making both the valve and the valve seat out of urethane over the sealing section. This would create a seal with greater compliance for the same normal stress with less (approximately half) the resulting maximum tensile stress).

PDH, CPP, LJC, JRK, JB-S, TGM, ADF, JRL, WS.....

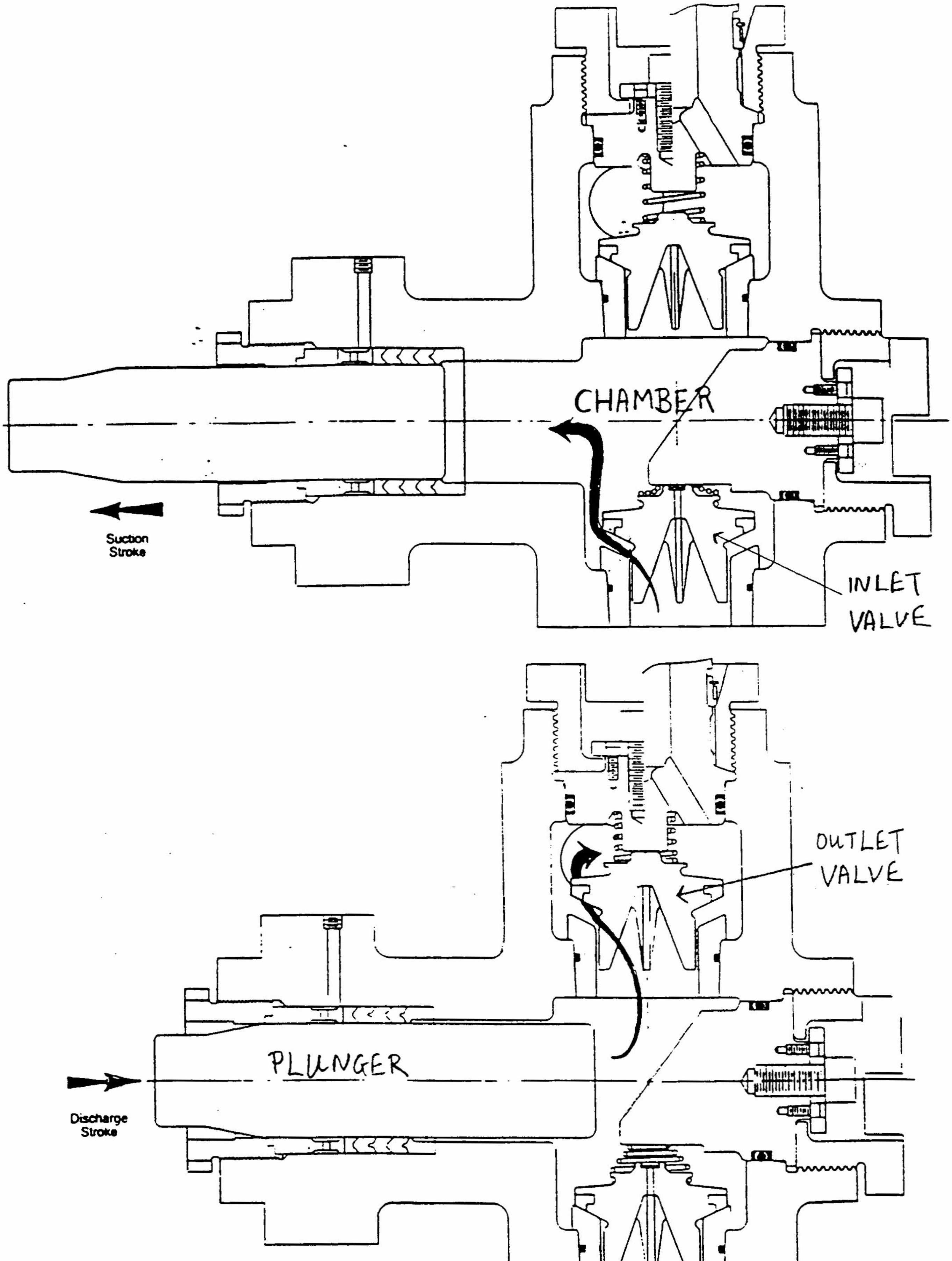

Figure 1: A two–stroke positive displacement pump

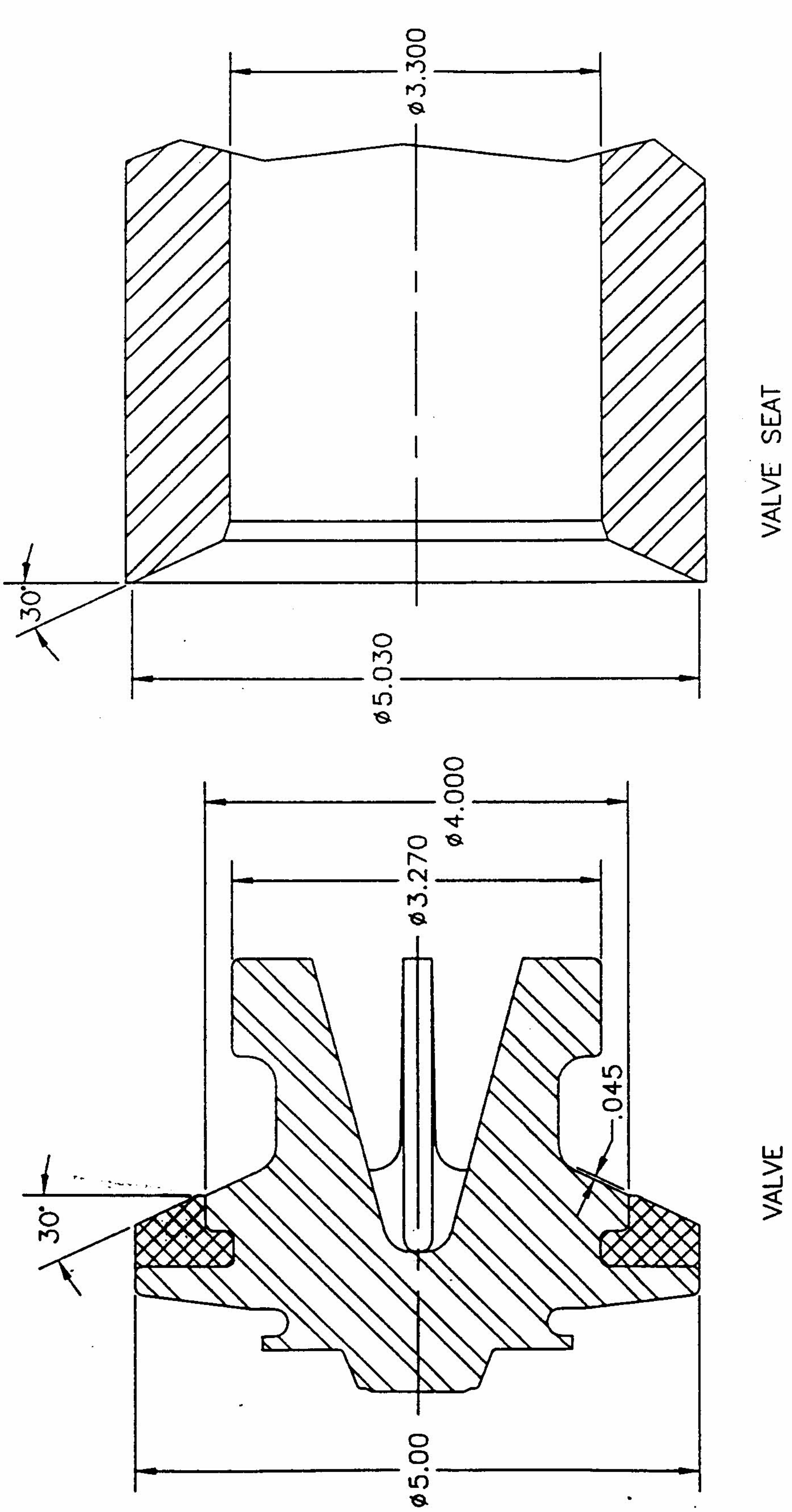

Figure 2: Detail of a valve

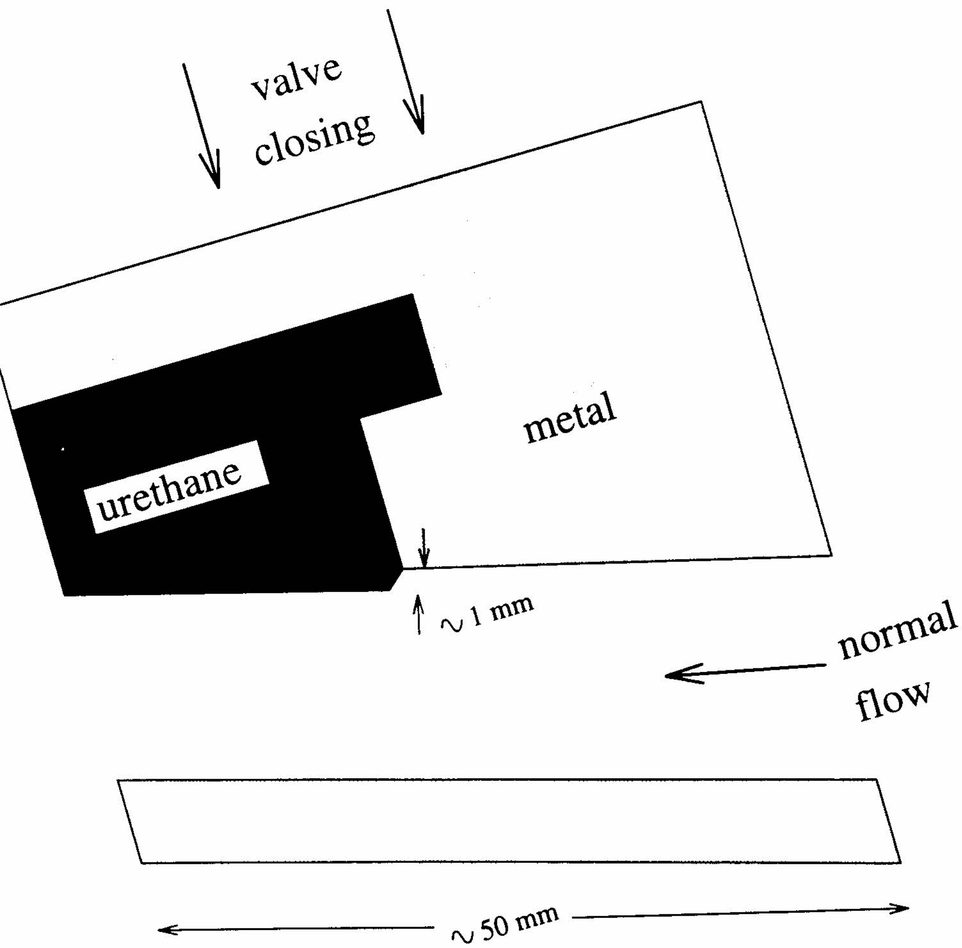

Figure 3: Detail of the valve showing the urethane seal

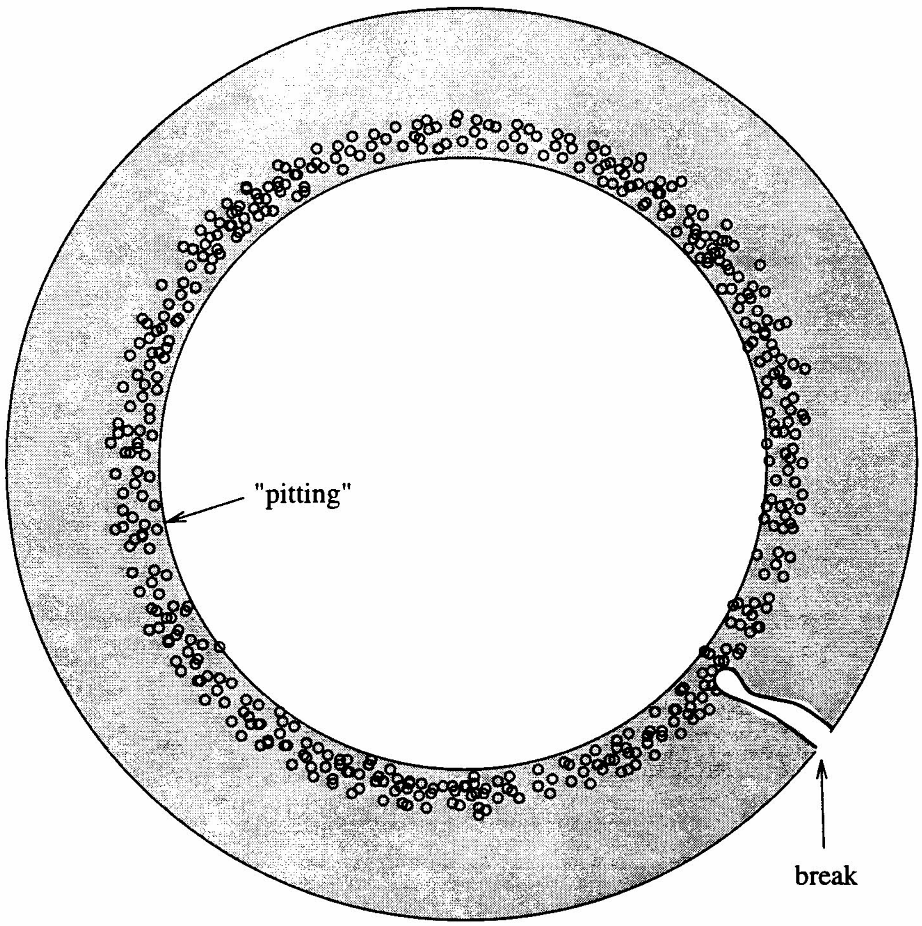

Figure 4: Schematic diagram of the wear pattern in a failed seal

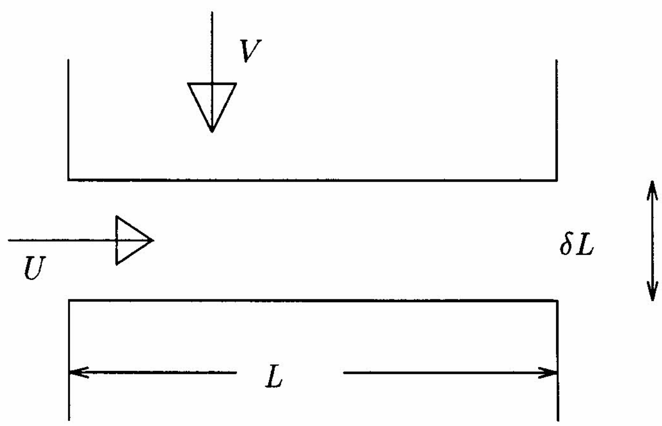

Figure 5: Definition sketch for the inviscid squeeze film

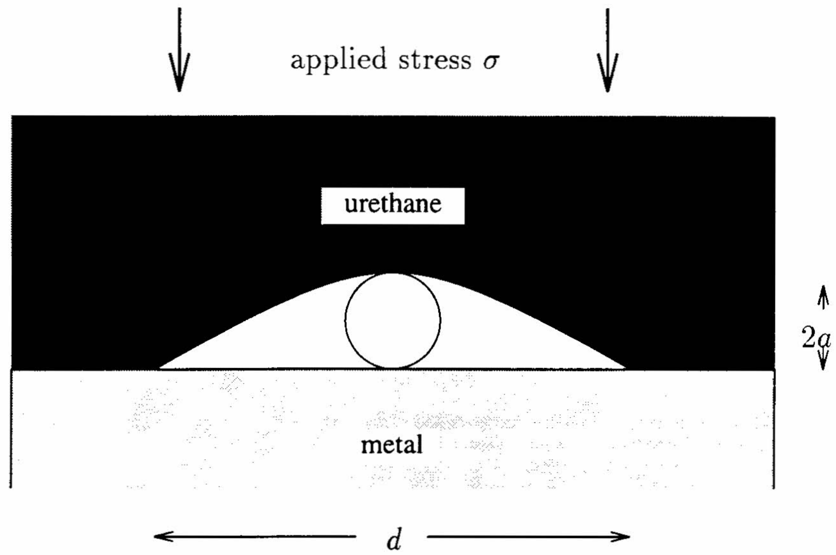

Figure 6: Schematic diagram of a trapped proppant particle

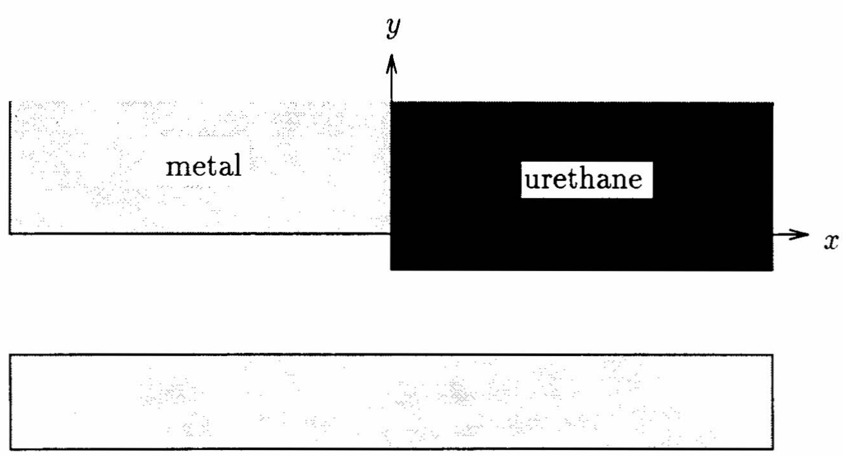

Figure 7: A two–dimensional seal design

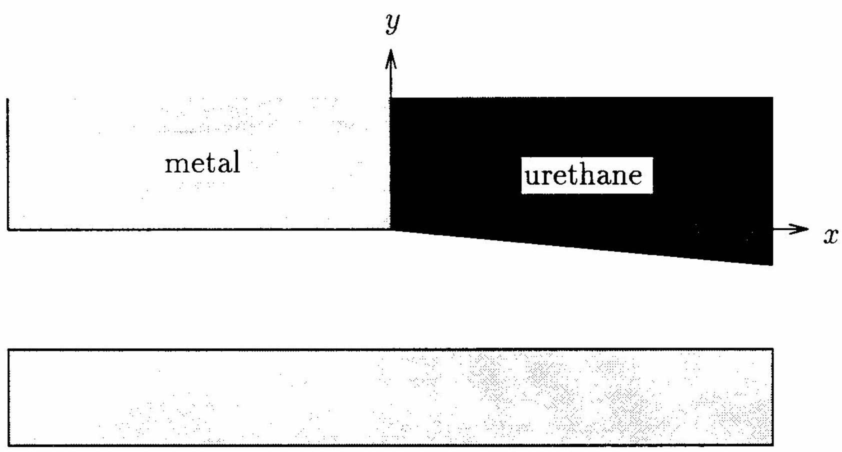

Figure 8: An alternative two–dimensional seal design

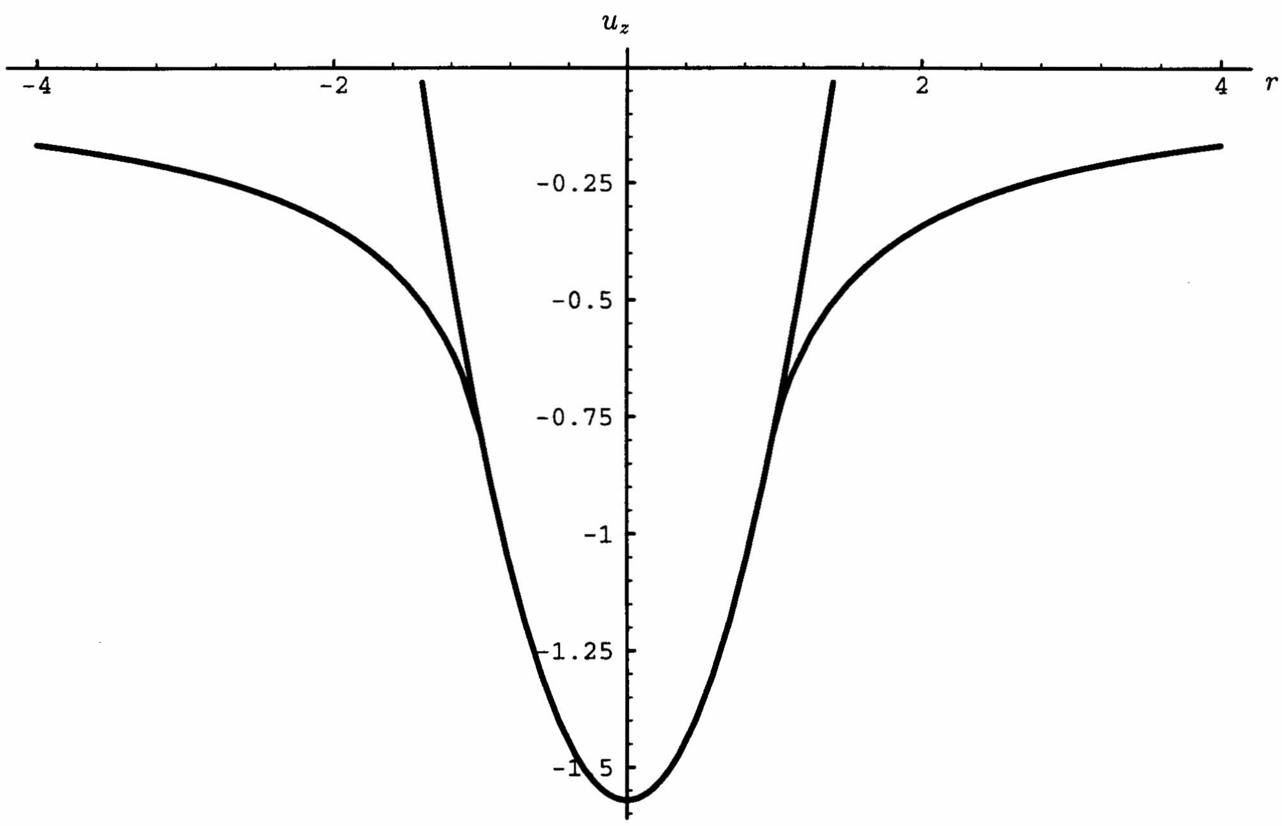

Figure 9: The penetration of a paraboloid into an elastic half–space

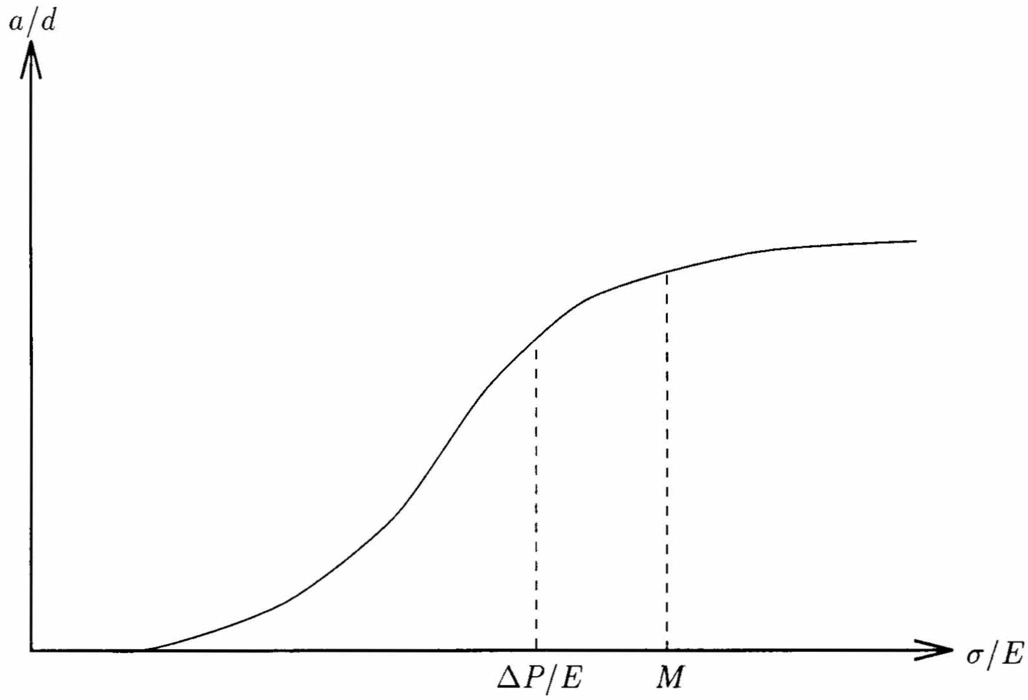

Figure 10: The relationship between applied stress and contact distance

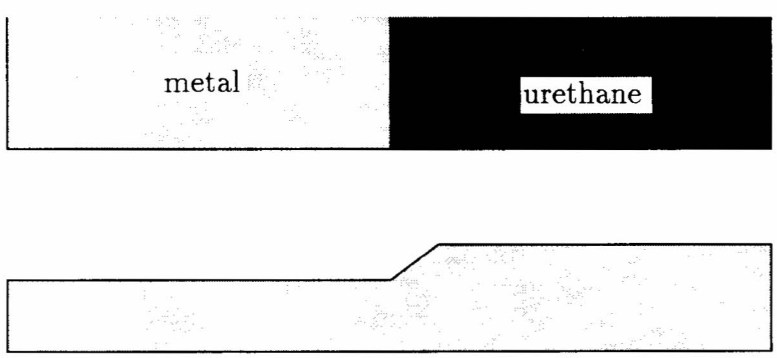

Figure 11: A seal design to eliminate sand–blasting