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Defining Impact

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ABSTRACT

Describing impact in terms of sonic velocity provides a dissection tool for this complex event. Impact velocity induces a proportional stress in the colliding objects. For flapper valves we also have viscous fluids at the interface, inviting stiction and cavitation. Stress in the reed develops at the speed of impact, while fluid layer dilation may take somewhat longer. Viscosity alters anchoring of the reed membrane just long enough for the stress to develop. A wider seat increases the clamping force in proportion to the area and the ambient viscosity. With the valve temporarily held down, reverse bending stress develops under gas/air pressure and inertia force. Regression of the test data supports this scenario. In the presence of any fluid, stiction effects match impact, and both participate equally in the outcome recorded as impact failure. There may be no such thing as pure impact in the micro world of flapper valves.

1. INTRODUCTION

The first human tool was made by hitting rock against rock to produce a cutting edge. Since then impact has been put to good use without knowing what makes it work. An understanding of impact has been lacking. Involving the sonic speed as the primary mechanism adequately defines impact in terms of time the compressive shock wave takes to traverse the material in both directions. What's left is to clarify what constitutes a definition. That has been dealt with in the past: there are some respectable references spelling out the terms:

Aristotle (384-322 BC) is credited with saying: "One ought not try to prove the obvious via less obvious".
1500 years later, William of Occam (1285-1345) follows with: "Entities should not be multiplied unnecessarily".
The simplest explanation that works is sufficient.

Dan O'Keeffe, (1999) of Camberwell Grammar School in Australia teaches his students:
" In fairly elastic impact, the duration of impact is determined **only** by the time it takes for the compression wave, produced at the beginning of the impact, to travel through the object, bounce off the other end, and return to the point of impact, at which time the object leaves the surface. The speed of the compression wave is, of course, the **speed of sound** in the material."

This description meets Occam's principle. Duration time fully defines impact. Speed of sound is the means.
Futakawa and Namura (1980), in a detailed impact study of two steel bars of different lengths and crossections, conclude on page 281: "…Since at this moment this reflect wave has reduced the pressure between the two bars to zero, the bars … separate and the impact is over."

The last two quotes describe large-scale events: a ball bouncing, or two massive metal bars hitting. On the smaller scale of flapper valves, this pure physics is contaminated by stiction and cavitation.

2. SHOCK WAVE TERMS

When the reed hits the anvil, an acoustic **shock wave** radiates through both materials in three directions:
- A compressive, square shape wave reflects back toward the top of the reed to bounce it off the anvil.
- A tangential wave bulges the edge of the reed. When the hit occurs near the valve edge, the apex of the bulge may exceed the elastic elongation limit of the material with regularly spaced cracks developing at the periphery. Böswirth (1980), p.200, postulates that the contact front of the shockwave exceeds the speed of sound at a shallow hit angle and generates sonic boom with the resulting stress. That corresponds to the notion of whiplash.

- A surface wave, which also works in a whiplash fashion. Russel (1999) generated an applet showing wave animations, and he describes the motion:
"The particles in a solid, through which a Rayleigh surface wave passes, move in elliptical paths, with the major axis of the ellipse perpendicular to the surface of the solid. As the depth into the solid increases the width of the elliptical path decreases. Rayleigh waves are different from water waves in one important way. In the water wave, all particles travel in clockwise circles. However, in a Rayleigh surface wave, particles at the surface trace out a *counter-clockwise* ellipse, while particles at a depth of more than 1/5th of a wavelength trace out *clockwise* ellipses."

The ratio of impact velocity to the speed of sound is a measure of strain. Surface speed of sound being the lowest of the denominators sets up the maximum stress potential.

$$\varepsilon = \frac{\sigma}{E} = \frac{v_i}{C_c} = \frac{v_i}{C_s} \tag{1}$$

Where: **ε –** strain ratio, **σ** – stress, **E** – Modulus of Elasticity in tension, $\mathbf{v_i}$ – impact velocity, [m/sec]
The speeds of sound: $\mathbf{C_c}$ – compressive, $\mathbf{C_t}$ **–** tangential, $\mathbf{C_s}$ – surface.
With strain **ε** tying together the varied aspects of impact velocity $\mathbf{v_i}$ and stress **σ**, any pair of equations in (1), constitutes a viable relationship.
Sonic velocity equations in steel for the three cases: (CS20 *to* SS716 numbers given within equations).

Compressive:
$$C_c = \sqrt{\frac{E}{\rho}} = 5172\,to\,5064;[m/\sec] \tag{2}$$

Tangential:
$$C_t = \sqrt{\frac{G}{\rho}} = 3178;[m/\sec] \tag{3}$$

Surface:
$$C_s = \frac{C_c}{\sqrt{3}} = 2986\,to\,2924;[m/\sec] \tag{4}$$

Where: E – Modulus of elasticity in tension. For CS20: E = 210 [GPa], and SS716: E = 200 [GPa],
G – Modulus of elasticity in torsion. G = 79.3 [GPa]
ρ – Material density. ρ = 7850 [kg/m^3] ρ = 7800 [kg/m^3]

The combination of equation (1) and (2) results in
$$\sigma_c = v_i \cdot \sqrt{\rho \cdot E} \approx v_i \cdot 40,[MPa] \tag{5}$$

Providing a shortcut to tie down the compressive stress $\mathbf{\sigma_c}$ to impact velocity $\mathbf{v_i}$ and the physical properties of the reed material. The superposition of the reflected wave doubles this stress. That is still too low to account for the breakage due to impact. Machu (1992) points: "An oblique hit sets a moment driving the opposite side toward the seat". That also generates whiplash: cracking the speed of sound is the prime characteristic of it. The next question is to find out what are the real reasons for impact fatigue failures.

The compressive wave layer thickness is assumed equal to elastic strain deflection **ε*h**, **h** being the reed thickness. At the moment of impact all the kinetic energy of the hit is condensed into the strain layer **ε*h** by a factor of **1/ε**, or around thousand-fold. The ratio of impact velocity to the compressive speed of sound provides the focusing mechanism. With the magnitude of strain **ε ~ .001**, 99.9% of the material thickness can otherwise be considered free of stress and simply provides a medium for the shock wave to travel back and forth.

2. THE MECHANICS OF IMPACT

2.1 Reviewing basics

This review is conducted on impact tests by Dusil and Johansson (1980).
The combination of a thin reed and high compressive sonic velocity results in impact time measured in hundreds of nanoseconds. (10^{-9} sec). In a typical h = .381 mm thick reed the one-way shockwave travel time is **τ.** The complete impact event takes twice as long.

$$\tau = \frac{h}{C_c} = \frac{.381}{5118 \cdot 10^3} = 74.4 \cdot 10^{-9};[\sec] \tag{6}$$

The deceleration **a = dv$_i$/dt** enters the picture. Only the strained layer carries the shock wave energy. At an arbitrary strain **ε ~ .001** the impact velocity is **v$_i$ = 5.118** [m/sec], and the deceleration **d$_c$** defined in units of gravity acceleration; **g = 9.81** [m/sec]

$$d_c = \frac{a}{g} = \frac{5.118}{74.4} \cdot \frac{10^9}{9.81} = 7.008 \cdot 10^6; [g's] \qquad (7)$$

The magnitude of this number deserves consideration. With the moving mass of reference reed **m$_i$** = 2.85 gram, only the strain layer portion of the mass is subject to deceleration **a**. For **ε = .001** this force is:

$$F = m_\varepsilon \cdot a = 2.85 \cdot 10^{-6} \cdot \frac{5.118 \cdot 10^9}{74.4} = 196; [N] \qquad (8)$$

F = 196 [N] (44.1 lbs) is obtained, with the mass **m$_\varepsilon$ = m$_i$*ε**. At impact, this full force hitting the seat stresses both the reed and the anvil equally, per Soedel (1974), p. 321; his equation (32) for stress concentration factor is quoted:

$$S_{hH} = \frac{\rho_h C_h}{1 + \dfrac{\rho_h C_h}{\rho_H C_H}} \qquad (9)$$

Where **h** and **H** respectively refer to the reed and the anvil (seat), **ρ** – density, and **C** – compressive speed of sound. When multiplied by the velocity **v$_i$** of the reed operating at its first operating frequency, **S$_{hH}$** becomes maximum stress under the assumed collinear impact. The impact velocity of the reed is determined by its first bending mode.

$$v_i = \omega_n \cdot y \qquad (10)$$

Where **y** – is the reed tip deflection, and the radial frequency **ω$_n$**:

$$\omega_n = \sqrt{\frac{k}{m}} \qquad (11)$$

With **k** = spring constant of the valve [N/m] and **m** = moving mass of the valve, [kg]
The actual operating frequency of the reed will be .88 to 1.0 of **ω$_n$**. Since it is the maximum impact velocity doing the damage, the full value of **ω$_n$** should be taken for initial design. If the valve motion trace is available, it can be used to check the above estimates as well as demonstrate presence of stiction. In any case the natural frequency will provide an asymptotic envelope. Harmonic resonance also needs to be reviewed, as the damping of freely oscillating reed is rather low, under .06. See Wylie & Barrett (1995), pp. 447-453 for treatment, especially the Fourier analysis, pp. 497-530, in particular p. 528, dealing with spring-mass system driven by periodic square wave force. With the positive force represented by odd-numbers in half-range sine expansion, the 3- and 5-th harmonics are most likely to be involved in harmonic resonance.

2.2 Analyzing impact test data

Dusil and Johansson (1980) have tested .381 mm thick reed. Seats widths of .5, 1.0, and 2.0 mm, with the anvil OD varying from 17.5 to 20 mm were tested. The sketch of steel anvil used in testing is shown below after Svenzon (1976), p. 70. Operating natural frequency is given at f_n = 250 Hz. Compressed and filtered air at 90^{0}C (194^{0}F) was used. The test coupons used were 100 mm long, 20 mm wide, the tip radius of 10 mm, working cantilever length of 50 mm, and the thickness of .381 mm. The two parameters subject to change were the port dia **d**, and the seat width **x**. The overhang distance **c** completes the picture of the tested reed geometry. See **Figure 4** on page 5.

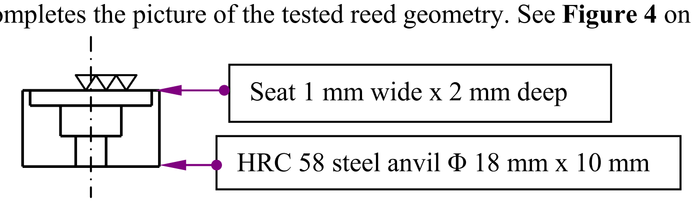

Figure 1. Shape of the anvil.

Table 1. Dusil and Johansson (1980) impact test results with the port and seat geometry added.

Test #	*Ov'lap z mm*	Ov'hang (c) = z+x mm	Port ID d = 20-2c mm	Seat OD D = d+2x mm	Seat area Ac mm^2	Overhang Ratio (c/d)	Seat Ratio (D/d)^2	*Impact fatigue limit "a" test data From Dusil & Johannson (1980), p. 370* *20C carbon steel*			*SS716*	*Std Dev s/a*
							Series number:	**1**	**2**	**3**	**4**	
							Seat width x:	***0.50***	***1.00***	***2.00***	***1.00***	
1	*0.0*	0.5	19.0	20.0	30.63	0.0263	1.1080	***0.56***				***0.07***
2	*0.5*	1.0	18.0	19.0	29.06	0.0556	1.1142	***0.84***				***0.12***
3	*0.0*	1.0	18.0	20.0	59.69	0.0556	1.2346		***0.64***			***0.07***
4	*0.0*	1.0	18.0	20.0	59.69	0.0556	1.2346				***0.81***	***0.20***
5	*1.0*	1.5	17.0	18.0	27.49	0.0882	1.1211	***1.24***				***0.13***
6	*0.5*	1.5	17.0	19.0	56.55	0.0882	1.2491		***0.96***			***0.12***
7	*0.5*	1.5	17.0	19.0	56.55	0.0882	1.2491				***1.13***	***0.13***
8	*1.3*	1.8	16.5	17.5	26.70	0.1061	1.1249	***1.48***				***0.15***
9	*0.0*	2.0	16.0	20.0	113.10	0.1250	1.5625			***0.67***		***0.10***
10	*1.0*	2.0	16.0	18.0	53.41	0.1250	1.2656		***1.32***			***0.09***
11	*1.0*	2.0	16.0	18.0	53.41	0.1250	1.2656		***1.32***			***0.07***
12	*1.0*	2.0	16.0	18.0	53.41	0.1250	1.2656		***1.31***			***0.07***
13	*1.0*	2.0	16.0	18.0	53.41	0.1250	1.2656				***1.41***	***0.10***
14	*1.0*	2.0	16.0	18.0	53.41	0.1250	1.2656				***1.48***	***0.30***
15	*1.3*	2.3	15.5	17.5	51.84	0.1613	1.2747		***1.70***			***0.15***
16	*1.3*	2.3	15.5	17.5	51.84	0.1613	1.2747				***1.82***	***0.24***
17	*0.5*	2.5	15.0	19.0	106.81	0.1667	1.6044			***1.14***		***0.13***
18	*1.0*	3.0	14.0	18.0	100.53	0.2143	1.6531			***1.68***		***0.23***

The two following graphs map out linear dependency of impact fatigue strength on port and reed geometry.

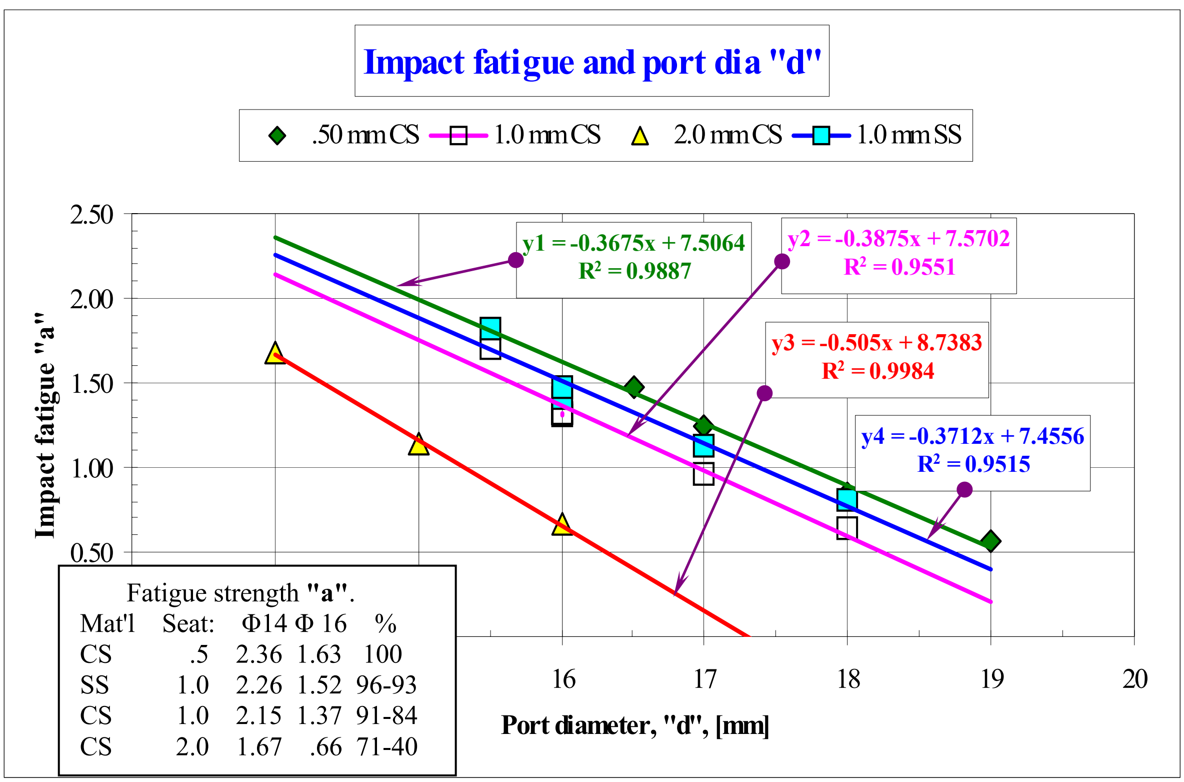

Figure 2. Impact fatigue limits "a" versus port diameter **"d"** at different seat width.

Similar treatment is given to the overhang **"c"** to account for the remainder of the 20 mm wide reed.

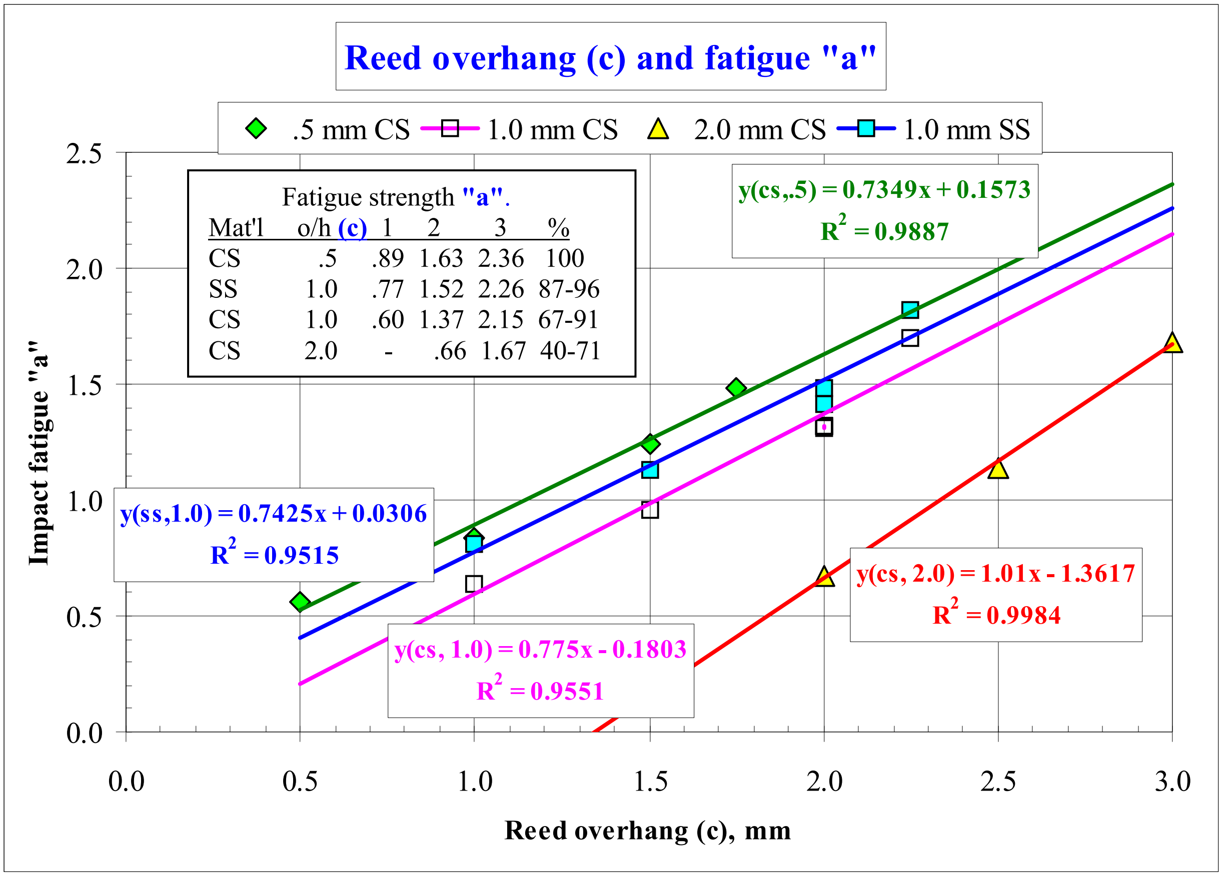

Figure 3. Impact fatigue limits versus reed overhang **(c)**

Presence of stiction offers a plausible explanation for this highly regular behavior. That involves consideration of the transient time scale of the events. At this point the definition of impact becomes important.
Both graphs linearly relate impact fatigue strength to geometric features of port, seat and the reed. Slopes of all lines are consistent, a signature of common physics. The underlying quadratic equations optimize seat width at .45 mm.

Figure 4 shows a stage just after the collinear impact as the reed deflects into the port under the inertia load.

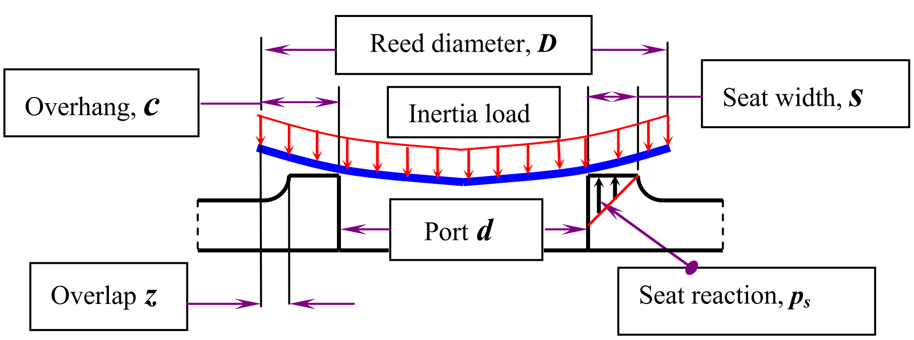

Figure 4. Valve under uniform load.

2.3 Stiction

In order to determine the role of stiction, an examination of valve displacement versus time is necessary. **Figure 5** shows one of four graphs used for that purpose. Oil dilation line **dh/dt** at the opening of the valve is visible. The summary of the remaining 3 graphs is presented in **Table 2**.

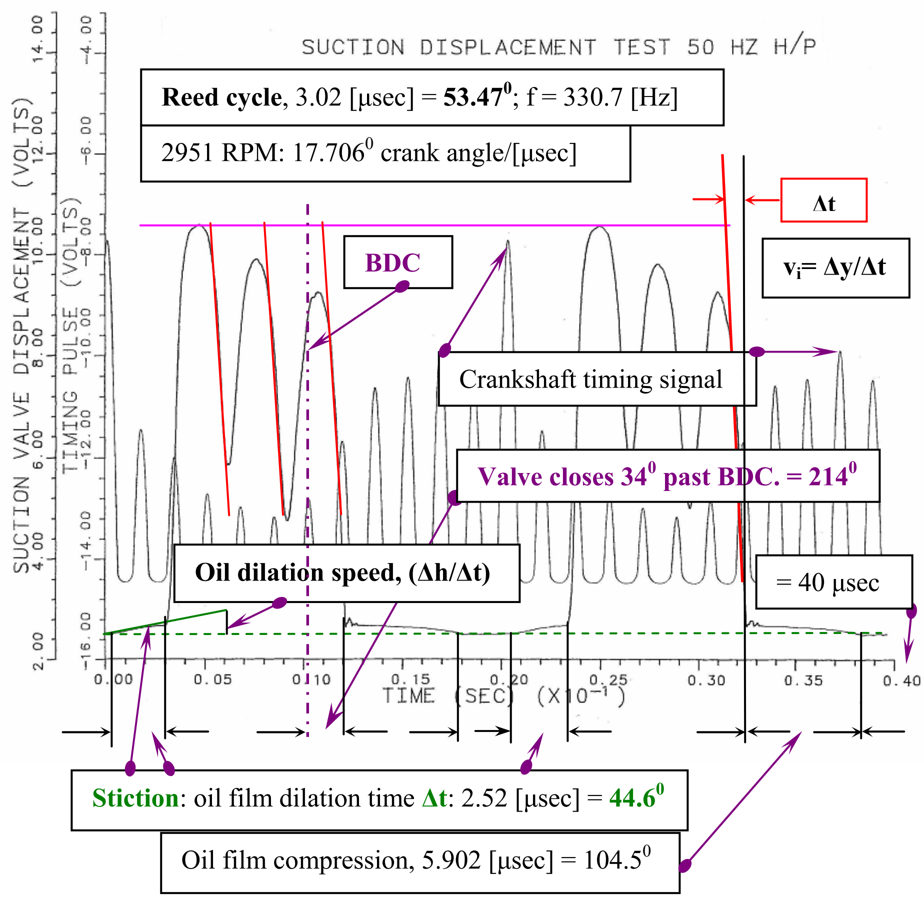

Figure 5. Suction valve signature at 50 Hz, HP operation showing stiction.

Table 2. Summary of suction valve signatures with respect to stiction timing.

Test condition	AC	AC	HP	HP
Condition, [Deg F]	45/130/65		30/110/50	
RPM	3444	2851	3544	2951
Est. temperature, [^{0} F]	95.7	110.8	80.7	95.8
Est. temperature, [^{0} C]	35.4	43.8	27.0	35.4
Dilation Time, [μsec]	1.477	1.863	1.981	2.515
Dilation Time, crank angle	40.9	33.0	42.1	44.5
Est. 1GS oil viscosity, [cSt]	10	7.4	14	10

Listed suction valve tests were not focused on stiction, so temperatures and viscosities are approximate. 1GS oil viscosity is used in the table for illustration: all oils exhibit similar temperature dependence.

Stiction guarantees the collinear impact stage when the reed stays glued to the seat until the gas force pops it off. The stress responses of the reed take place at the real time of impact. The full impact event lasts 148.8*10^{-9} [sec]. The oil layer dilation time has been clocked at 1.4773 to 2.5145 [10^{-6} sec], as shown in table above.

The longer duration of stiction radically changes the perception of impact from a purely dynamic event into one where viscosity is an equal, if not a prevailing participant. Dusil and Johansson (1980) tests had been conducted with filtered air at 90^0C, probably as free of oil as possible. There must have been enough stiction present to result in the graphs and regressions above. The higher viscosities encountered in compressors do make the problem worse, particularly in low temperature applications. Khalifa and Liu (1998), p. 89, in equation (15) establish the force of stiction:

$$F_v = +\left(\frac{3\pi\mu}{2h^3}\frac{dh}{dt}\right)R_i^4\left(1 - X^4 + \frac{1-2X^2+X^4}{\ln X}\right) \qquad (12)$$

Where the first bracket contains the viscous layer parameters: dynamic viscosity **μ**, [Pa.sec], oil film thickness **h**, [m] and dilation speed, **(dh/dt)**. The second bracket term manipulates the seat contact geometry: $\mathbf{R_i}$ port radius, [m] The factor $\mathbf{(X^2 = 1 + A_c/A_p)}$ is the ratio of contact to port area; it converts into seat **OD/ID** ratio: $\mathbf{X = (OD_s/ID_s)}$.

The sketch below illustrates interaction of forces and the reactions of the seat and the reed membrane.

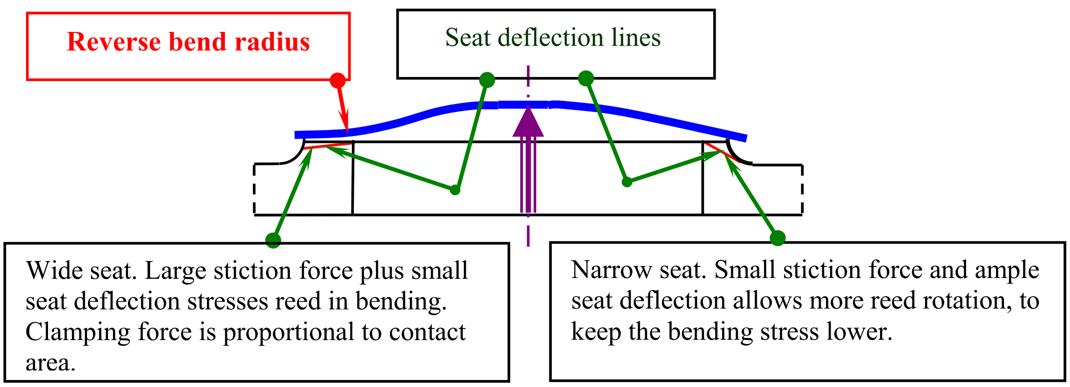

Figure 6. Reverse bending over wide and narrow seat.

DISCUSSION

Stiction over a wide seat (left) holds onto the edge while the central portion of the reed alternately bulges outward under the mounting gas pressure, or caves into the port under the force of inertia. At the opening, gas pressure generates **Figure 6** configurations. While stiction slowly relents its grip, and the oil film dilation takes comparatively longer, various stresses in the reed form at all 3 sonic speeds. This sequence is repeated every valve cycle. With the oil film thickness $\mathbf{h^3}$ in denominator of equation (12) at zero value, the clamping force approaches infinity. The right side of **Figure 6** shows the relative benefit of narrow seat.
Under the deceleration of the shocked layer in millions of g's, and stiction force near infinity, the reed behaves like a membrane flipping up and down with the alternating forces of gas pressure and inertia. The wide seat prevents the rotation of the valve edge and becomes a liability: Shallow deflection of the wide seat combined with high stiction force limits the angular motion of the reed and forces the bending strain in the reed to compensate.

Footnotes:
1. Oil layer dilation time for **HP** is consistently ~35 % longer compared to **AC**. Oil viscosity increases with lower operating temperature. This confirms the presence of stiction. The very substantial crank angle dilation time, 33 to 45 degrees is rather large and reflects the geometry of the components used.
2. The oil layer dilation time appears proportional to the square root of viscosity.

CONCLUSIONS

1. Impact is defined by the time it takes for the compressive shock wave to traverse the object of impact. Sufficient proof to that effect is furnished to validate this definition.
2. Existing impact fatigue tests show primary dependence on the reed, port and seat geometry.
3. Duration of the stiction events overlaps the stress development time by a sufficient margin to produce the graphed results. Real time range of nano- and microseconds apparently sets an adequate stage even at the minor ambient viscosity of the air driven test machine. This makes stiction an equal partner in any real impact problem.
4. There may not be such a thing as pure impact in the real world of steel flapper valves.
5. Cavitation, occurring in the presence of liquid at the contact faces, demands equally serious treatment for the very same reason.
6. Both aspects must be given pragmatic consideration to avoid costly omissions.

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