Hypothesis on the Failure of Spring Loaded Compressor Valve Plates

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ABSTRACT

Current ideas on the failure are discussed briefly and must be found insufficient. Impact of tilted plate is discussed in detail. This provides a basis for a new hypothesis on the failure of plates. The paper deals with stiff ring plates of Hoerbiger type valves but a lot of reasoning is also valid for flexible reed valves (as far as impact load is responsible for failure). Improvements in design, resulting from better understanding of the failure mechanism, are discussed.

PREVIOUS INVESTIGATIONS

Several authors have calculated and discussed stresses resulting from colinear impact on seat and guard, see e.g. [1, 2]. In general these calculated stresses are very low compared with the fatigue limits and are not able to explain failures.

Failure usually starts at a radial crack of the outer ring zone. The most disappointing feature in existing stress calculations is, that these do not predict any tangential stress which could be the cause of the cracks.

The papers mentioned[1, 2] are based on elementary theory of stress waves in rods. The author wants to point out that elementary theory of stress waves in rods and bars is based on some rather restrictive assumptions (see e.g. [3, 4]). The elementary theory does not take in account the lateral motion of a mass element. The lateral motion obviously is negligible for a long bar, because the kinetic energy absorbed by a mass element is mainly due to axial motion. A valve plate cannot be considered as a bar of a "length" equal to the thickness of the plate. Here in fact the lateral motion exceeds by far the axial motion, fig. 1.

If, e.g. a ring plate impacts colinearly, this can lead -apart from short duration axial stress pulses- to radial vibrations and hence to tangential stresses. This may also be connected with the fact, that coefficients of restitution are very small for impacts of flat plates.

\[
\sigma_t = \frac{E \cdot v_i}{c} \tag{1}
\]

\(E\) ...Young's modulus \(v_i\) ...impact velocity \(c\) ...sonic velocity \((c=\sqrt{E/\rho})\)

This is the same value as the axial stress in a long rod.

In addition, the elementary theory does not take into account shear stresses, acting in planes parallel to the impact plane. Such stresses are necessary, if appreciable lateral motion exist. These shearing stresses also occur in the impact plane (boundary conditions). So we can see that even the problem of colinear impact of a plate is a rather complex one.

A reasonable simplification for the impact problem may be; instead of dealing with an impact "steel on steel", one can use an infinitely hard impact plane. This corresponds to a mirror-symmetrical impact of 2 plates on each other at a relative velocity of 2\(v_i\). As the stress level is
proportional to the relative impact velocity, the stress level "steel on steel" is about half of the value "steel on hard plane".

A lot of interesting experimental studies (fatigue and metallographical) performed in Sweden, see e.g.[5,6,7,8,9,10], report that high stresses, restricted to a very small region, cause crack initiation. Dusil [9,10], localized crack initiation on surface, while Smith[8] found subsurface crack nucleation and very intense localized deformation. No positive evidence of the association of these intense bands of deformation with any element of microstructure was obtained. These papers mostly deal with flexible reed valves but fracture behaviour of ring valves is nearly identical[10].

Svenzon[5] found that oblique impact is essential for failure. With flexible reed valves obliquity of impact is caused by high frequency flexural and torsional vibrations of the specimen. Angles of impact vary at random at a most frequent angle of about 0.5 degrees[5].

The author discussed the originating of tilted motion for stiff ring plates elsewhere in these proceedings. For ring valves one usually finds 1 to 2 degrees maximum tilt angles (limited by maximum lift and outer diameter). Values of 1 degree are expected to occur very often due to

- instability of seat parallel motion
- unsymmetrical conditions of flow and spring forces
- oil stiction
- gas pressure waves coming from the piping and traversing the plate

From this arises that it is essential to consider the impact of the tilted plate in more detail.

Before doing so we discuss some aspects of the impact of a tilted strip with rectangular cross section and associated stress concentration effects.

DYNAMIC STRESS CONCENTRATION EFFECT

Consider a strip which impacts at a velocity \( v_i \) under a small tilt angle \( \gamma \) on a hard solid plane surface, fig.2. When the angle is very small, the contact front may advance at a velocity \( v_f \) which exceeds the sonic velocity \( c \):

\[
v_f = v_i / \tan \gamma \approx v_i / \gamma
\]

The part of the strip to the right of the contact front remains undisturbed. (This is similar to supersonic velocity of a sharp edged body in gas dynamics).

There is a special (critical) ratio of impact velocity \( v_i \) and tilt angle \( \gamma \) for which the contact front advances at sonic velocity:

\[
\frac{v_i}{\gamma}_{\text{crit}} = \frac{v_f}{c} = c
\]

The problem is complicated by the fact that we have to consider at least 3 different sonic velocities. A sonic velocity is the velocity of propagation of stress waves. For typical steel data we have:

- longitudinal waves (normal stresses) \( c_\sigma = \sqrt{E/\rho} = 5100 \text{ m/s} \)
- transversal waves (shear stresses) \( c_\tau = \sqrt{G/\rho} = 3200 \text{ m/s} \)
- Surface waves (Rayleigh waves) \( c_s = 0.92c_\sigma = 2960 \text{ m/s} \)

As surface waves are important in the reasoning in this paper and some readers may not be familiar with this special field, a short survey is given in appendix.

Let us return to eq(3). For \( c_s = 5100 \text{ m/s} \) and \( v_i = 5.1 \text{ m/s} \) e.g. we have

\[
\gamma_{\text{crit}} = 0.001
\]

If we consider critical conditions in connection with sonic velocity for transversal waves \( c_\tau \), we should note that in this case normal stress waves could reach also regions to the right of the contact front (\( c_\sigma > c_\tau \)).

In addition it might be noted that for a beam, the velocity of propagation of transversal waves \( c_\tau_{\text{beam}} \) is somewhat smaller than \( c_\tau \) in an unbounded body. The correction factor ranges between 0.9 and 1 and is often neglected.

For critical conditions, stress waves generated at successive contact front positions, move at the same speed as the contact front itself. Instantaneous energy input at contact front is always in phase with the oncoming stress waves from previous contacts. Hence we get a dynamic stress concentration effect.
IMPACT OF A TILTED RING PLATE

Let us now discuss the impact of a tilted ring plate, based on elementary concepts of elasticity and on adequate imagination. fig. 3.

Phase I: First contact period

The problem in this phase is of the three-dimensional elastodynamic type. The time for a stress wave to traverse the plate in its thickness \( h \) may be regarded as a typical measure of time in this phase. For a plate with 1mm thickness, this very short time becomes

\[ T_1 = \frac{h}{c_0} = 2 \times 10^{-7} \text{s} \]

In radial direction we always find "subsonic" conditions, hence no dynamic stress concentration effect is possible \((\alpha > \gamma_{\text{crit}})\). In tangential direction contact starts with a tilt angle \( \beta = 0 \) \((\approx \text{supersonic condition})\). When the contact front proceeds, \( \beta \) increases and passes the critical value \( \gamma_{\text{crit}} \). When this happens, a limited stress concentration effect is to be expected, as \( v_s \) is not constant \((v_s \text{ passes } c \text{ in a more of less short time interval})\). This stress concentration effect is assumed to be responsible for radial crack initiation.

For a ring plate with tilt angle \( \alpha \), \( \beta \) can be derived from simple geometrical considerations, fig. 4. \( \alpha, \beta, \gamma \) can be regarded as small \((\sin \alpha \approx \tan \alpha \approx \alpha \text{ etc.})\). The impact angle in tangential direction is

\[ \beta = \alpha \cdot \sin \varphi = \alpha \cdot \varphi; \beta_{\text{crit}} = \alpha \cdot \gamma_{\text{crit}} \]

FIGURE 4 Simplified geometric arrangement

FIGURE 3 Phases of oblique impact
The velocity of the contact front becomes

\[ v_f = \frac{v_i \alpha - v_i^2 t / (R \cdot \alpha^2)}{\sqrt{2v_i t - v_i^2 t^2 / R \cdot \alpha^2 + \frac{v_i^2 t^2}{R^2 \cdot \alpha^2}}} \]

Geometrical overlap in depth at \( \varphi = 0 \):

\[ v_i t \]

Let us calculate some values for a ring plate with \( R = 50 \text{mm} \) outer radius and \( \varphi = 1 \text{ degree tilt angle} \)

\[ v_i = 5.1 \text{m/s} (~ \beta_{\text{crit}} = 0.001, \text{ when normal stress waves are concerned}) \]

With these data we get:

\[ \beta_{\text{crit}} = \beta_{\text{crit}} / \alpha = 5.7 \text{mm} \]

\[ l = 2R \cdot \beta_{\text{crit}} = 5.7 \text{mm} \]

\[ t = \beta_{\text{crit}} \cdot 6 = 81 \text{N/mm}^2 \]

\( \varphi \) velocity of contact front:

see fig. 5

When the contact front passes one of the critical lengths "1", a stress concentration effect is to be expected at both ends of "1".

Principally the dynamic stress concentration effect is based on the fact, that the contact front initially advances at supersonic speed \( v_f > c \) in tangential direction. Slowing down, \( v_f \) passes the different sonic velocities. In the vicinity of the condition \( v_f = c \), stress concentration immediately in front or behind the contact front takes place.

As we have at least 3 relevant sonic velocities, namely \( c_s, c_r, c_p \), a lot of possible explanations of crack s'initiation exist. When considering the ring as a beam, there exists furthermore a velocity of propagation of flexural waves \( c_{fl} \). This will be discussed later.

To the author's opinion, the following explanation of crack initiator seems to be the most probable:

- Contact front velocity passes the value of propagation of Rayleigh waves (surface waves) \( v_f \sim c_s \).
- Small regions of tension resp. compression are formed in front of the contact front, fig. 6. As in this phase \( c_r, c_p \) are greater than \( v_f \), the corresponding waves can advance faster than the contact front and cause these stresses.
- Surface waves are generated in these regions and advance at \( c_s \) in direction of contact front.
- If \( v_f \sim c_s \), the energy input occurs in phase with these waves, stress amplitude (tension-compression in tangential direction) increase and cause cracks. Cracks could be created by a single stress concentration event or by repeated impacts causing plastic deformations.

![FIGURE 5 Geometric data for tilted impact](image-url)

![FIGURE 6 Stress concentration effect](image-url)
The increasing stress amplitude in the first sine wave immediately in front of the contact front will get the highest value—which may cause a crack or decrease before reaching a dangerous level (\(v_f\) passes \(c\) in a more or less short time intervals).

The reader may find this explanation of crack initiation somewhat "exotic", as the author does. However, it can explain a good deal of experience, especially:

- Cracks form in radial direction only
- Cracks form near outer ring edge only
- Cracks can form on both surfaces of the plate
- Cracks do not form on seat
- Large local deformations between the surfaces can be explained by dynamic stress concentration effect concerning \(c_T\).
- Stress concentration is highly local. Duration of stress pulses is roughly in agreement with the findings of Dusil (10^{-7}s), [10].
- If plates are used with diameters larger than the seat, surface waves penetrate also the outer ring zone and form oblique shock waves in gas dynamics. For critical conditions, head wave is approaching radial direction, which is the crack direction found in such plates.

**Phase II: Flexural waves leave contact area**

The stress waves preceding the contact front in the subsonic range transform to flexural waves. Shearing stresses decrease. The three-dimensional elastodynamic problem degenerates to a dynamic beam problem. According to dynamic beam theory flexural waves are dispersed. This means that pulses change their shape when propagated and one has to distinguish between group velocity \(c_{fl,gr}\) and phase velocity \(c_{fl,phase}\). The latter is relevant for an eventual stress concentration effect and depends on wave length \(\lambda\) [3,4]  

\[
c_{fl,phase} = \frac{c_T}{\sqrt{1 + \frac{\lambda^2}{4\pi^2 K^2}}} 
\]  

\(K\) is the concerned radius of gyration of the cross sectional area. For a cross section 1 x 8mm and a wave length \(\lambda = 20mm\) this formula gives \(c_{fl,phase} = 460m/s\). Shorter waves run faster. The group velocity is about twice the phase velocity, if \(K/\lambda\) is small. For our example we get \(c_{fl,gr} = 920m/s\), and a time \(T_{II} = 2.10^{-6}s\) for a wave to travel to a point opposite the impact point (180°) in a 100mm ring plate. This time may be compared with the duration of a complete opening process for a high speed compressor valve plate of about \(2.10^{-6}s\). Duration of closure is about five times longer.

An important parameter in phase II is the duration of impact \(T_i\). Stiff plates and great impact angles result in relatively smaller values \(T_i\). Geometrical conditions of impact are to complex for a realistic calculation of \(T_i\). In special cases the plate may rebound after a short time and impact on an opposite point (180°). Usually a motion like a hinged lid follows the impact of the tilted plate.

Now let us try to estimate stress level in phase II. Fig.7 shows typical velocity profiles before and after impact in a schematic manner without rebounding. A certain amount \(\varkappa\) of the total kinetic energy \(E_{kin}\) of the plate has to be dissipated:

\[
E_{diss} = \varkappa E_{kin} = \frac{1}{2}mv_i^2 \quad (6)
\]

Using moments of inertia, the quantity \(\varkappa\) can easily be estimated:

- \(\varkappa \approx 3/4\) for ring plates
- \(\varkappa \approx 5/8\) for full disc and multi-ring pl.

**FIGURE 7** Velocity profiles before and after impact

Each element of mass of the plate has to decrease its velocity to get the profile of fig.7b. This takes place in a dynamic process: stress waves coming from the impact point run around the ring. These waves are damped down and leave the plate oscillating with natural frequency (phase III).

While phase I is assumed to be responsible for crack initiation, the stress waves of phase II are assumed to be responsible for high general stress level (shearing and bending stresses), which may cause fracture.

As it is extremely difficult to calculate realistic stresses from the full elastodynamic equations, a rough energy method
is used to get some understanding of the process. The energy finally dissipated $E_{\text{diss}}$ must be roughly the energy of the stress wave (kinetic and strain energy) passing the plate ring segment in the vicinity of the impact point. The strain energy is a certain amount of this energy. If we consider the concerned ring segment as a simple clamped beam of length $L$ with a point load $P$ in the center, we get

$$E_{\text{strain}} = \frac{1}{2} \frac{P^2}{192EI} \sim E_{\text{diss}} = \frac{1}{2} \mu \nu^2$$  \hspace{1cm} (7)

Here: $I=bh^3/12$, $h$ plate thickness, $b$ width of outer ring. The maximum bending stress for this beam becomes

$$\sigma_{b, \text{max}} = \frac{Pb}{6I}$$  \hspace{1cm} (8)

Combination of eq (7) and (8) results in

$$\sigma_{b, \text{max}} \approx \sqrt{\frac{P}{3bh}} \frac{\mu \nu}{f_{\text{geom}}}$$  \hspace{1cm} (9)

The length $L$ is a somewhat arbitrary quantity. In the case of a multi-ring plate the length of an outer ring segment may be used for $L$. For a one-ring valve plate, the distance of the 2 stress waves at the right and the left of the impact point at the moment of impact termination may be used for $L$. This, however, is not easy to calculate, see e.g. [11, 12].

Using $m=\nu P$ (V=volume of valve plate) leads to

$$\sigma_{b, \text{max}} = \sqrt{\frac{P}{3bh}} \frac{\nu}{f_{\text{geom}}}$$  \hspace{1cm} (10)

$f_{\text{geom}}$ is a factor which takes in account the geometric arrangement of the plate ($V=A_h$, $A$...plate area)

$$f_{\text{geom}} = \frac{A_h}{bh} = \frac{A}{bh}$$  \hspace{1cm} (11)

Let us consider a tilted ring plate, diam. 100/84mm, 1mm thick, impacting at 5.1m/s. For a first estimation of stresses in phase I (9) we use a realistic value $L=50$mm. With eq (9) we find a value of about 1300 N/mm². Considering eq (7) and taking into account that strain energy will be about half the value of $E_{\text{diss}}$, we find a maximum bending stress of about 900N/mm².

**Phase III: Plate vibrating at natural frequency**

As already mentioned above, stress level in this phase is obviously lower. R.M. Weir [13] recently has published a hypothesis on the failure of valve plates based on resonances with natural frequencies. To the author’s opinion this hypothesis cannot explain the facts adequately.

Now the mechanism of failure can be summarized as follows:

- If by random impact velocity $v$, succeeds a certain limit and plate impacts at tilted position, a dynamic stress concentration effect can produce high local tangential stress. This forms a radial crack.
- Subsequent crack propagation is caused by general stress level resulting from flexural stress waves in the outer ring (phase II). But crack propagation may also be caused by the stress concentration effect. High stress pulses in a point of the plate occur at random and rarely.

Finally it might be interesting, to note that the stress concentration effect takes place on both sides of impact site. So the tear off of chips from the edges may have its origin from 2 cracks of a single impact event.

**CONCLUSIONS**

- Tilted motion of plate should be obviated whenever possible. To achieve this one should aim at:
  - strictly symmetrical flow conditions
  - symmetrical spring forces
  - high spring stiffness for stable seat parallel motion of plate
  - small maximum tilt angles
- If tilt of the plate cannot be avoided, consequences of severe local impact should be moderated.

An example for the latter is shown in fig. 8 (applied for a patent by the author [14]). If tilt cannot be avoided, it is advantageous to use prescribed tilting motion with invariable impact positions at seat and guard. This may be attained by specific unsymmetrical spring forces (i.e. spring 3 is weaker than spring 1 and spring 2). The impact position 4 can then adequately be designed; a circular segment 6 of the plate (or seat/or guard) can be ground so, that the plate impacts approximately parallel onto this segment area, thus the dynamic stress concentration effect avoiding. A semicircular area 7 around the theoretical impact point 4 is not weakened by perforations. In this design stress waves necessary for decreasing velocity of mass elements outside the impact area must not take exclusively the way via outer ring segment but can get straight (5) to the inner rings. This diminishes the general stress level in the plate. If critical impacts on seat and guard are to be expected, such a design can also be used for a
FIGURE 8  Improved valve design with prescribed impact position [12]

@ section I-I, ② valve plate, ④ valve plate near impact position 4, demonstrating the compact, unperforated semicircular region 7. ⑤ valve near impact position 4, demonstrating the circular segment 6, which is ground according to tilt angle α. This allows partially colinear impact of plate. Prescribed impact position 4 is provoked by unsymmetrical springs (spring 3 is weaker than spring 1 and spring 2).

5: Direct ways of stress waves to the interior regions of plate.
symmetrical position of the plate (if unsymmetrical springs are used, opening and closing impacts occur on opposite points, 180°).

The prescribed impact positions can be provided by further damping provisions, e.g. oil films, "soft seat" etc.

A valve usually is a compromise solution of three different aspects, fig.9. If the effects of impact can be moderated, a valve can e.g. endure higher lifts or higher compressor speeds or a smaller valve can be used.

It should be noted that the valve as proposed in fig.8 is just an idea, resulting from better understanding of plate failure. A lot of development has to be done before an industrial solution will be achieved.

Finally the author wants to note that the discussed dynamic stress concentration effect may be relevant for crack formation also in other fields of mechanical engineering.

REFERENCES
SURFACE WAVES (RAYLEIGH WAVES). Literature: see e.g. [4].

Some quantitative results for steel are given in fig. A.2 [4]:

- There is a plane parallel to the surface in a distance of about 0.25Λ, in which the stresses $\sigma_{xx}$ remain zero. Beyond this plane $\sigma_{xx}$ remains small ($< 0.18\sigma_{xx, surface}$).

- Thus the distance 0.25Λ is a rough measure for the penetration of the waves into the solid. A realistic wave length for the stress concentration effect in connection with valve plates is Λ/0.5mm. These waves penetrate only about 0.125mm. Thus it is meaningful to use the concept of surface waves even for plates of 1mm thickness.

- As a very thin layer is affected only by surface waves, the absolute amount of displacements $(u,w)$ is extremely small.

For a wave length Λ = 0.5mm the frequency $f$ and the duration for one oscillation $T$ becomes

$$f \cdot \Lambda = c_0$$

$$f = c_0 / \Lambda = 5.92 \cdot 10^6 \text{Hz}$$

$$T = \frac{1}{f} = 1.7 \cdot 10^{-7} \text{s}$$