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Impact Stresses in Flapper Valves - A Finite Element Analysis

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ABSTRACT

Impact stresses in flapper valves have been calculated on the basis of a finite element analysis. The impact velocity at initial contact used in the present calculations, 8 m/s, has been taken from previous high speed photographic measurements. Our calculations indicate that the final impact velocity at oblique impact exceeds 8 m/s significantly. This "whip-lash effect" gives rise to higher impact stresses than those obtained at colinear impact. It is concluded that impact fatigue cannot be explained without taking oblique impacts into account. When an obliquity of 1° is considered an impact stress of 730 N/mm² (106 ksi) is attained.

INTRODUCTION

In a previous investigation by Svenzon (1) attempts were made to estimate the stresses arising in flapper valves during impact fatigue testing. The impact velocity was determined from high speed photography (8 m/s) and when colinear impact was assumed a stress of 160 N/mm² (23 ksi) could be deduced. It was concluded in the same report that this stress is far too low to explain the observed impact fatigue failures in flapper valves. Svenzon also attempted to investigate the effect on stress of oblique impacts using the finite element method (FEM). However, due to time consuming numerical calculations, this investigation was not completed.

The situation has become different since efficient minicomputers are now easily available. This has enabled a reconsideration of oblique impacts and the purpose of this paper is to present results concerning this investigation.

It will be shown that oblique impacts give rise to "whip-lashes" as was suggested previously (1). The impact velocities and stresses verify that oblique impacts are much more harmful than colinear impacts. It will also be shown that oblique impacts can cause stresses sufficiently high to explain fatigue failures in flapper valves.

DESCRIPTION OF MODEL

In the theoretical model we have attempted to simulate the seat geometry in the Sandvik Impact Fatigue Tester (SIFT) as close as possible (see figure 1, ref. 2). However, in order to keep the number of elements down, the width of the valve has been reduced so that the radius of the valve tip is identical to the seat radius. Apart from this deviation the geometry is identical to the original geometry.

The model used is based on certain assumptions which are discussed below:

(i) The initial velocity was chosen to be 8 m/s which facilitates a comparison with results obtained by Svenzon (1).

(ii) Two initial valve configurations were considered: the first corresponding to the fundamental mode and the second corresponding to a superposition of the fundamental and the second mode of vibration. The first
situation which was analysed by Svenzon (1) corresponds to colinear impact whereas the second results in oblique impact with an angle of 1°. These two cases are schematically shown in figure 1.

(iii) In order to obtain a stable solution the time increment used in the numerical calculations needs to be smaller than ~ 50% of the transition time required for a stress wave to propagate across the thinnest part of the specimen. With a thickness of 0.38 mm the transition time becomes ~ 70 ns. Therefore 20 ns, which was used throughout all calculations has been judged to be sufficiently small.

The contact between the valve and the seat was governed by contact-impact relations which have been fully described by Nilsson (3). These relations automatically take possible release and multi impacts into account. When calculating the contact force a low pass digital filter (Butterworth filter of order 8, see ref. 4) with a corner frequency of 15 MHz was used to eliminate the spurious oscillations. Since the frequency of these is ~ 27 MHz it was considered that this would be a proper choice of the corner frequency of the filter. For instance, a lower corner frequency may have a tendency to underestimate the contact forces whereas a higher frequency allows the spurious oscillations to pass.

It should also be mentioned that the computations performed in the present analysis required 25 CPU-hours in a NORD-10/S computer.

RESULTS

The values obtained for the fundamental mode provide a basis for comparison with the second case which – in contrast to the fundamental mode – involves obliquity.

From the maximum contact force, which is 460 N, a maximum shear stress of ~305 N/mm² (44 ksi) may readily be calculated corresponding to a von Mises stress of ~530 N/mm² (77 ksi).

Figure 2 presents a sequence of pictures of the valve shape using the second mode at different stages of impact: 0, 10, 20, 50 and 60 µs after initial contact. It may be observed that the initial impact at node 1 causes a considerable distortion of the region in the immediate vicinity. Additionally, there is a clearly distinguishable distortion of the central part of the semi-circular tip (arrowed) causing it to assume a "bowl-shape".

Figures 3a and b show the displacement and corresponding velocity at node 5 as a function of time. In figure 3a it can be observed that the valve approaches the seat and comes very close at t = 25 µs but it does not touch until at t = 43 µs. Similar features were observed for other nodes close to node 5. The corresponding velocities do not exceed the initial velocity 8 m/s. The maximum contact force is 470 N corresponding to a shear stress of ~310 N/mm² (45 ksi) or a von Mises stress of ~540 N/mm² (78 ksi).

In figures 4a and b the same parameters as in the previous two figures are presented when node 9 is considered. This position corresponds to the very tip, where the velocity enhancement due to whip-lash effects may be expected to be pronounced. We find that contact occurs at = 30 µs at an impact velocity of 13 m/s, i.e. well above the initial velocity. The highest velocities were obtained for this node and it should be pointed out that at certain times the velocity is even higher than the final impact velocity, namely 18 m/s at t ~ 20 µs. This effect is attributed to the vibrations of the valve itself which are superimposed on the overall valve motion.

The maximum contact force for node 9 was found to be 630 N, which gives a shear stress of ~420 N/mm² (61 ksi). This is equivalent to a von Mises stress of ~730 N/mm² (106 ksi) which should be compared with the corresponding value obtained in case 1 which is 530 N/mm² (77 ksi).

DISCUSSION

The results clearly show that oblique impacts give rise to whip-lashes in certain parts of the valve tongue. This effect is convincingly shown in figure 4b, where an impact angle of 1° is considered. In this particular case the maximum velocity obtained is more than twice the initial velocity. Since impact angles often exceed 1° and may be as high as 3° (2) it is expected that "whip-lash effects" may become even more pronounced in real situations.

On the basis of results from nodes 1-9 we conclude that the "whip-lash effect", in terms of velocity and contact force is strongest at the very tip of the valve tongue. This result is consistent with the results obtained by Svenzon (1) who used an oblique seat which could be rotated. Svenzon found in his investigation that the position of the fracture could be controlled by altering the direction of obliquity, indicating that the whip-lash became more pronounced in the direction of obliquity.

From the "bowl-shape" of the valve in figure 2 we infer that large shear stresses
arise in this region. This is also in agreement with our calculations. In the present paper we have chosen to present our results in terms of the von Mises stress

\[ \frac{1}{2}[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2] + \]

\[ + 3(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)]^{1/2} \]

where \( \sigma_x, \sigma_y \) and \( \sigma_z \) are the normal stresses and \( \tau_{xy}, \tau_{yz} \) and \( \tau_{zx} \) are the shear stress. In our case \( \sigma_x, \sigma_y \) and \( \sigma_z \) are small enough to be omitted and we may write the von Mises stress

\[ \sigma \sim \sqrt{3} \tau \]

where \( \tau \) is the shear stress which arises in the immediate vicinity of the contact ring.

The maximum stress obtained was 730 N/mm\(^2\) (106 ksi). By way of comparison we may quote the fatigue limits obtained for AISI 1095 in reversed bending and in pulsating tension. These are 760 N/mm\(^2\) (109 ksi) and 600±600 N/mm\(^2\) (87±87 ksi) respectively (ref. 5). On the basis of this comparison it is evident that a stress of 730 N/mm\(^2\) (106 ksi) attained during oblique impact conditions is sufficient to cause fatigue in AISI 1095 valve steel. However, the stresses obtained when colinear impact is assumed is considerably lower (530 N/mm\(^2\), 77 ksi) and may consequently be expected to have little or no influence upon the fatigue process.

CONCLUSIONS

1) Oblique impacts give rise to "whip-lash effects", with a corresponding increase in contact force and final impact velocity.

2) The stresses obtained for oblique impacts in the direction of obliquity are sufficiently high to cause fatigue in valve materials.

REFERENCES


5. SANDVIK STEEL CATALOGUE No. 3, 41 and 3.43E.
A sequence of pictures representing the valve shape at $t = 0, 10, 20, 50$ and $60 \mu s$. The numbers 1-9 are nodes referred to in the text. The displacements are strongly exaggerated.
Figures 3a, b
Displacement and velocity at node 5. Note that there is a good agreement between the two diagrams, in particular the impact at $t = 43$ ms.
Figures 4a, b
Displacement and velocity at node 9. The onset of the whip lash occurs at ~15 ms.