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INFLUENCE OF SEAT POSITIONING AND SEAT DESIGN ON VALVE FATIGUE PERFORMANCE

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INTRODUCTION

The trend in compressor development is towards improved efficiency. Refrigerant, heat pump and air conditioning compressor manufacturers are primarily aiming for better energy efficiency paralleled with improvements in compression, volumetric and mechanical efficiency (1). The demand for more reliable function of the whole compressor system is steadily rising. Practical compressor tests indicate, that valve components can be considered as the most critically loaded compressor parts. Their durability is determined by the fatigue properties of the entire valve system and applied stresses.

The fatigue performance of compressor valves is influenced by a complex number of factors such as valve design, valve material and treatment (2-5), geometrical factors i.e. area of contact at stop or the area of the valve seat (6, 7) and operating conditions, primarily valve speed (i.e. setting velocity) and valve displacement (8). Fractographic analysis of damaged valves has shown that the positioning between valve reed and valve seat is another important parameter controlling the valve fatigue behaviour, fig 1 and 2.

In this paper an attempt is made to analyse the influence of the valve positioning above the seat on the valve fatigue strength during impact loading. Furthermore, various seat designs were investigated. The testing was performed in a specially constructed compressor valve simulator which gave the possibilities to analyse the above mentioned factors with a reasonable statistical significance. Laboratory compressor tests indicated it was difficult to separate the different factors determining the valve performance during dynamic loading.

MATERIALS

Two standard hardened and tempered valve steels were used, UHB 20C and UHB Stainless 716, strip thickness 0.38 mm (0.015 in). Strip materials from different heats were tested for each grade. The chemical composition is listed below.

\[ \text{Fig 1: Fracture at the point of shortest distance between impact contact area and valve edge.} \]

\[ \text{Fig 2: Damaged valve due to bad seat positioning.} \]
Table 1: Chemical composition.

<table>
<thead>
<tr>
<th>Grade UHB</th>
<th>wt %</th>
<th>C</th>
<th>Si</th>
<th>Mn</th>
<th>Cr</th>
<th>Mo</th>
</tr>
</thead>
<tbody>
<tr>
<td>20C A</td>
<td>1.03</td>
<td>.17</td>
<td>.39</td>
<td>.15</td>
<td></td>
<td></td>
</tr>
<tr>
<td>B</td>
<td>1.01</td>
<td>.25</td>
<td>.45</td>
<td>.14</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C</td>
<td>1.00</td>
<td>.20</td>
<td>.39</td>
<td>.16</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Stainless 716 A</td>
<td>.37</td>
<td>.41</td>
<td>.48</td>
<td>13.5</td>
<td>1.00</td>
<td></td>
</tr>
<tr>
<td>B</td>
<td>.38</td>
<td>.33</td>
<td>.45</td>
<td>13.3</td>
<td>1.02</td>
<td></td>
</tr>
</tbody>
</table>

Static tensile properties and hardness of test materials are given in table 2.

Table 2: Tensile properties and hardness.

<table>
<thead>
<tr>
<th>Grade UHB</th>
<th>Tensile strength</th>
<th>Yield strength</th>
<th>Elong. gauge 10 mm</th>
<th>HV 100 N</th>
</tr>
</thead>
<tbody>
<tr>
<td>20C A</td>
<td>1940 MN/m²</td>
<td>1740 MN/m²</td>
<td>3.5</td>
<td>560</td>
</tr>
<tr>
<td>B</td>
<td>1960 MN/m²</td>
<td>1850 MN/m²</td>
<td>3.5</td>
<td>565</td>
</tr>
<tr>
<td>C</td>
<td>1930 MN/m²</td>
<td>1750 MN/m²</td>
<td>4.0</td>
<td>560</td>
</tr>
<tr>
<td>SS 716 A</td>
<td>1920 MN/m²</td>
<td>1510 MN/m²</td>
<td>6.0</td>
<td>570</td>
</tr>
<tr>
<td>B</td>
<td>1860 MN/m²</td>
<td>1500 MN/m²</td>
<td>6.5</td>
<td>570</td>
</tr>
</tbody>
</table>

EXPERIMENTAL PROCEDURE

The fatigue testing of valve specimens was made in a specially constructed compressor valve simulator, fig 3. The specimens, dimension 100 x 20 x 0.38 mm, fig 4, were blanked from the strip parallel with the rolling direction. The edges were ground and polished. No surface preparation was made. Finished blades were operated by short-duration compressed air pulses, generated by a fluidic device. The reed was repeatedly lifted from and struck against the seat. The positioning between the valve seat and valve reed, i.e., valve overhang (Z) above the seat, fig 4, was checked with a micrometer. Some of the tested samples with various overhangs are illustrated in fig 5.

Fig 3: Compressor valve simulator for impact fatigue testing.

Fig 4: Valve reed vs valve seat, schematically.

Fig 5: Tested valve specimens with various overhangs.

The seat width/contact area, fig 6, was chosen as follows:

<table>
<thead>
<tr>
<th>Seat width mm</th>
<th>0.5</th>
<th>1.0</th>
<th>2.0</th>
</tr>
</thead>
<tbody>
<tr>
<td>Contact area mm²</td>
<td>27.5</td>
<td>53.4</td>
<td>100.5</td>
</tr>
</tbody>
</table>

Fig 6: Valve seats.
The operating frequency was 250 Hz, which is equivalent to the specimens natural frequency. When the reed hits the seat, the impact intensity was detected by a piezoelectric accelerometer. More details on testing procedure are given in ref (9). In agreement with published data, no units were specified in the present paper.

The impact fatigue tests were conducted at room temperature in a dry non-corrosive atmosphere. The fatigue limits at $10^7$ loading cycles were determined according to the stair-case method, (10). The fatigue limit is defined as the impact intensity at which 50% of the specimens failed within $10^7$ cycles. The experimental scope comprised 30 specimens.

RESULTS

The fatigue limits and corresponding standard deviations for the various positions between the valve specimen and seat are summarized in table 3. The testing data illustrating valve fatigue performance at different seat widths are also included.

Test results

Table 3: Impact fatigue limit at various seat positioning and seat widths.

<table>
<thead>
<tr>
<th>Grade UHB</th>
<th>Seat width</th>
<th>Impact fatigue limit</th>
<th>Standard dev</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>X mm</td>
<td>Z mm</td>
<td>a</td>
</tr>
<tr>
<td>20C, heat A</td>
<td>0.5</td>
<td>0.56</td>
<td>0.07</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>0.84</td>
<td>0.12</td>
</tr>
<tr>
<td></td>
<td>1.0</td>
<td>1.24</td>
<td>0.13</td>
</tr>
<tr>
<td></td>
<td>1.25</td>
<td>1.48</td>
<td>0.15</td>
</tr>
<tr>
<td></td>
<td>1.0</td>
<td>0.64</td>
<td>0.07</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>0.96</td>
<td>0.12</td>
</tr>
<tr>
<td></td>
<td>1.0</td>
<td>1.32</td>
<td>0.09</td>
</tr>
<tr>
<td>B</td>
<td>1.0</td>
<td>1.32</td>
<td>0.07</td>
</tr>
<tr>
<td>C</td>
<td>1.0</td>
<td>1.31</td>
<td>0.07</td>
</tr>
<tr>
<td>A</td>
<td>1.25</td>
<td>1.70</td>
<td>0.15</td>
</tr>
<tr>
<td></td>
<td>2.0</td>
<td>0.67</td>
<td>0.10</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>1.14</td>
<td>0.13</td>
</tr>
<tr>
<td></td>
<td>1.0</td>
<td>1.68</td>
<td>0.23</td>
</tr>
<tr>
<td>SS 716</td>
<td>1.0</td>
<td>0.81</td>
<td>0.20</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>1.13</td>
<td>0.13</td>
</tr>
<tr>
<td></td>
<td>1.0</td>
<td>1.48</td>
<td>0.30</td>
</tr>
<tr>
<td>B</td>
<td>1.0</td>
<td>1.41</td>
<td>0.10</td>
</tr>
<tr>
<td></td>
<td>1.25</td>
<td>1.82</td>
<td>0.24</td>
</tr>
</tbody>
</table>

The fatigue limits increased significantly with increasing distance between seat contact and valve edge tip, fig 5. It can be seen, that an increase in the valve overhang from 0 to 1 mm permits approximately doubled impact intensity.

Fig 7: Impact fatigue limit at various valve overhangs.

Generally UHB Stainless 716 exhibited a higher impact fatigue strength than UHB 20C. The differences between the tested grades are attributed to the differences in the fatigue crack initiation stage which is more retarded in stainless steel (9).

The varying contact area between valve specimen and valve seat had only a moderate effect on the impact fatigue limit when the overhang $Z$ was 0 mm. A more pronounced tendency was detected when the overhang $Z > 0$, fig 8.

Fig 8: Impact fatigue limit for various seat areas. UHB 20C.

IMPACT STRESSES

When the valve hits the seat or stop, compressive stresses are induced in the impact contact surfaces. The compressive stresses propagate as elastic waves through the valve material. They are transformed into tensile stresses, when they reach a free surface. Interference of tensile stress
waves reflected from different free surfaces creates the stress peaks which govern initiation and growth of impact fatigue cracks. Using Weibulls concept or critically stressed volume, the mechanical tensile stresses operating in the valve specimen were estimated to be about 1500 MN/m² for UHB 20C.

When stress waves propagate through the solid material, the stress amplitude decreases gradually. This stress wave damping implies that the highest tensile stress peaks will be situated close to the corners of edges and surfaces outside the valve contact area. This phenomenon is confirmed by extensive fractographic analysis of service failures from compressor ring valves and flexible valve reeds, (4, 9).

Following equations are valid for initiation and damping of elastic stress waves in solid material (11).

\[ \sigma_0 = \sigma_0 \sqrt{E \rho} \quad \text{eq 1.} \]

\[ \sigma = \sigma_0 e^{-t\sqrt{E \rho}/M} \quad \text{eq 2.} \]

where \( \sigma_0 \) and \( \sigma \) are initial and damped stresses respectively, \( \nu_0 \) setting velocity, \( t \) time, \( E \) modulus of elasticity, \( \rho \) density, \( M \) mass of the moving body, \( A \) seat area.

The velocity of the wave propagation \( C \) is constant and wave propagation distance \( l = C t \). When the wave front reaches the valve edge with a valve overhang \( Z \) then \( l = Z \).

\[ t = \frac{Z}{C} \quad \text{eq 3.} \]

Eq 2 and 3 give

\[ \sigma = \sigma_0 e^{-Z\sqrt{E \rho}/M} \quad \text{eq 4.} \]

The impact intensity \( a \) is proportional to the impact velocity \( \nu_0 \) (9) and thus to the initial stress according to eq 1.

\[ a = k \sigma_0 \quad \text{eq 5.} \]

where \( k = \text{constant} \)

if \( a(Z=0) = a_0 \) eq 4 and 5 give

\[ a(Z) = a_0 e^{-Z\sqrt{E \rho}/M} \quad \text{eq 6.} \]

Logarithmic transformation and simplification of eq 6 give

\[ \ln a(Z) = AZ + B \quad \text{eq 7.} \]

where \( A \) and \( B \) are constants.

The results shown in fig 7 are transformed according to eq 7 and illustrated in fig 9.

![Fig 9: Impact intensity \( \ln a \) versus valve overhang \( Z \).](image)

The correlation between experimental data and wave propagation theory is remarkably good. It is therefore appropriate to discuss the impact fatigue behaviour of the compressor valves on the basis of the relationships given above.

SEAT POSITIONING

Case studies of impact loaded compressor valves, fig 1, 2 and tests in a compressor valve simulator have shown clearly, that the positioning between the valve reed and the valve seat has a great influence on the resulting valve fatigue strength. This phenomenon can completely be attributed to stress wave damping. The stress controlling the fatigue cracking at the valve tip decrease as the distance between contact area and valve outer edge increases.

It is obvious that compressor valve operating conditions are difficult to accurately simulate in testing equipment. However, indicated results are in good agreement with practical experience.

During valve assembly, especially when rivet joints are used, the probability of inaccurate valve positioning above valve seat is fairly high. A high precision is required when ring valves are assembled.
SEAT CONTACT AREA

The stress at the instant of impact, time $t = 0$, overhang $Z = 0$, is given by eq 1. Besides the materials properties, modulus elasticity $E$ and density $\rho$, the only other influence on the stress level is speed at which the valve hits the seat. Theoretically the area of the contact surface does not influence the initial stress level. The observed tendency to lower impact fatigue limits at overhang $Z = 0$ for the seat having the smallest contact surface, 27.5 mm$^2$, is probably caused by non colinear impact. If the valve impact is not colinear, the variation of the seat area may influence the stress distribution at impact. This can affect the operating stresses without changing the average impact intensity (8).

When the overhang $Z > 0$, the damping of the stress wave has an effect on the stress level at the valve edge. Larger $A$ (seat area) increases the damping ratio. Consequently with larger seat areas an improved valve fatigue performance can be expected.

VALVE THICKNESS

As the stress wave damping increases when $M$ (the mass of moving body) decreases, a reduced valve thickness ought to improve impact fatigue limit for a given seat area. However, a thinner valve increases the biaxial bending stresses in the central part of the seat which restricts the possibility to decrease valve thickness. This point is of great importance particularly when mechanical loading of suction valves is considered.

MATERIAL PROPERTIES

The velocity of wave propagation $C$ is for the solid material equal to (11):

$$C = \sqrt{\frac{E}{\rho}}$$

eq 8.

Eq 1, 4 and 8 give

$$\sigma = \sigma_0 \sqrt{\frac{E}{\rho_a}} - Z \Delta A / M$$

eq 9

Theoretically a low modulus of elasticity has a favourable effect on impact loaded valves. However, compressor tests have shown that materials with lower modulus of elasticity than martensitic steels exhibited inferior properties. Unsurmountable difficulties were noted when compressor valve requirements for high cycle fatigue, wear resistance and leakage aspects were considered.

The influence of the material density is somewhat ambiguous. The initial stress level increases if the density increases but the operating stresses at the valve edge decrease due to increased stress wave damping. In practice the damping effect obviously has a dominating influence and the density should therefore be as high as possible.

It has been documented that the fatigue initiation stage corresponds to the total fatigue life for the high strength valve materials. The yield strength has a great influence on the crack initiation stage. At lower yield strength a more favourable plastic deformation takes place at stress concentrating points in valve components resulting in larger plastic zone. The blunting of the notch tip give a more retarded crack initiation due to reduced stress concentration and increased absorption of the elastic stress wave energy. This is in good agreement with the testing results. At the same tensile strength UHB Stainless 716 (yield strength $R_{p0.2} \approx 1500$ MN/m$^2$) exhibited higher impact fatigue limit than UHB 20C ($R_{p0.2} \approx 1800$ MN/m$^2$).

CONCLUSIONS

- The wave propagation approach was used to understand the origin of mechanical stresses and their damping after the valve impact.

- The positioning between the valve reed and valve seat i.e. valve overhang above the seat has a strong influence on the valve durability. It was found, that an increase in the overhang from 0 - 1.25 mm gave improved fatigue strength in the valve component.

- An increased seat width, i.e. contact area resulted in a higher fatigue performance of the valve reed.

- During valve assembly, especially when rivet joints are used, the probability of inaccurate valve positioning is fairly high. A high precision is therefore required when ring valves are assembled.

- The compressor valve operation is difficult to accurately simulate in testing equipment. However, documented results are in good agreement with practical experience. Laboratory compressor tests have indicated that it is difficult to separate the various factors determining the valve fatigue performance.

ACKNOWLEDGEMENT

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NOMENCLATURE

- $\sigma_0$ = initial stress (MN/m$^2$)
- $\sigma$ = damped stress (MN/m$^2$)
- $v_0$ = valve velocity at the instant of impact (m/sec)
- $t$ = time ($t = 0$ at the instant of impact) (sec).
- $E$ = modulus of elasticity (MN/m$^2$)
- $\rho$ = density (kg/m$^3$)
M = mass of moving body (kg)
A = seat area (mm²)
C = wave propagation velocity (m/sec)
l = wave propagation distance (mm)
Z = valve overhang (mm)
X = seat width (mm)
K, A, B = constants.

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