A Study on the Fatigue Failure of Valve System in Rotary Compressor

J. Y. Bae
GoldStar Company

J. W. Suk
GoldStar Company

Y. C. Ma
GoldStar Company

K. S. Im
GoldStar Company

Follow this and additional works at: https://docs.lib.purdue.edu/icec

Bae, J. Y.; Suk, J. W.; Ma, Y. C.; and Im, K. S., 'A Study on the Fatigue Failure of Valve System in Rotary Compressor' (1994). International Compressor Engineering Conference. Paper 1013. https://docs.lib.purdue.edu/icec/1013

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information. Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html
A STUDY ON THE FATIGUE FAILURE OF VALVE SYSTEM IN ROTARY COMPRESSOR

J. Y. Bae, J. W. Suk, Y. C. Ma, K. S. Im
LIVING SYSTEM LAB.
GOLDSTAR CO., LTD., SEOUL, KOREA

ABSTRACTS

For the reliability of the rotary compressor, it is essential to design the valve durably. For this purpose, the characteristics of the two major factors—bending and impact fracture of the valve—must be identified. To predict the valve life, we manufactured a testing apparatus and carried out computer simulation with respect to two factors. In bending fatigue test, the relation between the valve life and retainer radius was examined through the S-N curve and numerical analysis. And in impact fatigue test, the relation between the valve life and impact force was investigated with the same procedure as before. In this study, we acquired the dynamic behaviour of the valve, bending fatigue limit and impact fatigue limit. Also, we obtained the correlation factor between the impact force and the impact velocity by using the angular type load cell and gap sensor.

I. INTRODUCTION

These days, rotary compressor has merits in view of high efficiency, low noise and compact size. But, these merits must be lie on the basis of reliability. To satisfy that kind of reliability, we must take an interest in the valve fatigue fracture. It is well known fact that two major factors of valve fatigue fracture are bending and impact. Bending fracture is generated by alternating bending stresses during the valve lifting motion. Impact fracture is caused by alternating collisions between the valve and the valve seat. But conventional theoretical method is not enough to analyze the actual stresses and fatigue fracture limits of the valve. So, in this paper, we described the characteristics of impact and bending fatigue failure of valve by experiment and FEM. For this purpose, we manufactured a valve life tester and found the relation between bending (or impact) stress and life by varying retainer radius and valve impact velocity.


II. EXPERIMENTAL APPARATUS

To predict the bending and impact fatigue life of the rotary compressor valve, we manufactured a valve life testing apparatus as shown in Fig.1. By this apparatus, we simulated the valve behaviour of actual compressor operation. And this rig is operated by air supplier which is tuned with the constant speed of servo motor. Our apparatus has a shaft which consists of 4 air inlet holes and 4 air outlet holes whose phase difference is 22.5°. So we can use 4 times of regular
operating speed (240Hz). It takes about 70 minutes to reach $10^6$ fatigue stress cycles. All holes are located in the shaft rotated by the servo motor. We can vary impact velocities by varying the pressure differences between the up and down side of the valve. To measure impact velocities and impact forces, we set up a hollow quartz load washer and a proximity sensor around the valve and the retainer as shown in Fig. 2. And the pressure transducers are established to measure the pressure difference and some other pressures. There are two pressure regulation systems which are located at inlet and outlet pipes to regulate the valve impact velocities. With several retainers whose radius (R) are varied, it is possible to change the maximum bending stresses at will (Table 1). A sight glass is mounted to check the valve fracture during operation. Fig 3. shows the geometric shape of the valve specimen. The valve material is known as SANDVIC20C and its mechanical and geometrical characteristics are as follows; thickness = 0.254mm, tensile stress = 1930MPa and modulus of elasticity = $2.01 \times 10^{11}$. The port diameter is 9mm and the contact thickness is about 10 $\mu$m. So the total contact area is $2.8 \times 10^{-7}$ m$^2$. We compared two kinds of valve specimens, that is, one is made of the material against 0° direction of the roll (roll-valve), and the other is made of the material against 45° direction of the roll (45°-valve). The roll indicate the state of material winding.

III. BENDING FATIGUE FRACTURE

We could find the relation between the bending fatigue life and the bending stress by varying the retainer radius. Bending stress is a function of retainer radius and it can be represented as follows;

$$\sigma = \frac{Et}{2R} \quad \text{(1)}$$

$\sigma$: Bending stress
E: Modulus of elasticity
t: Valve thickness
R: Retainer radius

Table 1 shows the maximum bending stress with respect to the retainer radius (R) and height (h). We could find the relation between the retainer height (h) and the fatigue life by experiment as shown in Fig. 4. In case of bending fatigue failure, the fatigue failure limits are as follows; retainer height (h=4.3mm), retainer radius (R=31.9mm) and bending stress ($\sigma$ = 800MPa). According to the bending fatigue fracture experiment, the fatigue life of the 45°-valve is shorter than the roll-valve. Due to the rolling process, the 45°-valve has 45° directional initial scratch. And generally it is well known fact that the initial crack propagates toward 45° direction against tensile force. So, it seems that the crack of 45°-valve propagates more easily. Fig. 5-(a) shows the major trend of bending failure, that is, neck fracture. Through FEM, it is found that the weakness of the neck is observed at 2nd and 3rd natural modes as shown in Fig. 11. Fig. 7 shows the graph of stress propagation on the valve surface according to the time change. The stress propagates toward the neck of valve and causes neck fracture.

IV. IMPACT FATIGUE FRACTURE

To predict the relation between the impact force and the impact fatigue life, experiments were carried out with the change of impact velocities by using the valve life tester. The impact velocity was varied by changing the pressure difference between up and down side of valve with
fixed retainer radius \((R=71\text{mm})\) and fixed height \((h=1.8\text{ mm})\). The relation between the impact force and the impact velocity is dependant upon the contact characteristics, such as seat stiffness \((k)\), coefficient of restitution \((e)\), environment around the valve, etc. So, it is complicated to describe this relation precisely. Therefore for the simplicity, it could be assumed that the impact force is proportional to the impact velocity as the following equation.

\[
F = cv \quad \text{-----(2)}
\]

\(v\); impact velocity  
\(F\); impact force  
\(c\); constant

For the validity of the assumption of eq.(2) consider the case a mass\((m)\), whose velocity is \(v_1\), collides with the spring \((k)\) as shown in Fig.8. If the impact between the valve and the seat is perfectly elastic \((e=1)\) and unidirectional, the simple relation could be established as follows

\[
F \delta t = 2mv_1 \quad \text{-----(3)}
\]

\(m\); valve mass  
\(k\); valve stiffness  
\(v_1\); valve impact velocity  
\(\delta t\); impact duration

\[
dx/dt\big|_{t=0} = v_1, \quad x(\text{at } t=0) = 0 \quad \text{-----(4)}
\]

\(x\); valve displacement  
\(\text{initial condition}

therefore

\[
\delta t = \delta/wn \quad \text{----------(5)}, \text{ where } wn = (k/m)^{1/2}, \quad \delta ; \text{displacement}
\]

From eq.(4), we can drive eq.(5). Since \(\delta t\) is a function of only mass\((m)\) and stiffness\((k)\), \(\delta t\) is independent of \(v_1\). By rearranging eq.(5), eq.(3) we conclude that \(F\) is a linear equation of \(v_1\). Therefore, we could confirmed the assumption of eq.(2) represented in Fig.9. Fig.11 shows the behaviour of valve. The continuous line indicates in case that impact force is 66N and the dotted line is 44N. There are many peaks in the impact force signal. Each peak has equal time duration of \(1.4 \times 10^{-5}\) sec.. This fact agrees with the eq.(5). Fig.5-(b) shows the photograph of tip-fractured valve due to the impact collision. If two centers between valve and seat are misaligned, the fracture is found at the side of center line. Otherwise, the fracture is found at the tip of center line. Even though two centers are aligned well, some side-fractured valve could be found. These phenomena seem to be caused by geometrically asymmetric port shape. As a result, fatigue failure limits are obtained as follows; impact force is 61N and impact velocity is 5.3 m/s. These results are similar to many other preceding studies.

V. CONCLUSIONS

1. Bending and impact fatigue fracture limit are obtained against the design parameter of retainer radius and impact velocity, respectively. And the dynamic behaviour of valve acquired by experiments is compared with that by computer simulation.
2. The valve bending fatigue stress limit and the impact fatigue force limit are about 800 MPa and 61N, respectively. So, it is recommended that the valve radius be above 34mm
and the impact velocity be below 5.3 m/s in our valve.

3. It is possible to predict the maximum impact force by measuring only the impact velocity according to the linearization of valve impact force and impact velocity.

REFERENCES


[3] Dr. Werner Soedel, "Design and Mechanics of Compressor Valves".


Fig. 1 VALVE LIFE TESTER
FIG. 2 DETAIL OF VALVE ASSEMBLY

FIG. 4 S-N CURVE (BENDING FRACTURE)

FIG. 5 FRACTURED VALVE SPECIMEN

(a) BENDING FRACTURED SPECIMEN
(b) IMPACT FRACTURED SPECIMEN

FIG. 6 VALVE NATURAL MODES
(a) 1st mode (334Hz)  (b) 2nd mode (1560Hz)
(c) 3rd mode (2739Hz)  (d) 4th mode (3327Hz)

FIG. 7 STRESS WAVE PROPAGATION

TIME: 50 μsec.
TIME: 1.4 μsec.
TIME: 0.2 μsec.
**FIG. 8** SIMPLIFIED VALVE IMPACT MODEL.

**FIG. 9** IMPACT VELOCITY VS. IMPACT FORCE.

**FIG. 10** S-N CURVE (IMPACT FRACTURE).

**FIG. 11** TIME HISTORY OF
(a) IMPACT FORCE AT CENTER POINT
(b) VALVE BEHAVIOUR AT CENTER POINT
(c) VALVE IMPACT VELOCITY AT CENTER POINT

---

**Table 1. RETAINER RADIUS VS. BENDING STRESS**

<table>
<thead>
<tr>
<th>RETAINER RADIUS (mm)</th>
<th>RETAINER HEIGHT (mm)</th>
<th>BENDING STRESS (MPa)</th>
<th>FAILED (?) UNFAILED (X)</th>
</tr>
</thead>
<tbody>
<tr>
<td>17.4</td>
<td>10.5</td>
<td>1467</td>
<td>0</td>
</tr>
<tr>
<td>21.4</td>
<td>7.0</td>
<td>1193</td>
<td>0</td>
</tr>
<tr>
<td>28.1</td>
<td>5.0</td>
<td>908</td>
<td>0</td>
</tr>
<tr>
<td>30.1</td>
<td>4.6</td>
<td>848</td>
<td>0</td>
</tr>
<tr>
<td>31.8</td>
<td>4.3</td>
<td>800</td>
<td>X</td>
</tr>
<tr>
<td>34.0</td>
<td>4.0</td>
<td>751</td>
<td>X</td>
</tr>
<tr>
<td>36.4</td>
<td>3.7</td>
<td>701</td>
<td>X</td>
</tr>
<tr>
<td>42.0</td>
<td>3.2</td>
<td>608</td>
<td>X</td>
</tr>
<tr>
<td>48.0</td>
<td>2.8</td>
<td>532</td>
<td>X</td>
</tr>
<tr>
<td>55.0</td>
<td>2.4</td>
<td>464</td>
<td>X</td>
</tr>
<tr>
<td>71.0</td>
<td>1.8</td>
<td>350</td>
<td>X</td>
</tr>
</tbody>
</table>