LIFE TIME ESTIMATION OF A BUTTERFLY LAPPET VALVE

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Abstract. Butterfly valves are widely used in hydraulic installations transporting the water needed in petrol industry. The butterfly lappet is periodically loaded during opening and closing, the fluid pressure. This load generates normal bending stresses which are maximal on the lappet surface, around the rotation axle. Usually, the butterfly lappet fails due to crack initiation, leading to pressure-tight loss before braking. For this reason, estimation of the lappet life time is very important way to assure its functional safety. This paper gives the basic theoretical elements used in the proposed analysis and a case study is presented for a lappet having the diameter of 600 mm used in a hydraulic installation for water transportation at 20 atmospheres pressure (2MPa)

Keywords: crack propagation, fracture, butterfly lappet.

1. INTRODUCTION

Butterfly lappet valves are complex mechanical structures, and from that reason is difficult to determine theoretically the stresses and deformation that occur in their bodies. The shape and dimensions of the component parts can be selected such as to keep the stress and deformation amplitude in a secure level. However, taking in consideration the fact that the butterfly lappet is variable loaded during closing and opening, is possible that the functional safety to be affected by the phenomena related with this type of loads. The most important phenomena that affect the functional safety is crack initiation and its propagation leading to pressure-tight loss of the valve.

Usually, the crack affects the central area of the lappet as a surface crack in the rotational axle, where maximal bending stress occurs. During the opening and closing phase the pressure vary between zero and the maximum value, the lappet is loaded with a cyclic bending force having $R = 0.0$. Determination of the cycle number where the lappet pressure-tight is lost by crack initiation represent a efficient solution in estimation of valve life time.

2. CRACK INITIATION IS THE STABLE AREA

The crack initiate on the lappet exterior surface, around the rotational axle, where bending stress are maximal and grows proportional with the cycles number. The pressure-tight loss occurs before the braking point if the critical size of the crack is bigger that the lappet wall thickness. The state of the stabile crack initiation it can be computed with the Paris equation:

$$\frac{da}{dN} = c(\Delta K)^m \tag{1}$$

in which (c) and (m) are material coefficients.
In a coordinate system having $\log \Delta K$ on the abscise and $\log da/dN$ on the ordinate, equation (1), represents a straight line. The constant (c) is the abscise of the intersection point of $da/dN - \Delta K$ with he abscise axis

and the constant (m) is the declivity of $\log da/dN$ with respect of $\log \Delta K$ axis. Normally, (m) constant takes values between 3 and 4.

$\Delta K = K_{\max} - K_{\min}$ represent the variation of the stress intensity factor on the crack beginning, the computational equation depends of the crack type.

From equation (1) results:

$$N_{cr} = \int_{a_i}^{a_{cr}} \frac{da}{c(\Delta K)^m} \tag{2}$$

in which: a_i – the crack initial length

a_{cr} – the crack critical length, after which the propagation become instable.

Because ΔK is proportional with the crack length, direct integration in relation (2) can not be done, but it can be solved by numerical integration.

However, in some cases, when the crack length is very small with respect to the part length, is possible cu calculate the ΔK using the following equation:

$$\Delta K = 1{,}12 \cdot \Delta\sigma\sqrt{\pi \cdot a} \tag{3}$$

(3) in (1) results:

$$\frac{da}{dN} = c\left[1{,}12 \cdot \Delta\sigma\sqrt{\pi \cdot a}\right]^m \tag{4}$$

It is possible to calculate the cycle number when the length of the crack become critical using the following equation:

$$N_f = \int_{a_i}^{a_{cr}} \frac{da}{c\left[1{,}12 \cdot \Delta\sigma\sqrt{\pi \cdot a}\right]^m} = \frac{2}{(m-2)\cdot c\left[1{,}12 \cdot \Delta\sigma\sqrt{\pi \cdot a}\right]^m} \cdot \left[\frac{1}{a_i^{(m-2)/2}} - \frac{1}{a_{cr}^{(m-2)/2}}\right] \tag{5}$$

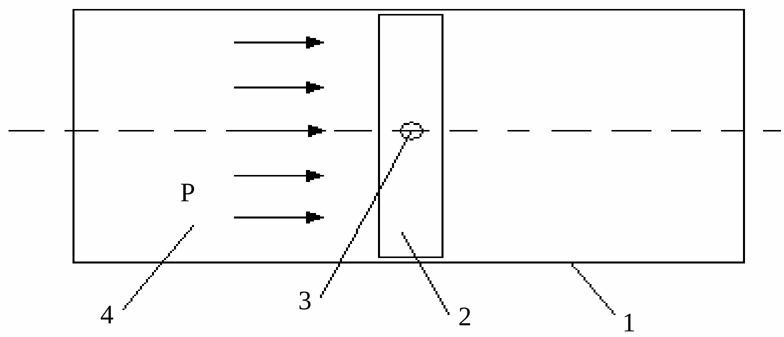
The critical crack length can be computed using the next equation:

$$a_{cr} = \frac{1}{\pi}\left[\frac{K_c}{1{,}12 \cdot \sigma_{\max}}\right] \tag{6}$$

in which K_c is the material fracture stickoitiveness.

3. LIFE TIME ESTIMATION OF THE BUTTERFLY LAPPET FOR DH600 VALVES

In industrial activities, its have been observed that a butterfly lappet valves are not used so often because of their pressure-tight loss caused by cracking phenomena. Especially, this phenomenon occur at very big valves, for pipes with diameters between 200 and 600 mm, for fluid transport at pressure between 10 and 20 atmosphere (1-2 MPa). In this experiment a valve having the lappet diameter 600 mm is used in an installation with 2MPa water pressure.



1 - pipe 600 diameter
2 - butterfly lappet
3 - rotation axle
4 - fluid

Fig. 1.

The lappet is made from alloy steel OTA17MoCr13 especially for valve, having the following chemical composition:

- 0,15-0,2 % carbon (C):
- 0,3-0,8 % silicon (Si)
- 0,5-0,8% manganese (Mn)
- 1,0-1,15 % chrome (Cr)
- 0,4-0,6% molybdenum (Mo)

The mechanical properties of the steel alloy are presented bellow:

- yield strength $\sigma_r = 907\, N/mm^2$
- tensile strength $\sigma_c = 761\, N/mm^2$
- $E = 2{,}02 \cdot 10^5\ N/mm^2$
- fracture stickoitiveness $K_c = 780 MPa \cdot \sqrt{mm}$

The lappet thickness in axle area is $t = 40mm$, so taking in consideration the pressure influence and making a simplified calculation is possible to compute the maximal normal stress.

$$\sigma = \sigma_{\max} = 168\, N/mm^2$$

The crack can be considerate having simplified shape, with initiation on the lappet surface, near the rotation axle (Fig. 2). We supposed that $a/c = 0{,}5$

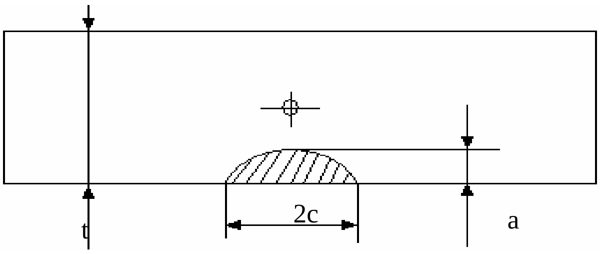


Fig. 2

In this case, the stress intensity factor has the following equation:

$$K = 1{,}12 \cdot M_k \cdot \sigma \sqrt{\frac{\pi \cdot a}{Q}} \tag{7}$$

in which $Q = \sqrt{\phi^2 - 0{,}212 \cdot \left(\frac{\sigma}{\sigma_c}\right)^2}$ and $\phi = \frac{3\pi}{8} + \left(\frac{\pi}{8}\right) \cdot \left(\frac{a}{c}\right)^2$

Taking in consideration the data obtained above, we obtain: $Q = 1{,}125$, $\phi = 1{,}276$

Considering $M_k \approx 1{,}25$, from equation (7), results:

$$Q_{cr} = \frac{K_c^2 \cdot Q}{1{,}254 \cdot \pi \cdot M_K^2 \cdot \sigma^2} \tag{8}$$

Using the data obtained above $a_{cr} = 3{,}942 mm$, smaller that the wall thickness. The conclusion is that the valve will suffer pressure-tight loss simultaneously with its fracture.
From equation (5) is possible to calculate the cycles number when the critical crack occur. So:

- shape parameter

$$M = (1{,}12 \cdot M_K)^2 \cdot \frac{\pi}{Q} \tag{9}$$

- the material coefficients $m = 3$ and $c = 3{,}565 \cdot 10^{-12} MPa^{-3} \cdot mm^{-1/2}$
- the initial crack length $a_i = 1mm$

is obtained $N = 44050$ cycles.

The obtained value allows life time estimation of butterfly lappet if is known its working frequency.

4. CONCLUSIONS

For the valves butterfly lappet loaded with variable forces, is necessary to study the crack initiation and its propagation in the stable area. Regarding this, the following aspects must be taken in consideration:

- pressure-tight can be lost even if the wall thickness is smaller that the critical length
- pressure-tight can be lost by lappet failure, if the critical length is smaller that the wall thickness.

The cycles number is calculated function to the above described situations and the life time can be estimated to assure a good functionality.

REFERENCES

1. Robert Blaudford, “Characterisation of Fatigue Crack Propagation”, hesis, Mississippi State University, May, 2001.
2. A., Miranda, L., Martha, “Propagation Curved Fatigues Crack under Variable Amplitude Loading”, 16th International Congres of Mechanical Engineering (COBEM) ABCM, Uberlandia, MG, 12: 257-266.
3. Royce Forman, Julie Henkenes, “An Evaluation of the Fatigue Crack Growth and Fracture Toughness Properties of Beryllium-Copper Alloy”, NASA, Technical Memorandum 102166, NASA, September 1990.
4. Oliviu Rusu, “Fatigues of Materials”, Ed. Tehnica, Bucharest, 1992.
5. D., Mocanu, “ Material testing”, Ed. Tehnica, Bucharest, 1982.