International Journal of Innovation and Applied Studies
ISSN 2028-9324 Vol. 32 No. 1 Feb. 2021, pp. 63-69

http://www.ijias.issr-journals.org/

An investigation on effect of mean stress on bending fatigue failures of compressor valve reeds

Kévin M. Tsapi Tchoupou[1], Bertin D. Soh Fotsing[2], and Desire Ndjanfang[3]

[1]Department of Petroleum Engineering, National Higher Polytechnic Institute, University of Bamenda, P.O. Box 39 Bambili, Cameroon

[2]Department of Mechanical Engineering, Bandjoun University Institute of Technology, University of Dschang, P.O. Box 134 Bandjoun, Cameroon

[3]Department of Mechanical and Industrial Engineering, National Higher Polytechnic Institute, University of Bamenda, P.O. Box 39 Bambili, Cameroon

ABSTRACT: Reed valves in a compressor are critical parts that have a high fatigue failure potential due to cyclic bending and impact caused by the cyclic nature of the compression process. A sudden failure of a valve renders the compressor useless. Although the refining process of methods of fatigue design has already taken more than 50 years, older criteria such as Gerber and Goodman models are still attractive for engineering design of high cycle fatigue components. This paper presents an investigation on the effect of nonzero mean stress on the design of valve reeds that are widely used in compressors. The investigation relates the choice of a mean stress compensation models, with the predicted fluctuating bending fatigue strength and estimated safety coefficient values. The calculations have been performed using Gerber, Goodman, Soderberg, ASME, Crossland, and Tsapi-Soh models. The most relevant goal of this paper is to verify the efficiency of classical and advanced stress based multiaxial fatigue criteria to estimated value of fluctuating bending fatigue strength. The criterion proposed by Tsapi-Soh was found to gives estimated value of the fluctuating bending fatigue strength very close to the typical value from technical data and satisfying results in predicting the survival of the reed valves under bending fatigue failure.

KEYWORDS: Mean stress effect, fatigue limit, fatigue criteria, safety coefficient, sandvik.

1 INTRODUCTION

Valve leaves of compressors that are supposed to work properly during compressor lifetime, experience severe bending and impact stresses during operation. Bending fracture is generated by alternating bending stresses during the valve lifting motion [1, 2]. Impact fracture is caused by alternating collisions between the valve and the valve seat (Fig. 1). Therefore, bending fatigue strength and impact fatigue strength of the reed valves play a deterministic role. According to industry studies, valve failures account for the unscheduled compressor shutdowns [3, 4, 5].

The bending stress in the flapper valve depends on the maximum deflection at the outside fiber of the strip and the modulus of elasticity. When the valve reaches to stopper, its displacement is maximum. The bending stress is calculated under this condition. Maximum bending stress is mainly distributed around valve root. For a given deflection, the material with higher modulus of elasticity can suffer from a higher bending stress.

Impact fracture is caused by alternating collisions between the valve and the valve seat. Every time a valve hits the stop when opening and the valve seat when closing, the valve material is affected by high compressive stresses induced at the impact area as well as in the seat. The formed stresses, located at the surface, are transformed into shear and tensile stresses and will be transported through the material as elastic waves. The maximum impact stress on the valve surface can be defined as [6]:

$$\sigma_0 = v_0\sqrt{\rho E} \quad (1)$$

As a consequence of damping properties, the stress levels will decay as the waves propagate through the material:

$$\sigma = \sigma_0 e^{-tA\sqrt{\rho E}/m} \quad (2)$$

Where v_0 is the impact velocity, ρ the materials density, t the time, m is the impacting, A the impact area, E is the modulus of elasticity; while σ_0 and σ are the initial and damped stresses, respectively.

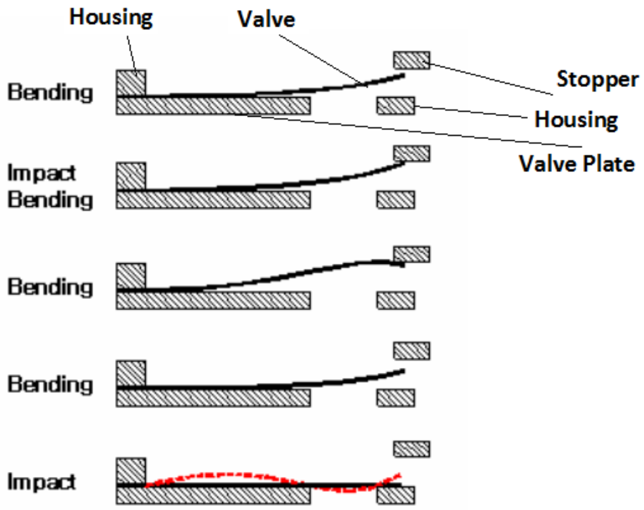

Fig. 1. Stress conditions in compressor valves during one load cycle [9]

During impact contact of a compressor valve, the crack initiation and propagation are mainly caused by induced elastic tensile stress waves and shear stress waves. However, these stress waves will decay during their travelling due to internal material damping. A typical fracture due to cyclic impact stress is when small fragments are torn off from the edges, which is usually termed as edge chipping. Valve materials for very high efficient compressors should have both high bending fatigue strength and high impact fatigue strength, and high stress damping capacity to reduce the crack propagation rate.

This impact is largely influenced by the torsional movement of the reed which makes the motion of a reed valve very complex and unmanageable in a rotary compressor. However, there is little information on the torsional movement of the reed valve in comparison to bending and impact caused by the cyclic nature of the compression process [7].

2 MATERIALS AND METHODS

Compressor valves steel must be capable of working continuously for long periods without failure. This places stringent requirements on materials, especially on bending and impact fatigue strength. Consequently, valve materials with higher fatigue strengths are required. The materials used in this study are compressor in Sandvik 20C [8], Sandvik 7C27Mo2 [9], and Sandvik Hiflex [10]. shows most the mechanical properties of the valve strip materials used in this study.

Table 1. Mechanical properties

Sandvik grade	Thickness (mm)	Tensile strength, S_{ut} (MPa)	Yield strengths, S_y (MPa)	Fatigue strength, f_{-1} (MPa)
20C	0.381	1850	1665	680
7C27Mo2	0.381	1800	1450	710
Hiflex	0.381	1900	1500	750

These strip materials have high tensile strength and high fatigue strengths under bending and impact stress conditions. Sandvik 20C is a carbon steel, meanwhile Sandvik 7C27Mo2 and Sandvik Hiflex are two martensitic, stainless chromium grades. Sandvik Hiflex is recommended particularly for use in carbon dioxide compressors for automotive air conditioning, where higher pressures and temperatures place increasing demands on the valve material.

2.1 MEAN STRESS COMPENSATION MODELS

The nonzero mean stress has a huge effect on the fatigue life and is often taken into account inthe transformation pro-cess of stress amplitudes. In this investigation, the choice of mean stress compensation models in the prediction of the fluctuating bending fatigue strength of the compressor valves steel and determination of the safety coefficient in fatigue of compressor valve are discussed. Various choices of models are available in the litterature; the study mainly focuses on the following models: Gerber [11], Goodman [12], Soderberg [13], ASME [14], Crossland [15] and Tsapi-Soh [16]. All the models considered, propose an equation to determine the alternating stress amplitude σ_a at the limit, for a given σ_m with $(\kappa = f_{-1}/t_{-1})$.

Gerber

$$\sigma_a = f_{-1}\left(1 - \left(\frac{\sigma_m}{S_{ut}}\right)^2\right) \tag{3}$$

Goodman

$$\sigma_a = f_{-1}\left(1 - \frac{\sigma_m}{S_{ut}}\right) \tag{4}$$

Soderberg

$$\sigma_a = f_{-1}\left(1 - \frac{\sigma_m}{S_y}\right) \tag{5}$$

ASME

$$\sigma_a = f_{-1}\sqrt{1 - \left(\frac{\sigma_m}{S_y}\right)^2} \tag{6}$$

Crossland

$$\sigma_a = f_{-1}\left(1 - \frac{3-\sqrt{3}\kappa}{3}\frac{\sigma_m}{f_{-1}}\right) \tag{7}$$

Tsapi-Soh

$$\sigma_a = \frac{\sqrt{4\sigma_m^2\left(\frac{\kappa^4}{9} - \frac{\kappa^2}{3}\right) + 4f_{-1}} - \frac{3-\kappa^2}{3}(2\sigma_m)}{2} \tag{8}$$

2.2 BENDING FATIGUE STRENGTHS AND SAFETY FACTOR

The maximum stress applied to a compressor valve can be easily calculated thanks to commercial finite element analysis software. The stress state analysis in fatigue consists in identifying a mean and alternating stress amplitude. Due to the valve plate, the assumption is often done that the minimum stress is zero. Then, mean and alternating stress can be determined from the maximum stress: $\sigma_a = \sigma_m = \sigma_{max}/2$.

The tensile strength S_{ut}, impact fatigue strength and bending fatigue strength (f_{-1}, for $\sigma_m = 0$) at a failure rate of 5% are obtained from tests [8, 9, 10]. The fatigue limit under fluctuating bending stress, with the minimum stress equals zero (f_0) is calculated from reversed bending values using the mean stress compensation models as in

The intersection between the previous equations (Eqs. 3, 4, 5, 6, 7, 8) and the equation $\sigma_m = \sigma_a = f_0/2$, gives the calculated value for f_0 [17]. The choice of a model is very important because it defines the admissible stress, which is used in the safety coefficient calculation. Conservative designers often use the modified Goodman criterion, while the Gerber model is used in optimistic design.

The safety coefficient n is defined as the ratio between the admissible stress and the alternating stress σ_a applied. Its value depends on the model used. In order to achieve the optimal geometry of the reed valve for the safety factor, the fatigue strength should be analyzed and determined in the design process. The various expressions of safety coefficients are listed below, based on the fact that both stresses σ_a and σ_m, vary arbitrarily; n is the factor of safety; $k_f = 1.32$ is the fatigue stress-concentration factor [18].

Table 2. Fatigue models and predicted bending fatigue strengths and safety factor $(\boldsymbol{\kappa} = \boldsymbol{f_{-1}/t_{-1}})$

Models	**Fluctuating bending fatigue strength**	**Safety coefficients**
Gerber	$f_0 = \sqrt{\frac{1}{4} + \left(\frac{f_{-1}}{S_{ut}}\right)^2} - \frac{1}{2} \Big/ \frac{f_{-1}}{2S_{ut}^2}$	$n_{GE} = \frac{1}{2}\left(\frac{S_{ut}}{k_f\sigma_m}\right)^2 \frac{k_f\sigma_a}{f_{-1}}\left(\sqrt{1 + \left(\frac{2\sigma_m f_{-1}}{S_{ut}\sigma_a}\right)^2} - 1\right)$
Goodman	$f_0 = 2\, f_{-1}\Big/\left(1 + \frac{f_{-1}}{S_{ut}}\right)$	$n_{GO} = \frac{f_{-1}}{k_f\sigma_a + f_{-1}\frac{k_f\sigma_m}{S_{ut}}}$
Soderberg	$f_0 = 2f_{-1}\Big/\left(1 + \frac{f_{-1}}{S_y}\right)$	$n_{SO} = \frac{f_{-1}}{k_f\sigma_a + f_{-1}\frac{k_f\sigma_m}{S_y}}$
ASME	$f_0 = 2f_{-1}\Big/\sqrt{1 + \frac{f_{-1}^2}{S_y^2}}$	$n_{AS} = \frac{f_{-1}}{\sqrt{\left(k_f\sigma_a\right)^2 + \left(f_{-1}\frac{k_f\sigma_m}{S_y}\right)^2}}$
Crossland	$f_0 = \frac{2f_{-1}}{\frac{1}{3}\frac{f_{-1}}{t_{-1}}\left(1 - \frac{2}{\sqrt{3}}\right) + 2}$	$n_{CR} = \frac{f_{-1}}{k_f\sigma_a + \left(1 - \frac{1}{\sqrt{3}}\frac{f_{-1}}{t_{-1}}\right)k_f\sigma_m}$
Tsapi-Soh	$f_0 = \frac{2f_{-1}}{\sqrt{4 - \left(\frac{f_{-1}}{t_{-1}}\right)^2}}$	$n_{TS} = \frac{f_{-1}}{\sqrt{\left(k_f\sigma_a\right)^2 + \left(2 - \frac{2}{3}\frac{f_{-1}^2}{t_{-1}^2}\right)k_f^2\sigma_a\sigma_m + \left(1 - \frac{1}{3}\frac{f_{-1}^2}{t_{-1}^2}\right)\left(k_f\sigma_m\right)^2}}$

The material information that is being used in the calculation process of predicting a safety coefficient has been obtained from literature [19] life. shows the maximum bending stress with respect to the retainer radius. The valve is known as and its mechanical and geometrical characteristics are as follows [19]; thickness= 0.254mm, tensile stress= 1930MPa and modulus of elasticity = 201GPa.

Table 3. Retainer radius and bending stress

Valve radius $(\boldsymbol{mm})$	**17.4**	**21.4**	**28.1**	**30.1**	**31.8**	**34.0**	**36.4**	**42.0**	**48.0**	**55.0**	**71.0**
Retainer height (mm)	10.5	7.0	5.0	4.6	4.3	4.0	3.7	3.2	2.8	2.4	1.8
Bending stress (MPa)	1467	1193	908	848	800	751	701	608	532	464	360

3 RESULTS AND DISCUSSION

Fluctuating bending fatigue strength values calculated from reversed bending values are presented in Using fatigue limit stress values under reversed bending, we analyze how each model predicts the fluctuating bending fatigue strength for Sandvik valve steels (Fig. 1).

The results of the comparison of this steel grades presented in Fig. 1 shows the importance of the selection of stress compensation models on fluctuating bending fatigue strength value of the reed valves and the impact fatigue test parameters such as valve lift and impact velocity.

The results in indicate that the Crossland model is too conservative compared to Goodman model. The predicted fluctuating bending fatigue strengths using Crossland model are not realistic. However, the predictions from the Tsapi-Soh model are close to that recommended by the American Society of Mechanical Engineers (ASME), and the optimistic Gerber model.

Table 4. Calculated value for f_0

	Crossland f_0 (MPa)	Soderberg f_0 (MPa)	Goodman f_0 (MPa)	Gerber f_0 (MPa)	ASME f_0 (MPa)	Tsapi-Soh f_0 (MPa)
20C	712±712	966±966	994±994	1214±1214	1250±1250	1291±1291
7C27Mo2	743±743	953±953	1018±1018	1249±1249	1275 ±1275	1348±1348
Hiflex	785±785	1000 ±1000	1075±1075	1319±1319	1342±1342	1424±1424

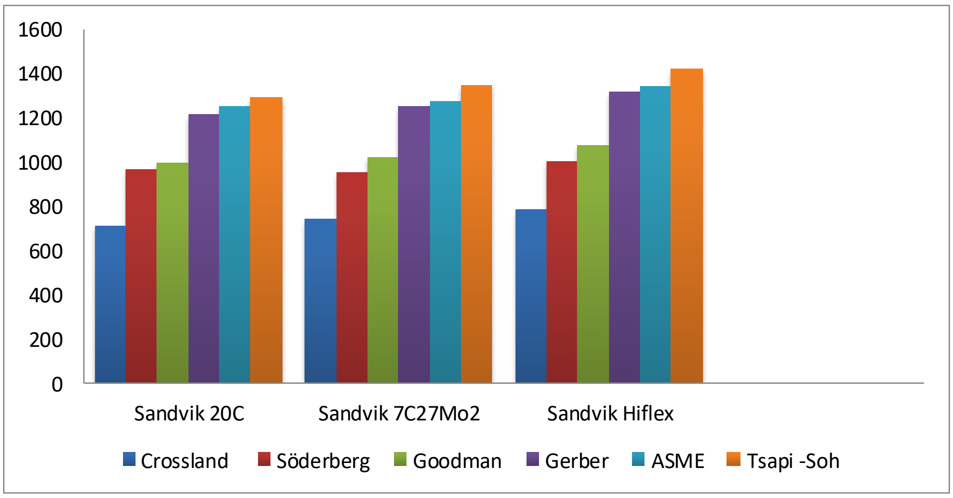

Fig. 2. Predicted fluctuating bending fatigue strength for various models and materials

From results presented in Table 5, Goodman and Soderberg models were not efficient in predicting the survival of the reed valve as for the fifth and sixth loading case as observed experimentally. Gerber and ASME elliptic were not efficient in predicting the survival of the reed valve as for the fourth loading case, however. The predictions of the Tsapi-Soh model are very close to the predictions obtained from Gerber and ASME models. Crossland model yielded similar results. Safety coefficient above 1.1 is required to ensure a conservative level of reliability, which should lead to infinite life. Safety coefficient between 1 and 1.1 imply marginal design for which the confidence level is not adequate to infinite life requirement.

Table 5. Predicted Safety coefficient for various models

Maximum Bending stress (MPa)		**1467**	**1193**	**908**	**848**	**800**	**751**	**701**	**608**	**532**	**464**	**360**
Failed (√) Unfailed (×)		√	√	√	√	×	×	×	×	×	×	×
Safety coefficient	Gerber	0.60	0.74	0.97	1.0	1.1	1.2	1.3	1.4	1.7	1.9	2.4
	Goodman	0.49	0.60	0.79	0.85	0.90	0.96	1.1	1.2	1.4	1.5	2.0
	Soderberg	0.48	0.59	0.77	0.83	0.88	0.93	1.0	1.2	1.3	1.5	1.9
	ASME	0.62	0.77	1.0	1.1	1.1	1.2	1.3	1.5	1.7	2.0	2.5
	Crossland	0.66	0.81	1.1	1.1	1.2	1.3	1.4	1.6	1.8	2.1	2.7
	Tsapi-Soh	0.64	0.78	1.0	1.1	1.17	1.25	1.3	1.5	1.8	2.0	2.6

4 Conclusion

This paper reported on methods for estimating the fatigue limit under fluctuating bending for reed type valves used in compressors. It was shown that the Tsapi-Soh fatigue criterion was efficient in predicting the fluctuating bending fatigue strength with results close to that used in data sheets. Moreover, the fatigue criterion was efficient in predicting the failure or survival of the reed valve under cyclic bending stress based on calculated safety coefficient.

No improvement has been observed with the Crossland and Soderberg that rendered too conservative results compared to Goodman model. The application of the proposed procedure can result in important gains in terms of time for designing high performance valves for new compressor designs or operating conditions. With the new materials, designers confronted with fatigue problems will just have to select the material with the best inherent fatigue resistant properties and safety factor. Development of new compressor reed material with higher bending fatigue limits and high impact fatigue limits are under progress.

The method presented was a first step to help designer to deal with stress results from calculation. The analysis of the stress state applied doesn't take into account the cycling effect of the load, and the resulting biaxial stress states history experienced by the valve critical mode [20].

Acknowledgment

The University of Bamenda is strongly acknowledged for the financial support for this research. The support of Mrs. Magnou Belinda, Mr. Ekokem Asiel, Mr. Tchoupou Salathiel, Mr. Esselem Phanuel in Christ and Nguyon Zephaniah are gratefully acknowledged.

References

[1] Soedel W., Design and mechanics of compressor valves, Ray W. Herrick laboratories, School of Mechanical Engineering, Purdue University, 110 - 112, 1984.

[2] S. Dhar, “Repeated impact on fatigue fracture of carbon spring En-42J steel, ” Theoretical and applied fracture mechanics, vol. 10, no. 2, pp. 165–171, 1988.

[3] A. Burgstaller, D. Nagy, R. Almbauer, W. Lang, “Influence of the main parameters of the suction valve radius on the overall performance of a small hermetic reciprocating compressor, International Compressor Engineering Conference, Paper 1934, 2008.

[4] G. Hatch and D. Woollatt, “The dynamics of reciprocating compressor valve springs, ” International Compressor Engineering Conference, Paper 1555, 2002.

[5] Woo S, Salilew WM, Teklay Y, et al. “Reliability design of mechanical systems such as compressor subject to repetitive stresses, ” MOJ Civil Engineering, vol. 6, no. 1, pp. 12–15. 2020.

[6] P. Malin, M. Martin, C. Guocai, "Modulus of Elasticity and Its Influence on the Performance of Flapper Valve Materials, "International Compressor Engineering Conference, Paper 1937, 2008.

[7] Y. Xiaoling, R. Yumei, T. Qin, L. Zhao, J. Xiaofei, W. Xiaolin “Study on the torsional movement of a reed valve in a rotary compressor, ” Advances in Mechanical Engineering, vol. 10, no. 6, pp. 1–10, 2018.

[8] Sandvik Material brochure, Sandvik 20C Compressor valve steel, 2020.
[Online] https://www.materials.sandvik/en/materials-center/material-datasheets/strip-steel/sandvik-20c-flapper-valve-steel/ (December 12, 2020).

[9] Sandvik Material brochure, sandvik 7C27MO2 Compressor valve steel, 2020.
[Online] https://www.materials.sandvik/en/materials-center/material-datasheets/strip-steel/sandvik-7c27mo2-flapper-valve-steel/ (December 12, 2020).

[10] Sandvik Material brochure, Sandvik Hiflex Compressor valve steel, 2020.
[Online] https://www.materials.sandvik/en/products/strip-steel/strip-products/compressor-valve-steel/sandvik-hiflex/ (December 12, 2020).

[11] Gerber, W.Z. Bestimmung der zulässigen Spannungen in Eisen-Konstruktionen (Calculation of theallowable stresses in iron structures). Z. Bayer Archit. Ing. Ver, vol. 6, pp. 101–110, 1874.

[12] Goodman, J. Mechanics Applied to Engineering; Longmans, Green & Company: Harlow, UK, 1899.

[13] C. Soderberg, Factor of safety and working stress, Transactions of ASME, vol. 52, pp. 13–28, 1939.

[14] Budynas, R.B., and Nisbett, J. K. Shigley’s Mechanical engineering design, 9 Th Ed. McGraw-Hill, New York, 2011.

[15] B. Crossland, "Effect of large hydrostatic pressure on the torsional fatigue strength of an alloy steel, " In Proceeding of the International Conference on Fatigue of Metals, Institution of Mechanical Engineers, pp. 138-149, 1956.
[16] K. M. Tsapi Tchoupou and B. D. Soh Fotsing, “A new high cycle fatigue equivalent stress applied to out-of-phase biaxial stress state, ” International Journal of Innovation and Applied Studies, Vol. 12 No. 3, pp. 676-684, 2015.
[17] P. Grolier, “A method to determine the safety coefficient in fatigue of compressor valves, ” International Compressor Engineering Conference, Paper 1418, 2000.
[18] J. R. Leng, “Finite element analysis of dynamic flapper valve stresses, ” Proceeding of the International Compressor Engineering Conference, West Lafayette, IN, USA, 369-375, 2000.
[19] J. Y. Bae, J. W. Suk, Y. C. Ma, K. S. Im, “A study on the fatigue failure of valve system in rotary compressor, ” International Compressor Engineering Conference, Paper 1013, 1994.
[20] M. Hareland, A. Hoel, S. Jonsson, D. Liang, G.Chai, “Selection of Flapper Valve Steel for High Efficient Compressor, ” International Compressor Engineering Conference, Paper 2330, 2014.