A Method to Determine the Safety Coefficient in Fatigue of Compressor Valves

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ABSTRACT

This paper proposes a method of evaluating the fatigue life of a refrigeration compressor reed valve, given the applied stresses as might be computed by a valve simulation. Mean and alternating values of equivalent stresses (e.g. Von Mises) are plotted in the Haigh diagram. Various choices of models are considered for the safe area of the diagram. This method takes into account surface finish, thickness, ultimate tensile strength and residual stresses related to the strip valve material and process used to obtain the parts. For a given level of confidence, it is possible to predict a safety coefficient, the ratio between admissible stress and applied stress.

NOMENCLATURE

$\sigma VonMises$	:	equivalent Von Mises stress
σ min	:	minimum stress amplitude
σ Max	:	maximum stress amplitude
σm	:	mean stress amplitude
σa	:	alternating stress amplitude
R	:	stress ratio
N	:	number of cycle to failure
$\sigma_{-1}(N)$	:	fatigue limit reversed bending stress at N cycles
$\sigma_0\ (N)$	:	fatigue limit fluctuating bending stress, minimum stress=0 at N cycles
σ_D	:	bending fatigue strength at $N = 2\ 10^6$, 50 % probability of fracture
Rm	:	tensile strength
σe	:	elastic limit
σR	:	residual stress
t	:	strip valve thickness
Ra	:	surface roughness
σadm	:	admissible stress
S	:	safety coefficient factor

INTRODUCTION

Compressor valves are critical parts in a compressor. The failure of a valve renders the compressor useless. The stress level applied to a compressor valve can be easily calculated. In the past, calculation was done by hand using formula for beams. Now, thanks to finite element analysis software, precise calculation can be done using shell elements without any problem. The problem remaining is that very often the designer doesn't know how to deal with the stress levels obtained. Thanks to field experience some " rule of thumb" value often circulates to help the designer to make decision. In fact, the maximum admissible stress depends on various properties of the material and the processes used on it. The following paper intends to present a comprehensive method to fix the acceptable limit, by the calculation of a safety coefficient.

CONTEXT

The method, presented below, consists in comparing the cyclic stress state to the mechanical properties of the material by calculating a safety coefficient, ratio between the admissible stress and the alternating applied stress. Depending on the safety coefficient value, it is possible to conclude on the safety level of the valve design. The main steps of the method are the analysis of the stress state, the determination of the mechanical properties, and the definition of an admissible endurance area in the Haigh diagram. A general example of the safety coefficient calculation is described in the reference [1]. The reference [2] presents a definition of an admissible endurance area in the Haigh diagram, based on the original formulation coming from the reference [3]. The mechanical properties of the strip steel for compressor valves depends on many factors. Various technical information were found in the references [4], [5], and [6]. An analytical approach of the endurance limit, taking into account the residual stress is proposed in the reference [7]. Practical data and technical information on the residual stresses due to the process used to obtain compressor valve are presented in reference [8].

METHOD

Stress state analysis

The cyclic stress state history of the valve, loaded under various conditions, during his motion, need to be determined at each cross section area of the valve. The analysis is generally led in the critical cross section area where the stress amplitude is maximum. The analysis is based on the normal stress versus the cross section area which is often the most significant among the stresses matrix of compressor valve, under standard load and boundary conditions. Traditionally, under bending load, the critical stress for both suction and discharge leaf, is close to the anchored area. In this case, the stress state often considered is the stress normal to the cross section area. In other case, special attention need to be paid to the stress state, specially if the critical cross section area is close to the port area of the valve plate, where shear stress can exist due to the extrusion stress of valve through the port. The reduction of a multiaxial stress state history to an uniaxial equivalent stress combining both normal and shear stress, even if certainly unperfected fatigue point of view, is often used to simplify the approach. The Von Mises equivalent stress is often chosen ($\sigma VonMises$).

The stress state analysis in fatigue consists in identifying a mean and an alternating stress amplitude to be plotted in the Haigh diagram, presenting the mean stress on an horizontal scale and the alternating stress on a vertical scale. Mean and alternating stress amplitude are determined from the maximum and minimum stresses. The maximum stress corresponds to the largest algebraic value of the applied stress ($\sigma\ Max$). The minimum stress corresponds to the least algebraic value of the applied stress (σ min). The mean stress amplitude is equal to the algebraic mean value between the maximum and minimum stress : $\sigma m = (\sigma Max + \sigma \min)/2$. The alternating stress amplitude is equal to the algebraic difference between the maximum and minimum stress value, divided by two : $\sigma a = (\sigma Max - \sigma \min)/2$. We use the Haigh diagram presenting the stress state on a plan with σm on the horizontal scale and σa on the vertical scale. Due to the valve plate, the assumption is often done that $\sigma \min = 0$. Then, mean and alternating stress can be determined from the maximum stress : $\sigma m = \sigma a = \sigma\ Max / 2$.

Mechanical properties

The tensile strength Rm and the fatigue strength σ_D required to do the fatigue analysis on the stress state plotted in the Haigh diagram, are generally obtained from test. The tensile strength is the stress calculated for the highest load withstood by a material sample in a tensile test, after passing the elastic limit.

The fatigue limit is defined as the limit reached by the alternating stress amplitude, for a given mean stress amplitude, under a certain number of cycles, with a certain probability of fracture : $\sigma_R(N) = \sigma m \pm \sigma a$, where R is the stress ratio, algebraic ratio between the minimum stress and the maximum stress, in a stress cycle : $R = \sigma \min / \sigma\ Max$.

Two particular fatigue limits are commonly used for flapper valve.
The fatigue limit under reversed bending stress at N cycles ($\sigma_{-1}(N)$, for R =-1 or $\sigma m = 0$).
The fatigue limit under fluctuating bending stress, with the minimum stress equals zero at N cycles ($\sigma_0\ (N)$, for $R = 0$ or σmin=0).

The fatigue strength is the stress under which unlimited life is obtained in a simple model of the Wöhler curve. To pass from a fatigue limit to the fatigue strength it is necessary to fixe the number of cycle to failure required, and the percentage of admissible failure. Main of the technical data available for strip steel for compressor valve, are related to the fatigue limit under reversed bending stress, σ_{-1} for $N = 2\,10^6$ at 50 %. Then the fatigue strength is often noticed σ_D, the letter D corresponding to the German word "Dauerfestigkeit". The fatigue strength is often determined by tests, or estimated from the tensile strength thanks to a fatigue limit ratio, defined as the fatigue strength divided by the tensile strength.

The confidence of the fatigue calculation depends on the accuracy of the fatigue strength which is influenced by a many factors : strip valve thickness, surface roughness, residual stress, temperature effect ... and failure risk allowed. The corrosion and particular metallurgical composition or inclusion have also influence, but there is no analytical formulation available up to now to take into account those factors.

The reference [5] gives a list of the effects of factors influencing the endurance performance of the strip steel material for compressor valve. The surface finishing quality is very important since cracks can be initiated at surface defects. The residual stresses depend on the process used to obtain the valve. Compressive residual stresses increase fatigue strength, while tensile residual stress decrease it. Valve edges in stamped condition present tensile residual stresses. Tumbling operation bring compressive residual stresses.

The residual stresses have a major influence on the fatigue performance. Blanking operation on valve brings tension (worst typical value is around + 100 MPa (+ 14500 psi)). Tumbling operation brings compression (best typical value is around - $0.25 \times Rm$). Those typical values can be measured, by X ray diffraction, for example, at the surface and in the thickness of the valve. Negative stress from compression are favorable along the fiber in the direction of bending load, where the stress is maximum. Residual stresses decrease in the thickness and become even positive in the central fiber. Typical residual stresses repartition is presented in the reference [8]. The residual stresses can't be ignored when running a fatigue calculation on compressor valves. Normally, the residual stresses should be added at the mean stress amplitude. The consequence in the Haigh diagram is to have less safety if the residual stresses are positive (tension) and more safety if the residual stresses are negative (compression). Due to accommodation, only a certain amount (between 40 to 60%) of those residual stresses remain, after very first running cycle of the valve, in the compressor.

An analytical equation, presented in the reference [7], for a certain kind of strip steel, proposes to take account of the tensile strength Rm, the surface finishing of the valve Ra, the thickness of the valve t, and the residual stresses σR, in the determination of the fatigue strength. In this equation, the residual stresses are acting directly on the fatigue strength, by increasing it, if compression, and by decreasing it, if tension . The residual stresses σR from compression considered positive in the equation of reference [7], increase the endurance limit. The fatigue strength is often estimated from the fatigue limit ratio σ_D / Rm . Depending of supplier strip steel material, this ratio is generally included between 0.35 and 0.5 .

Whatever the method used to estimate the initial fatigue strength σ_D, calculation or experiment, the failure risk chosen in the estimation of the fatigue strength need to be known. The technical data available often give the fatigue strength for a 50% probability of failure. A Weibull model can help in choosing the suitable level of confidence required according to defect rate objective. The confidence level advised, to insure a reliable valve design, requires to run calculation for a failure risk of 0.1%. We made the assumption that the endurance limit follows a Gauss Model. In order to obtain a confidence level of 99.9%, it is necessary to consider the mean value of the fatigue strength, minus three times its standard deviation. The standard deviation often considered on fatigue strength is at around 5% of the mean value. To obtain the fatigue strength at 0.1%, we decrease the value at 50% by 15%, based on the fact that the mean value at 50% is decreased of 3 times the standard deviation :

$$\sigma_D(99.9\%) = 0.85 \times \sigma_D(50\%)$$

To end with the factor's influence on the mechanical properties, it is necessary to pay attention to the temperature effect. Data on tensile strength and fatigue strength are often given at 20°C (68°F). It is important to keep in mind that valves in compressor can reach temperature value of 200°C (392°F). Based on information available, a decrease of 10% on both the tensile strength and fatigue strength, can be proposed :

$$\sigma_D(200°C) = 0.9 \times \sigma_D(20°C)$$

$$Rm(200°C) = 0.9 \times Rm(20°C)$$

Step by step, the two main mechanical properties need to known to use the Haigh diagram, have been estimated with some major influence such as valve thickness, surface finishing, residual stress, temperature, and statistical constraints. Other factors, which could also have influence on the fatigue properties are not considered : scale effect between test sample and effective valve dimension, test frequency versus effective operating frequency of valve in compressor,... .

Haigh diagram and endurance models

The Haigh diagram represents the stress state, reduced to a mean stress amplitude and an alternating stress amplitude in a plan. The points, representing the stress state ($\sigma m, \sigma a$), are plotted in the Haigh diagram. In order to insure reliability, by preventing damage and fracture, the points must stay within a safe or an admissible area, defined by the mechanical properties, according to different possible models. Many models are available to define the admissible area. Among them, we can recall the now well known models, such as Söderberg, Goodman, Gerber ... . The choice of a model is very important because it defines the admissible stress, which is used in the safety coefficient calculation.

Three main models have been analyzed : Goodman, Söderberg, and the VDI2226 recommendation described in the references [1], [2], and [3]. The Gerber model too optimistic, fatigue point of view, was dropped. We point out the analysis on the positive side of the horizontal scale in the Haigh diagram, corresponding to the mean stress amplitude axis. All the models considered, propose an equation to determine the alternating stress amplitude σa at the limit, for a given σm :

Söderberg : $$\sigma a = \sigma_D \left(1 - \frac{\sigma m}{\sigma e}\right)$$

Goodman : $$\sigma a = \sigma_D \left(1 - \frac{\sigma m}{Rm}\right)$$

VDI2226 : $$\sigma a = \sigma_D \left(1 - \frac{\sigma m}{2 \times Rm - \sigma_D}\right)$$

We analyze how each model match the point $\sigma m = \sigma a = \sigma_0 / 2$. The intersection between the previous equations and the equation $\sigma m = \sigma a = \sigma_0 / 2$, gives the calculated value for σ_0 :

Söderberg : $$\sigma 0 = \frac{2 \times \sigma_D}{1 + \frac{\sigma_D}{\sigma e}}$$

Goodman : $$\sigma 0 = \frac{2 \times \sigma_D}{1 + \frac{\sigma_D}{Rm}}$$

VDI2226 : $$\sigma 0 = \frac{\sigma_D}{Rm} \times \left(2 \times Rm - \sigma_D\right)$$

Based on different available data on mechanical properties (σ_D ,Rm ,σe) required to do the analysis, we notice that the VDI2226 model gives estimated value of σ_0 very close to the typical value from technical data. The VDI2226 model recommendation is suggested to be used to define the admissible endurance area in the Haigh diagram, based on the actual available data analyzed.

Admissible stress and safety coefficient

The admissible stress σadm is the alternating stress amplitude determined, for a given mean stress amplitude σm, by the set of equations presented below, based on the admissible area defined thanks to the VDI2226 recommendation.

If $\sigma m < \dfrac{\sigma_D - Rm}{1+\dfrac{\sigma_D}{2\times Rm-\sigma_D}}$ then $\sigma adm = Rm + \sigma m$

If $\dfrac{\sigma_D - Rm}{1+\dfrac{\sigma_D}{2\times Rm-\sigma_D}} \le \sigma m < Rm - \dfrac{\sigma_D}{2}$ then $\sigma adm = \sigma_D\left(1-\dfrac{\sigma m}{2\times Rm-\sigma_D}\right)$

If $\sigma m \ge Rm - \dfrac{\sigma_D}{2}$ then $\sigma adm = Rm - \sigma m$

The safety coefficient S is defined as the ratio between the admissible stress σadm and the alternating stress σa applied.

$$S = \frac{\sigma adm}{\sigma a}$$

As a preliminary design criterion, it is possible to propose the following advice values, for a very first analysis, at early step of design process. Safety coefficient above 1.5 is required to insure a conservative level of reliability, which should lead to infinite life. Safety coefficient between 1 and 1.5 imply marginal design for which the confidence level is not adequate to infinite the life requirement. Safety coefficient below 1 is unacceptable.

Discussion

The concept of safety coefficient in fatigue is an easy approach for the fatigue life prediction, because it can be comprehensive for engineering product designer, who already uses safety coefficient in static analysis, for example. To well understand the limit of such approach, we have to keep in mind that many approximation are assumed, specially on the stress history treatment. The analysis of the stress state applied doesn't take into account the cycling effect of the load, and the evolution versus time. The reduction of stress state history to an uniaxial equivalent maximum stress doesn't represent the possible multiaxial variable stress history.

Next development in fatigue life prediction for compressor valve, need to pay attention to the progress made during recent years in fatigue assessment, for other industrial product, with the new concept of damage indicator definition, as a fatigue criterion.

CONCLUSIONS

A method to determine the safety coefficient in fatigue of compressor valve has been described. A stress state analysis, and a summary of the main mechanical properties and some influencing factors, have been listed. Three models limiting the admissible area in the Haigh diagram, have been presented. Among them, the VDI2226 was proposed as a representative one to determine the admissible alternating stress amplitude, for a given mean stress amplitude. The safety coefficient has been defined, and some advised values proposed.

The method presented is a first step to help designer to deal with stress results from calculation. New concept of damage indicator as a fatigue criterion need to be investigated to improve the confidence level of fatigue calculation, to obtain a better fatigue life prediction, with less approximation or assumption, discussed.

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