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Fatigue Modelling of Reed Valve as a Function of Crank Angle for Piston Compressor

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Abstract - *Reciprocating compressors are used to deliver high-pressure gas. As the limiting elements in the design of the reciprocating compressor, the valves can be considered. These valves are heart of the compressor, after failure, it would lead to the shutdown of the compressor and to costly downtimes. A compressor running at even moderate speeds such as 700 rpm requires for each valve to open and close over one million times a day. Valve design be highly reliable and operate efficiently in adverse conditions, such as in applications where liquids, debris in gas stream. Among all cause of failure of reed valve, fatigue is being major reason. As reed valve acts as a vibrating system, it undergoes thousands of stress cycles within a minute; which causes fatigue failure. In this project, the reasons for fatigue failure are discussed. Also, the factors affecting fatigue failure of valve will be evaluated. Bending stress due to lift of valve will be considered for fatigue life calculation. The SN curve will be drawn to calculate fatigue life which will be used to evaluate life of valve. The natural frequency of valve will be calculated and finally fatigue stress will be related in terms of cack angle*

Key Words: fatigue life, reed valve, fatigue stress, Reciprocating compressors

1 INTRODUCTION

Reciprocating compressors are one of the most commonly utilised type. They're utilised in a wide range of industries, including the oil and gas sector and the chemical industry, where they're primarily used for their capacity to deliver high-pressure gas. Because piston compressors are such an important aspect of any process in which they are used, their dependability has attracted a lot of attention.

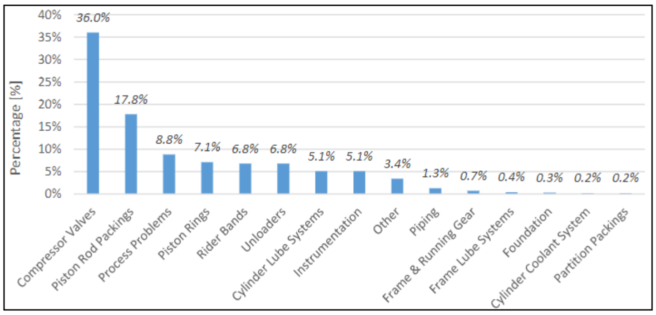


Figure -1: Reasons of compressor failure

Each valve on a compressor having normal speed of around 800 rpm, must open and close over one million times every day. As a result, when designing a valve, keep in mind that it must be not fail i.e., should be reliable and perform effectively even under difficult conditions, such as when there are liquids or debris in the gas stream.

Fatigue is one of the most common reasons for reed valve failure. Because the reed valve is a vibrating mechanism, it goes through thousands of stresses cycles each minute, resulting in fatigue failure. The reasons for fatigue failure, factors impacting valve fatigue failure, and eventually fatigue will be related in terms of cack angle in this project.

2 LITERATURE REVIEW

In 1777, James Watt built a stream engine, which us used to run an air compressor. Although the use of air compressor getting popular, valve design was not included in literature. The early attempts to explain valve behavior, however, were solely empirical, despite the lack of experimental methods for recording valve activity, such as measuring lift as a function of time.

Having [5] recorded change in design of valve from time to time. This type of evolution is shown in following Fig 2

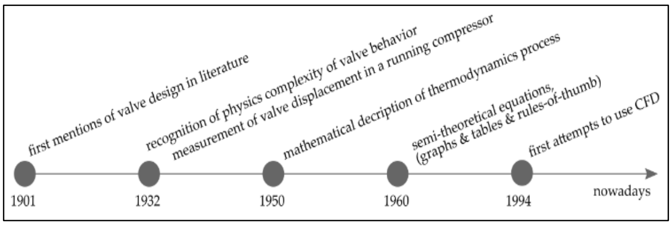


Figure -2: Timeline of evolution of valve design methods

Sittenfeld (1901) [6] published a book in which he attempts to solve valve properties using theoretical assumptions instead of empirical methods. The resultant model was simple as it was based purely on Newton's second rule and Bernoulli's equation for incompressible fluid.

Hirsh (1932) [7] published a book in in which he acknowledged the complexities of the physical condition involving valve function, which is influenced by a multitude of circumstances. Later that year, Lanzendörfer [8] used a mechanical (pressure) indicator to measure valve displacement in a running compressor.

Our review of contemporary valve design literature reveals that there is a fast-expanding number of publications these days, although they are mostly in the form of conference proceedings:

- International Compressor Engineering Conference at Purdue University
- International Conference on Compressors and their Systems of IMechE and City University in London
- European Forum for reciprocating compressors.

3 METHODS

3.1 Failure ways of reed valve

The various ways a reed valve fails are listed below. Failure of reed valve is mainly due to fracture, but the ways to initiate fracture varies widely

3.1.1 Impact Fatigue fracture

The movement of the reed valve is controlled by a back plate when it reaches its highest point during lift. Due to the rapid speed of the reed during lift, it collides heavily with the back plate and seat.

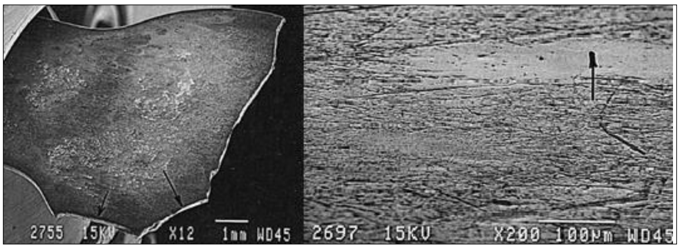


Figure -3: SEM photograph of back of reed valve plate after impact

3.1.2 Failure due to bending stress

Bending stress created in the section of the reed covering the port can cause bending, according to stress analysis of reed contact conditions. Bending the reed during contact will result in a state of stress fatigue. The bending condition is determined by the reed's thickness and the gas pressure on it. This situation can cause fatigue in the reed contact zone if it exists.

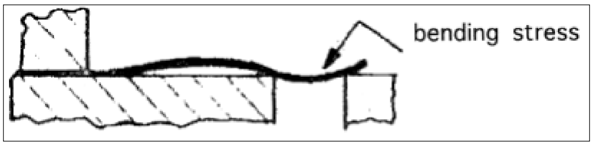


Figure -4: Simple bending stress in reed plate

3.1.3 Failure due to debris

Debris wedged between the reed and the valve plate causes the reed to fracture. The origin of the debris was considered to be valve plate material from the vicinity of the two ports implicated because the nibbling happened on two nearby reeds. The debris dents caused fatigue failure by acting as stress raisers. Cleaning the valve plates after machining is required to guarantee that all loose debris is eliminated.

3.1.4 Failure due to fatigue stress

Compressor reeds open and close a million times every day, even at a moderate speed of 700 rpm. Valve fatigue is caused by these stress cycles. As a result, the valve's lifespan is significantly reduced. We must examine the fatigue life of the valve to ensure safe functioning under cyclic loading

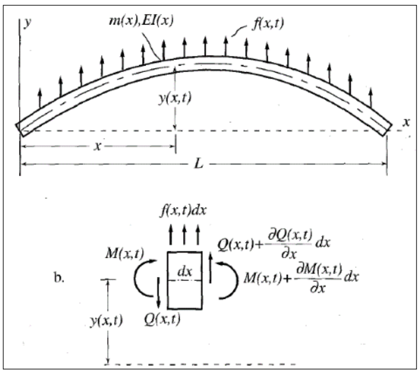
3.2 Behavior of reed valve under vibrations

Reed valves open when there is a pressure differential between the gases, and the frequency with which they vibrate is influenced by its inherent frequency as well as air movement. A cantilever deflection up to the stop and a propped cantilever deflection after touching the stop characterize the valve action. The motion of a reed is described by the oscillating motion of a bar in its fundamental natural frequencies. While oscillating at higher natural frequencies, reed behavior changes fast.

3.3 Modal analysis of a slender cantilever beam

Depending on the nature of the parameters, vibrating system models can be split into two categories: lumped and continuous. Ordinary differential equations regulate the motion of discrete systems, and there is one equation for each mass, with the number of masses defining the number of degrees of freedom of the system. When the mass of the elastic members is not negligible, the members can no longer be considered equivalent springs and must instead be viewed as continuous systems or distributed-parameter systems. We can find natural frequency as following.

Figure -5: Force analysis in beam



Force equation of motion in the vertical direction is

$$\left[Q(x,t)+\frac{\partial Q(x,t)}{\partial x}dx\right]-Q(x,t)+f(x,t)dx = m(x)dx\frac{\partial^2 y(x,t)}{\partial t^2} \quad ..(1)$$

Moment equation of motion about an axis normal to x and y and passing through the center of the cross section is

$$\left[M(x,t) + \frac{\partial M(x,t)}{\partial x}dx\right] - M(x,t) + \left[Q(x,t) + \frac{\partial Q(x,t)}{\partial x}dx\right]dx + f(x,t)dx\frac{dx}{2} = 0 \quad ..(2)$$

Bending moment is related to the bending displacement by

$$M(x,t) = EI(x)\frac{\partial^2 y(x,t)}{\partial t^2} \quad ..(3)$$

After solving above equations simultaneously and putting external force zero, we get

$$\frac{d^2}{dx^2}\left[EI(x)\frac{\partial^2 y(x,t)}{\partial t^2}\right] = \omega^2 m(x)Y(x) \quad ..(4)$$

Solution of similar problem is

$$F(t) = C\cos(\omega t - \varphi) \quad ..(5)$$

From equation (5), rearranging terms and taking constant terms out of differentiation, we get

$$\frac{d^4Y(x)}{dx^4} - \beta^4 Y(x) = 0\,;\ \beta^4 = \frac{\omega^2 m}{EI}$$

The boundary conditions are

$$Y(x) = 0, \frac{dY(x)}{dx} = 0 \ at\ x = 0$$

$$\frac{d^2Y(x)}{dx^2} = 0, \frac{d^3Y(x)}{dx^3} = 0 \ at\ x = L$$

After solving we get characteristic equation

$$\cos(\beta L)\cosh(\beta L) = -1 \quad ..(6)$$

The first three values for solution of the equations (6) are $\beta L = 1.8751, 4.6941, 7.8548$

Hence first three frequencies are

$$\omega_1 = 3.5160\sqrt{\frac{EI}{mL^4}}, \omega_2 = 22.0345\sqrt{\frac{EI}{mL^4}}, \omega_3 = 61.6972\sqrt{\frac{EI}{mL^4}}$$

After solving above expression and putting values we get displacement of valve as

$$Y_r(x) = A_r\left[\sin\beta_r x - \sinh\beta_r x - \frac{\sin\beta_r L + \sinh\beta_r L}{\cos\beta_r L + \cosh\beta_r L}(\cos\beta_r x - \cosh\beta_r x)\right]$$

3.4 Fatigue stress

Fatigue is the degradation of a material's structural qualities as a result of damage induced by cyclic or fluctuating stresses. Fatigue is defined by the damage and loss of strength produced by cyclic stresses that are less than the material's yield strength.

3.4.1 Fatigue life (S-N) curves

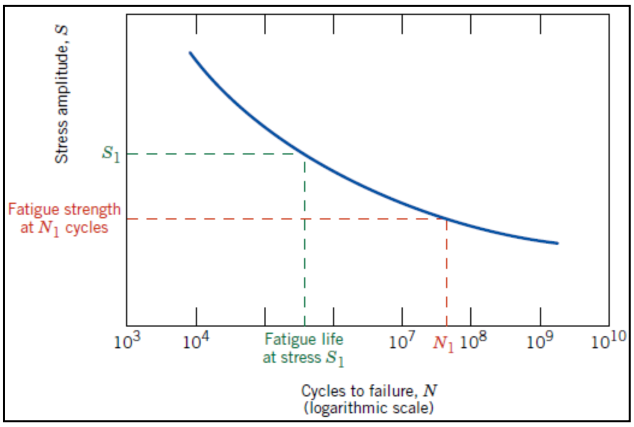


Figure -6: Behavior of material under cyclic loading

The fatigue life (S–N) graph, which is a plot of the maximum fatigue stress S (or σ_{max}) versus the number of stress cycles-to-failure of the material N, is the most basic method for evaluating the fatigue resistance of materials.

3.4.2 Fatigue-crack growth in metals

Pre-existing defects such as voids, large inclusions, or surface imperfections that function as stress raisers are more likely to produce fatigue fractures. Machining burrs, scratches, corrosion pits, and sharp corners are common pre-existing surface defects that originate fatigue fractures and must be avoided to delay fatigue crack initiation.

3.5 Fatigue strength

The maximal stress that a material can sustain for a certain number of cycles without breaking is called fatigue strength. The fatigue strength Sf is the number of cycles N to which the ordinate of the S-N diagram corresponds; a statement of this strength value must always be accompanied by a statement of the number of cycles N to which it relates

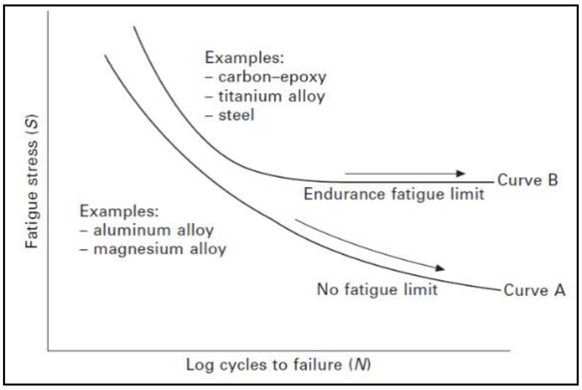


Figure -7: crack growth in metals

3.6 Fatigue strength calculation (S-N curve)

The maximal stress that a material can sustain for a certain number of cycles without breaking is called fatigue strength. The fatigue strength Sf is the number of cycles N

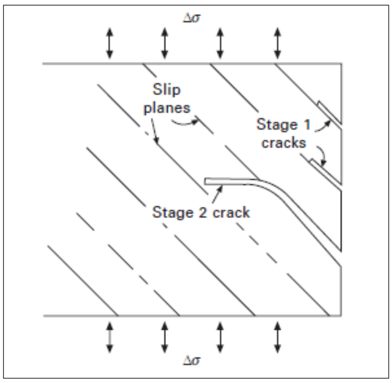


Figure -8: SN curve showing fatigue strength for fatigue life

to which the ordinate of the S-N diagram corresponds; This strength value must always be accompanied by a declaration of the number of cycles N to which it applies.

To calculate fatigue life for a given number of cycles, we must interpret the SN curve. A connection $S_f = aN^b$ can be used to approximate the SN curve. Where Sf denotes fatigue strength over N cycles, and a and b denote material constants.

3.7 Endurance limit modifying factors

The rotating-beam specimen used in the laboratory to establish endurance limits is meticulously prepared and subjected to rigorous testing. It is impractical to expect a mechanical or structural member's endurance limit to match the values determined in the lab. Some of the distinctions are as follows:

Material: composition, basis of failure, variability
Manufacturing method: heat treatment, fretting corrosion, surface condition, stress concentration
Environment: corrosion, temperature, stress state, relaxation times
Design: size, shape, life, stress state, stress concentration, speed, fretting.

To account for all factors impacting fatigue life, the Marin equation is employed. It takes into account the results of scientists' experiment and establishes parameters to account for changes in fatigue life.
A Marin equation is written as

$$S_e = k_a\, k_b\, k_c\, k_d\, k_e\, k_f\, S'_e$$

Where,
k_a = surface condition modification factor
k_b = size modification factor
k_c = load modification factor
k_d = temperature modification factor
k_e = reliability factor
k_f = miscellaneous-effects modification factor
S'_e = rotary-beam test specimen endurance limit
S_e = endurance limit at the critical location of a machine part in the geometry and condition of use.

3.8 Methods to reduce stress concentration

3.8.1 Design with progressive transitions.

A rapid change in geometry or a form transition is the most prevalent cause of stress concentrations. To account for this, designers might include specialized design elements such as fillet radii or tapers to smooth the transition between shapes. The flow of stress is influenced by smoothing out important portions in the part design, which prevents tension from accumulating in one place.

3.8.2 Use relief notches

Designers can also reduce stress concentrations by eliminating material around notches and creating a relief notch. Relief notches can be used to manage the lines of stress in a part, despite the fact that they add a little extra geometry to the design. Through repeated FAE evaluations, these characteristics can be tested and improved.

3.8.3 Opt for several small relief notches as opposed to a single long one

When holes and notches are unavoidable, utilize a large number of little relief notches. Rather than using a single groove to remedy any anomalies as in the prior technique, this strategy entails encircling each deliberate notch with multiple smaller notches to smooth out potential stress concentrations. Engineers can remove extra material while preserving the original notch to make tiny notches.

3.8.4 Avoid sharp corners

Sharp corners should be avoided in general, especially when CNC machining interior part geometries, but designers should avoid them as well if they are concerned about stress concentrations. Designers should always employ a fillet radius at sharp corners if the pattern allows it. This design feature guarantees that the cross-section area declines gradually rather than abruptly, distributing stress more evenly across the portion.

3.9 Fatigue stress mode

Our objective in analysis of fatigue is to predict the stresses in the valve causing failure. As a result, our failure criteria are connected to stresses, or structural behavior that is a result of the structure's stress fields. In an elastic system, stresses are a direct result of relative displacement and may be calculated using material constitutive relations and compatibility equations. In the spatial coordinates, stresses may always be represented by some differential operator [6]:

$$\sigma_i = KL_i[y(x,t)]$$

Where K is a constant that relies on the material characteristics and geometry of the structure, Li is a differential operator, and y(x, t) is the structural deflection as a function of the location x and time t. The index I denotes the type of stress being computed. For example, $\sigma_i = \sigma_x$ may be the bending stress in the x direction.

The bending stress in the tensed side of a loaded deflected beam equals,

$$\sigma_i = -\frac{Eh}{2}\frac{\partial^2 y}{\partial x^2}$$

The normal modes and generalized coordinates may be used to characterize the deflection as a linear function.

$$y(x,t) = \sum \emptyset_j(x) \cdot \eta_j(t)$$

Thus $$L_i[y(x,t)] = \sum L_i[\emptyset_j(x)] \cdot \eta_j(t)$$

Hence, $$\sigma_i = K \sum L_i[\emptyset_j(x)] \cdot \eta_j(t)$$

Which can be written as $$\sigma_i = \sum \psi_j(x) \cdot \eta_j(t)$$

$$\psi_j(x) = KL_i[\emptyset_j(x)]$$

$\psi_j(x)$are called stress modes. Thus, they will give us the maximum stress values that will be acting on valve. After calculating the stress modes, a comprehensive mapping of the stress field in any structural member owing to dynamic loading may be assessed

4 RESULTS AND DISCUSSION

4.1 Fatigue life calculations

To calculate fatigue life, we will use the SN curve of steel. SN curve depends on Sut of the material and fatigue correction factor. I have summarized all factors and results into an EXCEL file. Where we can input the required details and it will generate life. Below is a snap of an example of such calculations.

Ne	1000000	cycles
Sut	827.4	MPa
Sut	120	kpsi
Se '	700	MPa
sf '	1172.4	MPa
b	-0.03555	
f	1.08149	

k	0.72	
Se	504	
a	1588.7	
b	-0.0831	
Sf	290	
N	7.73E+08	
Life	29.84	months
	2.48651	years

Figure -9: Fatigue strength calculations using excel

4.2 Natural frequency calculations

To calculate natural frequency, we will use the formulas derived for rectangular plate. natural frequency depends on shape of the plate. I have summarized all factors and results into an EXCEL file. Where we can input the required details and it will generate natural frequency. Below is a snap of an example of such calculations.

Modulus of Elasticity	E	9.00E+07	N/m2
Density	Rho	7850	Kg/m3
Free Length	L	0.01227	m
Width	w	0.0258	m
Thickness	t	0.0002	m
MOI	I	1.72E-14	m4
mass per length	m	0.040506	kg/m

		Circular			
First Natural Frequency	ω1	144.37	rad/s	22.99	Hz
Second Natural Frequency	ω2	904.773503	rad/s	144.07	Hz
Third Natural Frequency	ω3	2533.39045	rad/s	403.41	Hz

Figure -10: Natural frequency calculations using excel

4.3 Modes of valve

Valve of following dimensions is taken

Thickness = 0.0007 m Width = 0.025 m

Length = 0.061 m Free lift at tip = 2.5 mm

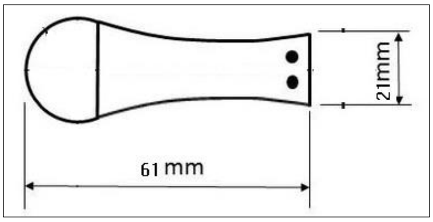


Figure -11: Case valve dimensions

Using MATLAB to generate results, we get graph as in fig. 12

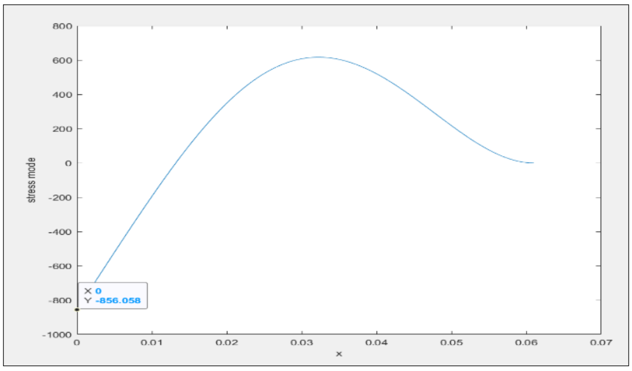


Figure -12: Second stress mode of valve

Results are compared with similar analysis done by S. Papargiou [9] and McLean [10]

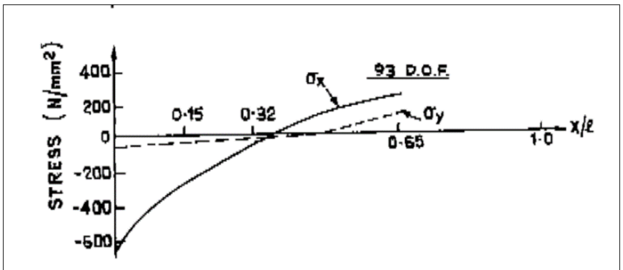


Figure -13: Results of second stress mode of valve [9]

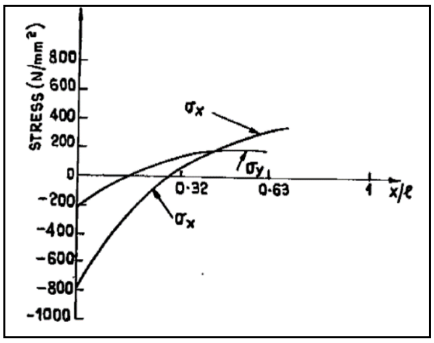


Figure -14: Results of second stress mode of valve [10]

Comparison between result is

	Result	Result [9]	Result [10]
Bending stress at root of valve (N/mm^2)	856	795	703

5 CONCLUSIONS

The general approach to handling fatigue in valve design is addressed. We can compute the real-world fatigue strength of the desired material using fatigue correction parameters. We can identify the modal shapes and frequency of valves using modal analysis, which may then be utilised to map valve stresses. Compressor valve fatigue failures are explored, as well as possible solutions.

ACKNOWLEDGEMENT

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NOMENCLATURE

y	Deflection of tip of valve	[m]
σ	stress	[N/mm^2]
ω	Natural frequency of valve	[Hz]

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