Available online at http://www.journalcra.com

International Journal of Current Research
Vol. 8, Issue, 07, pp.33988-33995, July, 2016

INTERNATIONAL JOURNAL OF CURRENT RESEARCH

ISSN: 0975-833X

RESEARCH ARTICLE

FAILURE ANALYSIS OF GIMBAL TYPE EXPANSION JOINT CARRYING FLUE GASES

*Sagar Desai and Prof. G. D. Korwar

Pune University, India

ARTICLE INFO

Article History:
Received 20th April, 2016
Received in revised form
15th May, 2016
Accepted 04th June, 2016
Published online 16th July, 2016

Key words:
Fea,
Radioss,
Frequency response analysis,
Expansion joint,
Failure analysis.

ABSTRACT

Expansion joints used to absorb any misalignment due to positioning axial angular or due to thermal expansion of pipes. In large steel reduction plants corex process is implemented. In corex process all compressed hot gases reused and are carried to each station by pipes. To ensure safe working with these hot pressurized gases expansion joints used in piping system. Expansion joint provides flexibility to piping systems. Different expansion joints can be used to give flexibility in different direction like axial, angular and lateral. Aim of this project was to investigate the failure and modify the design of expansion joint used in steel plant. Finite element method is used for modal and Frequency response analysis. Modal analysis results which are basis for the frequency response analysis are validated with the analytical results. Fatigue life is also calculated considering maximum and minimum stresses from FRF analysis. Frequency response is considered for analysis based on failure pattern. Pulsating frequency of compressor is considered as the excitation frequency for expansion joints. After analyzing the results from the frequency response analysis modifications implemented from EJMA standards. All implemented modification are reanalyzed. Modification with reversed sleeve is found suitable for desired working.

Citation: Sagar Desai and Prof. G. D. Korwar, 2016. "Failure analysis of Gimbal type expansion joint carrying flue gases", *International Journal of Current Research*, 8, (07), 33988-33995.

INTRODUCTION

An expansion joint can be defined as a device containing one or more bellows used to absorb dimensional changes in the pipe line, duct or vessel caused by thermal expansion or contraction. The main function of expansion joint is to provide flexibility in the piping in order to compensate for variations in length. The expansion joint possess several advantages over conventional compensating devices such as capable of absorbing any movement, compact arrangement, and maintenance free long life. The function of expansion joints in various applications is to provide flexibility in the piping. The performance of expansion joint is mainly depends on the design of bellows. The whole assembly of expansion joint consists of elements like liner, flanges, sleeve, cover, collar etc. Bellow is welded with the collar and collar material, which is further welded with the flanges. Generally expansion joints are manufactured along with flanges in order to facilitate the assembly in long piping. The flanges are connected with flanges of pipe with fasteners. In current paper failure of expansion joint is analyzed. Two expansion joints are considered RGC-2 for analysis. Different analysis is carried out to find out the causes of failure of expansion joints.

**Corresponding author: Sagar Desai,*
Pune University, India

Problem definition

Expansion joints

There are 5 Turbo Compressors installed in plant. Out of 5 Centrifugal compressors 3 are Corex EGC (Export Gas Compressors) & 2 are RGC (Recycle Gas Compressors). Each stage of compressor is connected to the piping with metallic bellow type expansion joint at discharge side.

Table 1. List of expansion joints with different inlet and outlet conditions

S. No	Equipment	Suction press(bar)	Discharge press(bar)	Handling Media
1	RGC	0.4	1.5	Top gas

Expansion joints installation & replacement history

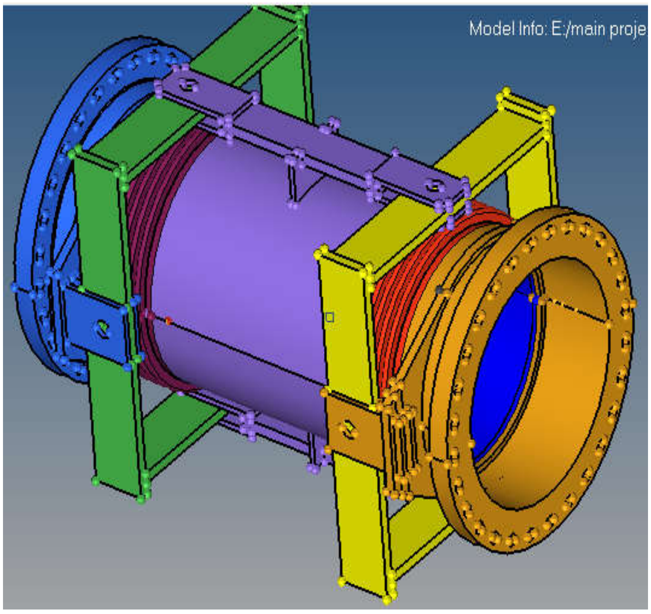
All bellows with 1.6 mm thick SS316 sleeves installed, First failure reported in one month, Bellow Leak in RGC-2 discharge is observed. Bellow having 3 mm thick SS316 sleeveis also observed in damaged condition.

Objectives

1. Study of expansion joints.
2. Detailed study to investigate the causes for the Expansion Joint failures.
3. Provide suitable design modifications and operating parameter modifications to ensure reliable performance at the given operating conditions.
4. Reanalyze the modified design.

Finite element analysis

Recycle Gas Compressor expansion joint (RGC-2)



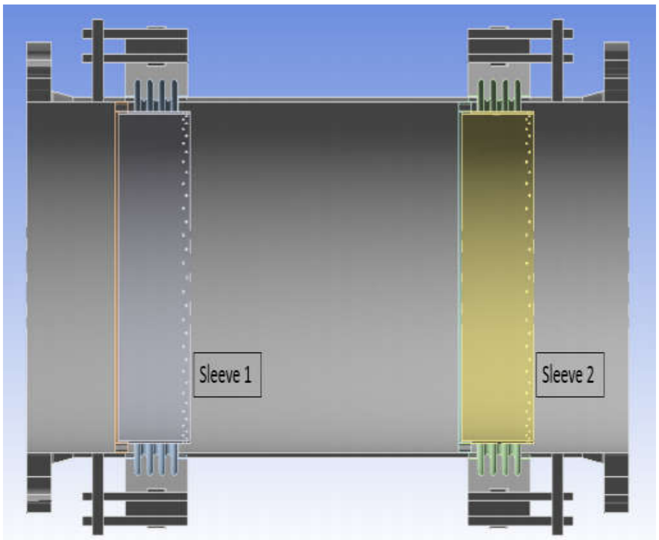


Figure 1. a) Isometric view of RGC2 expansion joint, b) Cut section view for RGC2

Gimble type Universal expansion joint as shown in figure used to mainly to take angular movements. Diameter is 800mm and length is 1000mm. Ends bolted to the compressor outlet on one side and pipe on other side. Three models are analyzed with different sleeve thickness of 10, 6, and 3mm.

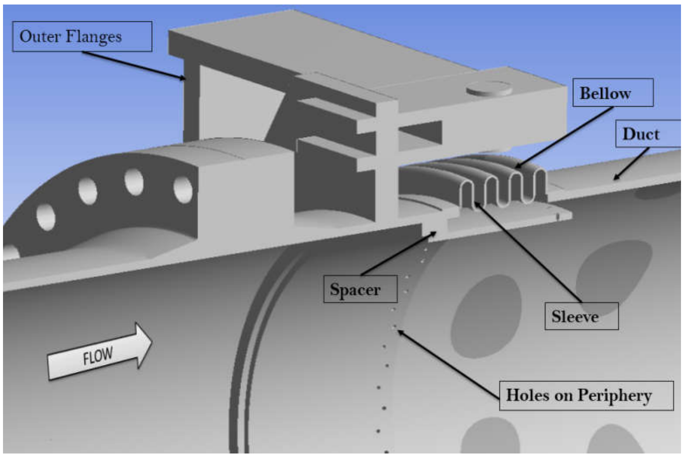


Figure 2. Detailed cut section view of RGC2 model

Material and boundary conditions

Excitation force: pressure is used as the excitation force, Frequency range: 1675-1775 Hz, Modal analysis method: Lancoz method, FRF analysis method: Mode superposition method, Damping ratio: 0.05Young'smodulus (E): 2.1E5, Density (Rho): 7.8E-6, Poisson's ratio: 0.3

MODAL ANALYSIS RESULT COMPARISON

Analytical modal analysis

As formula given by the EJMA analytical results were calculated. Modal frequency for axial vibration is considered for comparison. Bellows axial spring rate mostly deals with geometry of the bellow and is derived experimentally. Overall spring rate is then calculated in order to get overall bellow spring rate per convolution. Using this overall spring rate per convolution and weight supported by the whole model natural frequency derived for the axial movement of the bellow.

Analytical calculations

According to Expansion joint manufacturing association edition 9 axial vibration of double bellow expansion joint is as follows

$$fn = 4.43 \sqrt{\frac{Ksr}{W}}$$

Ksr Overall bellows spring rate

$$Ksr = \frac{f1}{N}$$

f1 Bellows axial elastic spring rate

$$f1 = \frac{1.7 \quad Dm \quad Eb \quad tp^3 \quad n}{w^3 \quad Cf}$$

Where
fn -Axial vibration frequency

Ksr - Overall bellows spring rate (lb./in.)
N - Number of convolution in one bellow
$f1$-bellows initial axial spring elastic spring rate per convolution
Dm -Mean diameter of bellows convolution (in.)

$$Dm = Db + w + nt$$

Db -inside diameter of cylindrical tangent and bellow convolution
w -convolution height (in)
Eb modulus of elastic at design temp
tp- Bellow material thickness for one ply corrected for thinning during forming

$$tp = t \sqrt{\frac{Db}{Dm}}$$

t -nominal material thickness of one ply (in)
Cf –factors used in specific design calculation to relate U-shaped bellows to relate u shaped convolution segment behaves to a simple strip beam determined experimentally.

FEA modal analysis

Modal analysis is used to get natural frequency of the model. Mode shapes derivedso as to get rough idea about how model behave in certain conditions. Lancoz method is used for modal analysis. The Lanczos method has the advantage that the eigenvalues and associated mode shapes are calculated exactly. This method is efficient for calculations in which the number of modes are large and the full shape of each mode is required. The basic linear differential equation of motion, of a multi-degree-of-freedom (m.d.o.f.) structure, is given by

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+[K]\{x\}=\{F(t)\} \quad (1)$$

Where [M] is the mass matrix, [C] the damping matrix, [K] the stiffness matrix, $\{x(t)\}$the displacement vector, $\{\dot{x}(t)\}$ the velocity vector, and $\{\ddot{x}(t)\}$ the acceleration vector. The above equation balances the structure's internal forces, which are a combination of mass (inertial), damping (dissipative), and stiffness (elastic restoring) terms (referred to the spatial model) with the externally applied forces. Defects existing in a structure cause a change in its stiffness, and could also affect its mass distribution, and damping properties. Consequently, there would also be a change in the dynamic response of the structure. In addition to Eq. (1) shown above, the linear dynamics could also be represented by other equivalent expressions such as the FRF, modal parameters or the impulse response function. Fixed fixed modal analysis is carried out using FEA Lancoz method. Fixed Fixed modal analysis analytically carried out using the formulas from EJMA standard.

RESULTS AND DISCUSSION

Table 3. Analytical and FEA modal analysis results for RGC expansion joint

First modal frequency	Analytical(Hz)	FEA(Hz)	Error %
RGC-2	54.13	49.32	8.80

Discussion on results

1. In case of sleeve and bellows 2d modelling is done by extracting midsurfaces as thickness is small
2. Contact is defined between sleeve-spacer, Spacer-duct and other many places which will add non linearity to analysis
3. Material is not homogeneous in actual case due to imperfect manufacturing methods

Frequency Response Analysis (FRF)

Frequency response analysis is used to calculate the response of a structure. Analysis is to compute the response of the structure which is actually transient, in static frequency domain. The loading case used is sinusoidal loading. The excitation in this case is pressure.

Direct frequency response analysis

The direct frequency response analysis computes the structural responses directly at discrete excitation frequencies Ω by solving a set of complex matrix equations.

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+[K]\{x\}=P\ e^{i\Omega t}$$

The quantity Ω is loading angular frequency. The harmonic motion assumes a harmonic response.

$$x=d*e^{i\Omega t}$$

where is displacement. Then complex matrix for dynamic analysis can be derived that has the following real and imaginary parts:

$$\left|K - \Omega^2 M + iGK + iK_E - i\Omega B_1\right| d e^{i\Omega t} = P e^{i\Omega t}$$

The matrix K is stiffness matrix, matrix M is mass matrix.

There are three ways to define damping in the system.

1. Using a uniform damping coefficient G.
2. Structural element damping using the damping coefficients GE on the materials as well as GE on bushing and spring element property definitions. These form the matrix Ke
3. Viscous damping formed by thr damper elements. These form the matrix B.

Further equations solved directly by complex algebra

Modal frequency response analysis

Modal method first performs normal modes analysis to obtain the eigenvalues

$$\lambda = \omega^2$$

And the corresponding eigenvectors u= {U} of the system. The response can be expresses as a scalar product of the eigenvector U and modal responses d.

$$x = Ude^{i\Omega t}$$

The equation of motion without damping is then transformed into modal coordinates using the eigenvectors.

$$\left[-\Omega^2 X^T M X + X^T K X\right] d e^{i\Omega t} = X^T P e^{i\Omega t}$$

The modal mass matrix XMX and the modal stiffness matrix XKX are diagonal. If the eigenvectors are normalized with respect to the mass matrix, the modal mass matrix is unity matrix and the modal stiffness matrix is a diagonal matrix holding the eigenvalues of the system. This way the system equation is reduced to a set of uncoupled equations for the components of d that can be solved easily.

With including damping

$$\left[X^T K X - \Omega^2 X^T M X + i\Omega X^T K X + i X^T K_E X - i\Omega X^T B_1 X\right] d e^{i\Omega t} = X^T P e^{i\Omega t}$$

Here the matrices XKX and XBX are generally non diagonal. Then coupled problem is similar to the system solved in the direct method, but of much lesser degree of freedom. It is solved using the direct method.

In this case the modal frequency response analysis is adopted. In this case excitation is Pulsating frequency due to motor vanes. Pressure is used as the excitation parameter which is caused due to the flowing gases.

Compressor speed (s) is 7400 RPM.

Number of vanes (n) on impeller are 14.

$$f = \frac{s \quad n}{60}$$

Excitation frequency (f) is equal to 1726 hz. Considering fluctuations in the speed of compressor, frequency bandwidth for analysis is 1675 to 1775 hz. And 0.15 Mpa is excitation pressure force. Frequency response analysis is carried out in this range maximum and minimum stresses are considered for the fatigue analysis.

Fatigue life

Fatigue is a phenomenon associated with variable loading or more precisely to cyclic stressing or straining of a material. Metallic components subjected to variable loading get fatigue, which leads to their premature failure under specific conditions. Fatigue loading is primarily the type of loading which causes cyclic variations in the applied stress or strain on a component. Thus any variable loading is basically a fatigue loading. Variable loading results when the applied load or the induced stress on a component is not constant but changes with time i.e load or stress varies with time in some pattern. Most mechanical systems and devices consists moving or rotating components. Maximum and minimum stresses in the range of frequency 1675 to 1775 are considered for the fatigue life calculation. Different Modification factors used to get the endurance limit in actual situation. Goodman's criteria used to calculate Sf in actual case

Se = ka*kb*kc*kd*ke*kf *Se'

ka =surface condition modification factor,
kb= size factor
kc=loading factor,
kd=temperature factor
ke=reliability factor
kf= miscellaneous factor
Se'= rotary beam endurance strength
Se= for current loading condition endurance strength

RESULTS AND DISCUSSION

Table 4. Results for RGC-2

Sleeve thickness(mm)	Max Stresses on sleeve (MPa)	Min Stresses on sleeve (MPa)	Fatigue life (days)
10	21.59@1770	2.08@1660	15
6	12.92@1770	1.8@1660	120
3	27.35@1750	4.589@1670	27
1.6	41.99@1750	8.54@1675	2

Discussion on results

1. Most of cases the sleeve attached to middle sleeve is failing. Reason behind it was energy gets accumulated at the second bellow and being less flexible sleeve at the spool was failing.
2.

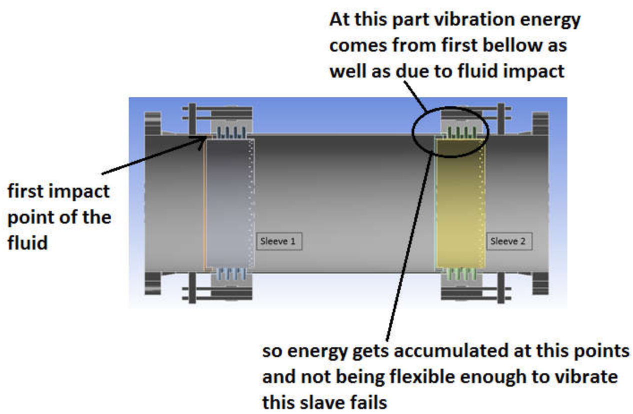


Figure 4. Analysis for failure of RGC2 expansion joint

3. Failure of RGC-2 sleeve is due to the pulsating operating frequency is close to that of natural frequency of the model
4. Failure is due to cyclic fatigue caused due to variation in pulsating frequency due to variation in motor speed.
5. In actual the fatigue life is more than what we received by calculations because it is assumed that in each cycle the stress level goes from 1675 to 1775 but in actual case it doesn't happen. Most of the cycle's maximum and minimum stress won't reach the extreme values.
6. Attempts should be made to increase or decrease the stiffness and mass combination of the model so that it will be out of resonance zone.

7. And joints like at second bellow should be made more flexible.

Modifications

Failure in frf is due to matching the natural frequency to the working frequency. So any attempt to change mass or stiffness will change the natural frequency. So certain rough methods are provide by EJMA for sleeves to alter the natural frequency. From EJMA standards the modification of sleeve are been made as follows

1. Adding rim to sleeves
2. Adding stiffeners to sleeve
3. Making floating sleeve fixed

Adding rim to sleeves:

As the component is failing in modal analysis any change in mass and stiffness will cause change in natural frequency. So rim is added so as to take natural frequency of the sleeve away from working range. Different sizes of rim been used with different combination of sleeve thickness. 4 and 3 mm thick rim is been used with 10 and 3mm sleeve thickness. Results used are been listed into table along with fatigue life as shown below

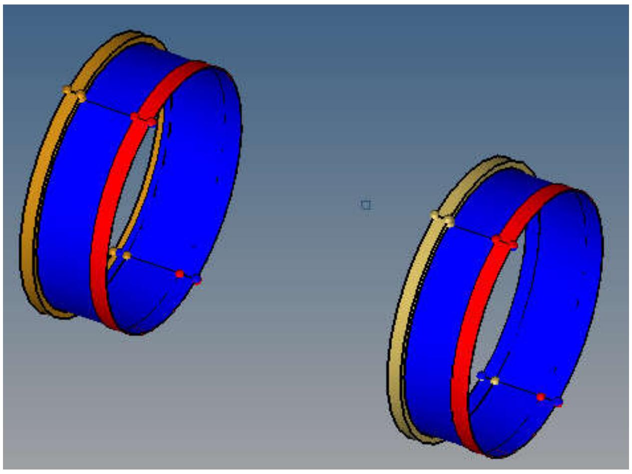


Figure 5. Sleeve with rim attached at the end

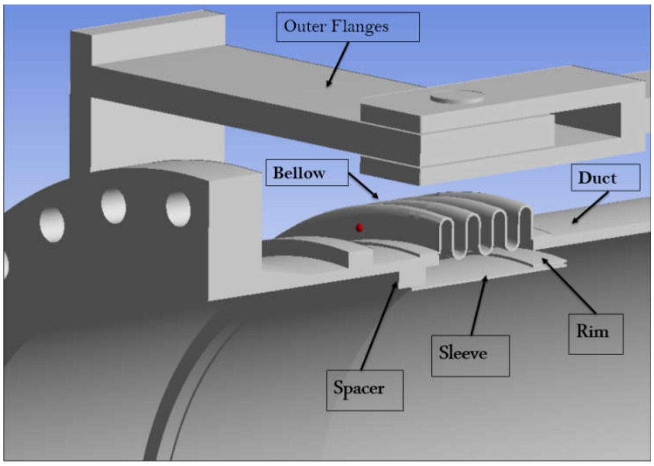


Figure 6. Detailed view of sleeve with rim attachment

Table 5. Results of FRF analysis and fatigue life analysis after attaching rim

Sleeve thickness (mm)	Rim thickness (mm)	Max Stresses on sleeve (MPa)	Min Stresses on sleeve (MPa)	Fatigue life (days)
6	4	25.1@1775	9.26@1675	29
3	4	25.18@1775	9.244@1675	28
3	3	24.46@1775	9.0@1675	31

Discussion on results

1. Adding rim to the sleeve had almost no effect on stress
2. Adding rim we are actually not varying the mass and stiffness much so ultimately not changing natural frequency much.
3. Adding rim is not much affecting the flexibility

Adding stiffeners to sleeve

As seen from above results rim is not much contributing to natural frequency so axial stiffeners are been added to increase the axial stiffeners of the model. And checked with FRF analysis total of 8 stiffeners are been added axially.

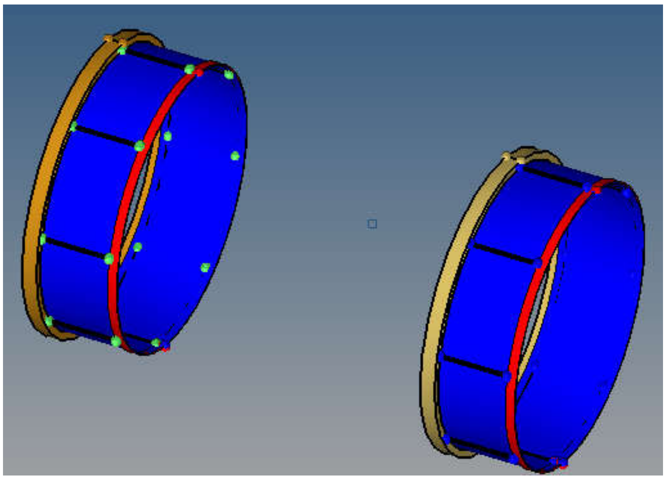


Figure 7. Sleeve with rim and axial stiffener arrangement

Stiffeners are added to increase axial stiffness of bellows. 8 stiffeners are equally distributed around circumference of the sleeve are added

Table 6. Results of FRF analysis and fatigue life analysis after attaching rim and axial stiffener

Sleeve thickness (mm)	Rim thickness (mm)	Axial stiffeners	Max Stresses on sleeve (MPa)	Min Stresses on sleeve (MPa)	Fatigue life (days)
6	3	8	15.01@1775	9.55@1675	100
3	3	8	22.0@1750	14.88@1675	41

Discussion on results

1. Adding stiffener and rim to the sleeve had almost no change on fatigue life

2. Adding stiffener and rim have varied the mass and stiffness combination but it has made natural frequency more close to that of the operating frequency
3. It is reducing the flexibility of the stiffener and making stiffer.
4. Stiffness with some other parameter will define energy so in this case we are increasing the stiffness means we are giving more scope to energy accumulation and also not giving enough room to release energy from flexibility point of view. Braking of material means energy absorbed and utilized to brake the bonds and flexible by braking bonds.

Spacer position modification

In all cases maximum stress are been observed on the sleeve which is attached on the floating duct i.e. middle spool. The sleeve which is attached to fixed duct have considerably less stresses. So the spacer is attached to the fixed duct instead of floating duct. And extra baffle is added. Baffle will steam line the flow and also will stop gases from entering the cavity between the sleeve and bellows. As the sleeve is attached to fixed ends its and duct to which it is attached is also fixed so it almost act as fixed sleeve on one side. So stress in the sleeve are been minimized.

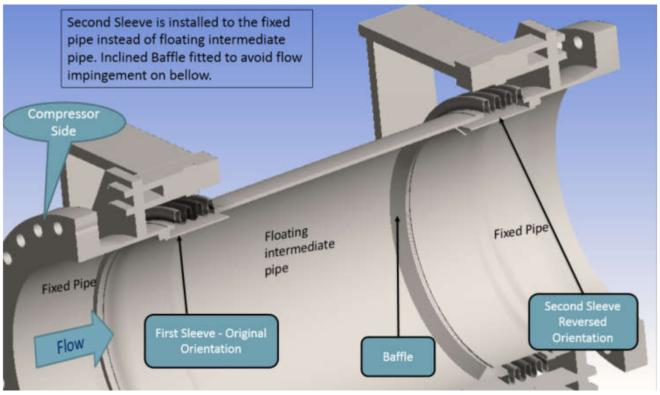
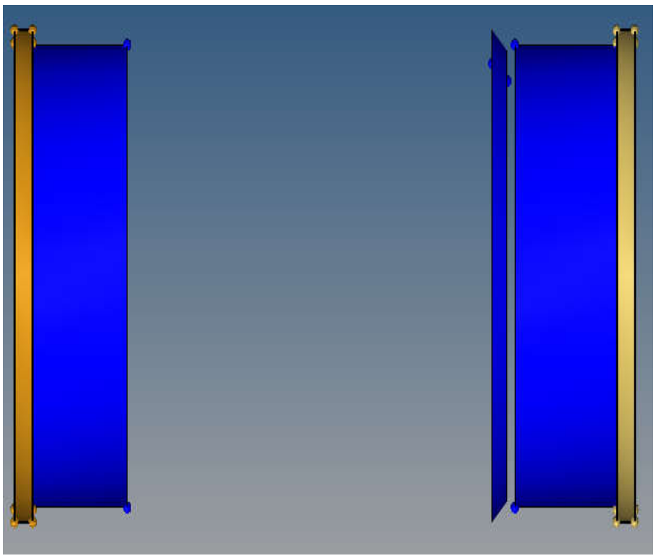


Figure 8. Detailed view of new sleeve location and baffle attachment



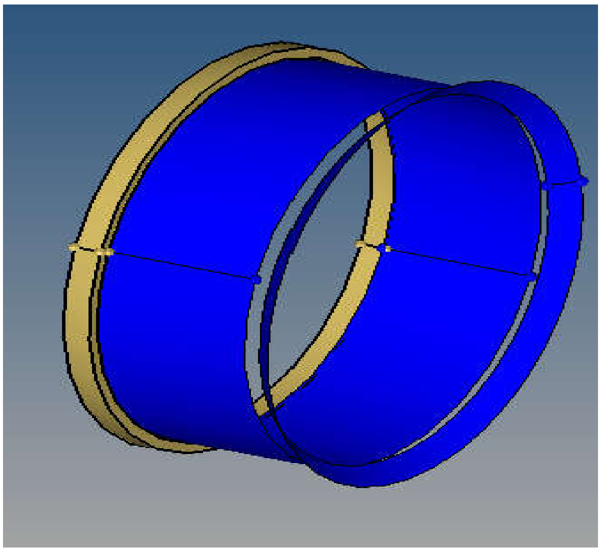


Figure 9. a) Original 1st sleeve and 2nd sleeve with baffle attachment and reversed position b) Isometric view of 2nd sleeve with baffle and reversed condition

Table 7. Results of FRF analysis and fatigue life analysis after modifying the location of the sleeve

Sleeve thickness (mm)	Max Stresses on sleeve (MPa)	Min Stresses on sleeve (MPa)	Fatigue life (years)
3	11.78@1775	6.565@1675	3
6	5.28@1775	1.42@1700	4

Discussion on results

1. As second impact point at second bellow is removed so less energy is accumulated at second bellow and also baffle added to second bellow will make the flow more streamlined.
2. As middle spool is vibrates freely in between to bellows and also can sustain the vibrations transmitted. Removal of spacer and sleeve at second sleeve made spool more flexible
3. Second sleeve is attached to the duct which is fixed on one side and most of vibrations also get arrested in between two bellows so less stress observed in this case

REFERENCES

"Standards of The expansion joint Manufacturers Association, inc.", Ninth Edition

Apoorv Bapat," Practical Aspects of Finite Element Simulation".

Bijayani Panda, M. Sujata, M. Madan, S.K. Bhaumik," 2014. Stress corrosion cracking in 316L stainless steel bellows of a pressure safety valve", *Engineering Failure Analysis,* 36 379–389

Englund T., Wall J. Vaidutis Feliksas Jakuskas, 1995. "Transverse vibrations of bellows expansion joints", McMaster University, June.

Gawande S. H., N. D. Pagar, V. B. Wagh, and A. A. Keste, 2015. "Numerical Investigations on Characteristics of Stresses in U-Shaped Metal Expansion Bellows", *International Journal of Metals*, Volume 2015, Article ID 957925

Shigley's," Mechanical Engineering Design", Eighth Edition

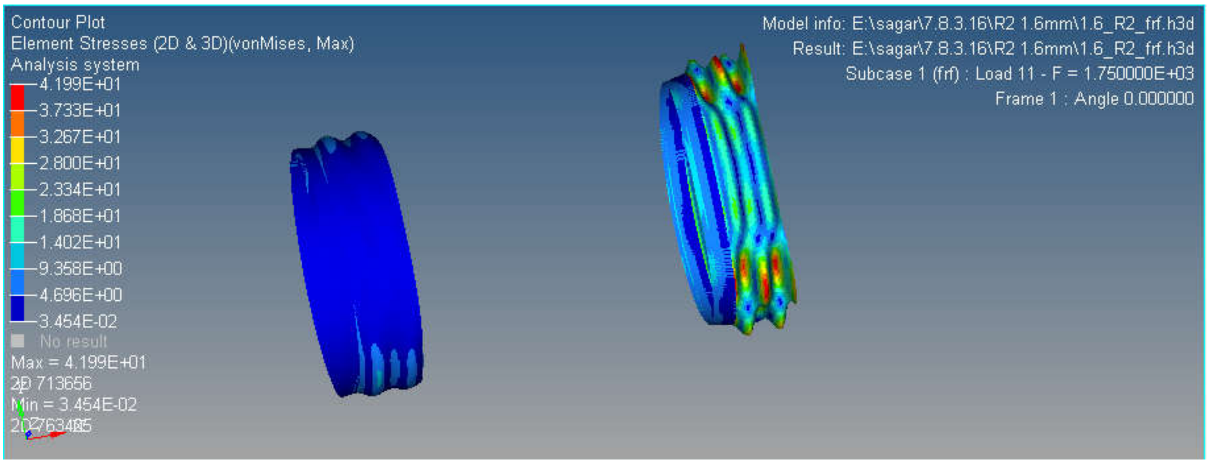
Sunil Kumar Shetty, Raghunandana, 2015. "Transverse vibration of expansion joints", Engineering Failure Analysis 40, 569–251.

Vaidutis Feliksas Jakuskas, "Vibration Analysis of a Piping System Attached With Pumps and Subjected to Resonance", *International Journal of Emerging Technology and Advanced Engineering.*

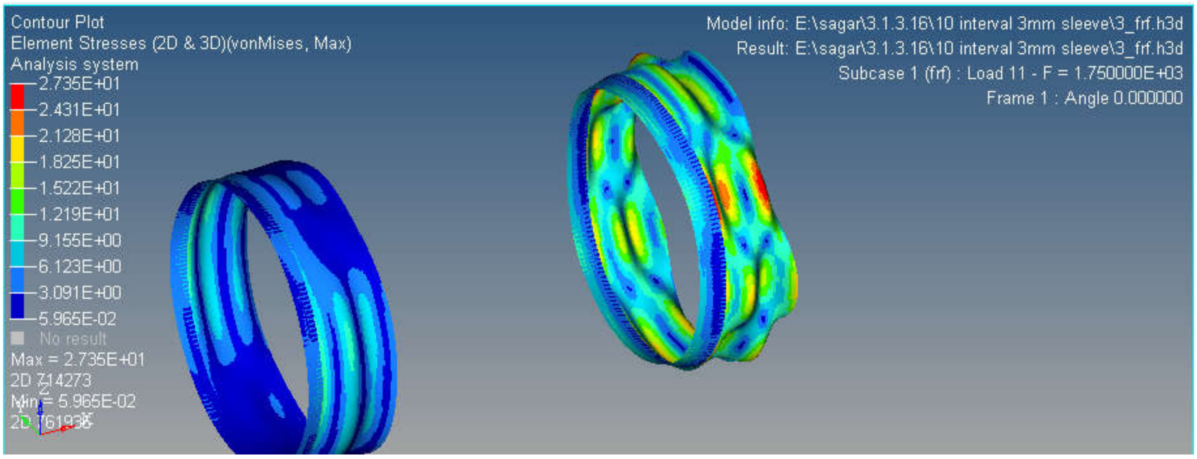
Wachai J.C. and J.D. Tison, 1994. "Vibrations in reciprocating machinery and piping systems", Texas University.

APPENDIX

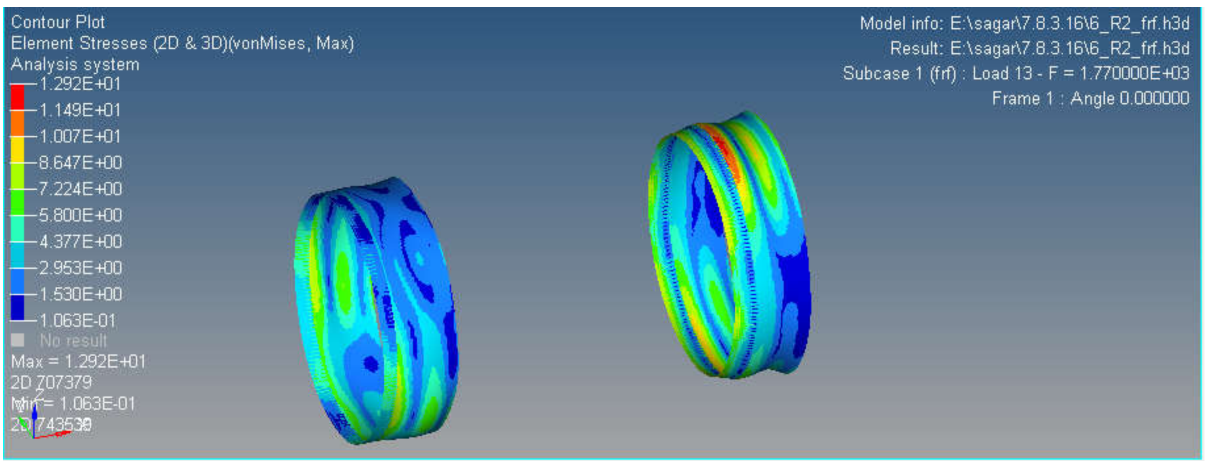
RGC-2 FRF result images



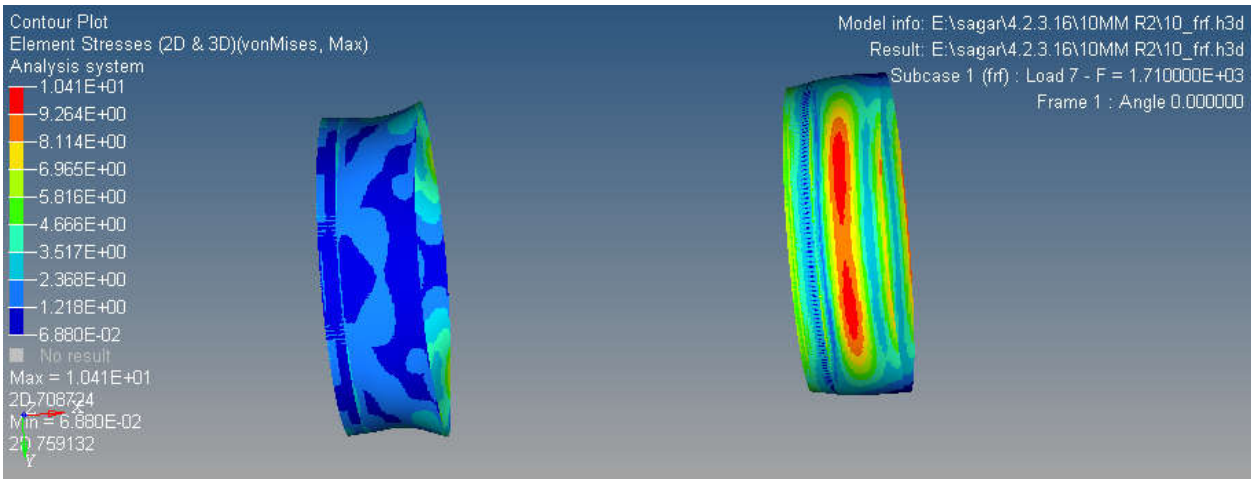
RGC-2 sleeve thickness 1.6mm



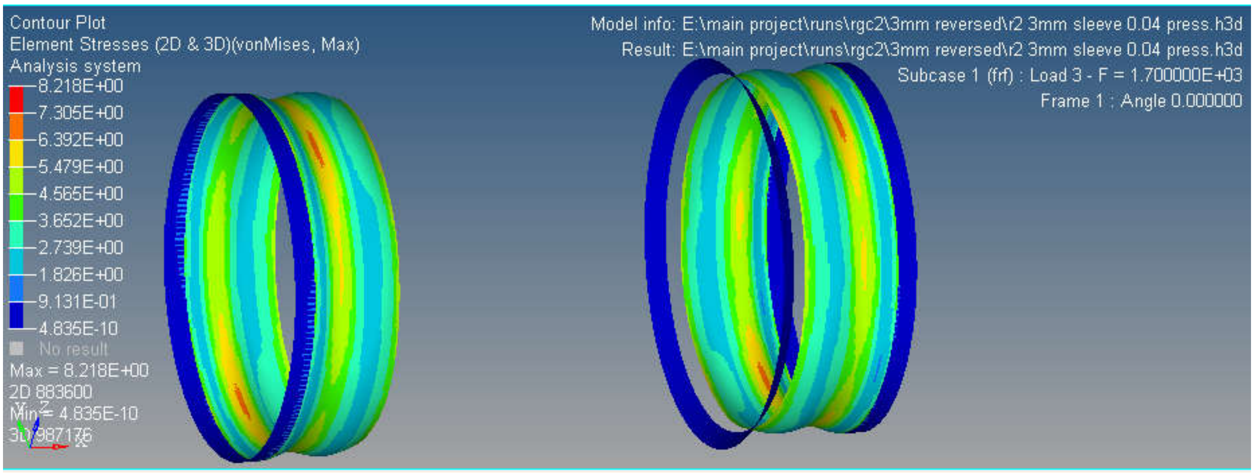
RGC-2 sleeve thickness 3mm



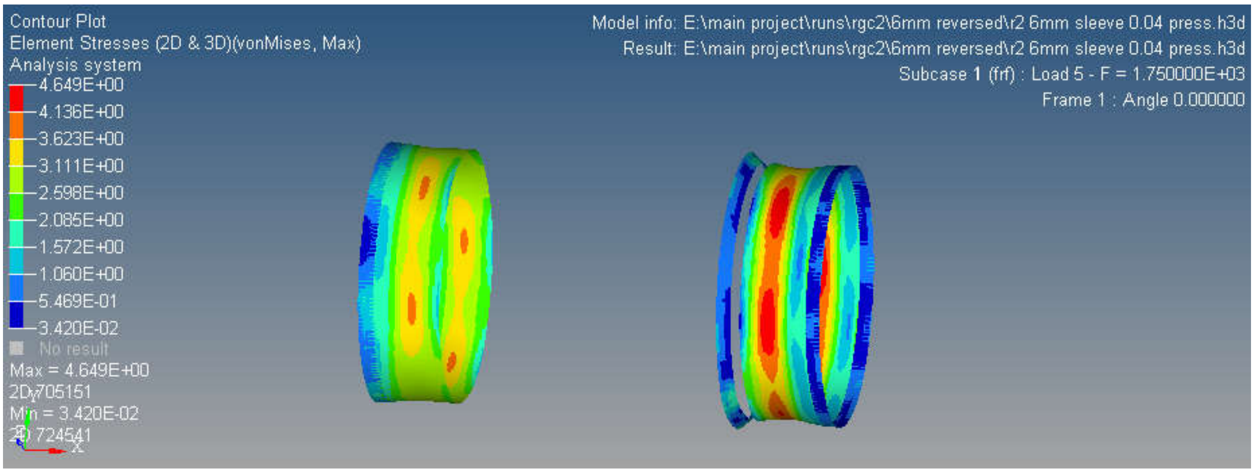
RGC-2 sleeve thickness 6mm



RGC-2 sleeve thickness 10mm



RGC-2 with 3mm sleeve location reversed and baffle



RGC-2 with 6mm sleeve location reversed and baffle

****