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"Fatigue Life Estimation of Pressure Reducing Valve Diaphragm"

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***Abstract*— Predicting the fatigue life of component exactly under the operating conditions is a challenging task in design engineering. In this work, fatigue life of pressure reducing valve diaphragm has been predicted which works under steam pressure. The fatigue life is predicted analytically by Goodman diagram using stress values given by different approaches suggested by M. Di Giovanni, Roark's, Timoshenko and Nadai. The stress and deflection values given by different analytical approaches have shown good agreement with Finite Element Analysis (FEA) results. Finally experimental fatigue testing for fatigue life estimation of the pressure reducing valve diaphragm has been done for its maximum stroke.**

***Index Terms*— Pressure reducing valve, rigid center, endurance limit, fatigue life.**

I. INTRODUCTION

Fatigue design is one of the observed modes of mechanical failure in practice. For this reason, fatigue becomes an obvious design consideration for many structures, such as aircraft, bridges, railroad cars, automotive suspensions and vehicle frames. In practice, accurate estimates of fatigue life are difficult to obtain as small changes in the loading and manufacturing conditions may strongly affect fatigue life. M. Di Giovanni [1] has presented all information which is necessary for complete design and analysis the performance of flat and corrugated diaphragms. In this handbook, they presented detailed information with respect to the materials for diaphragms, design procedure for flat and corrugated diaphragms, flat and corrugated diaphragms with rigid center, analytical solution for frequency response of flat and corrugated diaphragms and design of corrugated diaphragms with and without rigid center. This handbook also provides stress and deflections values for various diaphragm cases which are useful for fatigue life calculation. Richard C. Rice [2] has given elements of fatigue design process and how those elements must be tied together in a comprehensive product evaluation. The handbook covered various current technologies and procedures of importance in a fatigue design evaluation. The handbook is mainly focused on fatigue design considerations, effect of processing on fatigue performance, numerical analysis method and structural life evaluation. Piyush Gohil et al. [3] developed fatigue analysis test set up for predicting the life of sheet metal components.

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Authors demonstrated that the prediction of fatigue life for a sheet metal component has reasonable accuracy and efficiency by utilizing commercially available software's like ANSYS and Fe-Safe software were utilized as the finite element and fatigue life prediction solvers, respectively. Presented methodology of experimentally investigating the fatigue life of IS 2062 MS steel sheet metal component under constant amplitude loading. Albert E. et al. [4] described a fixture for making normal pressure tests of flat plates 5 inch in diameter in which particular care was taken to obtain rigid clamping at the edges. The centre deflection and extreme fiber stresses at low pressure and at high pressure were compared with the theoretical values. The empirical equations given were useful for finding the deflection values and stress values considering large deflection of plates. S. Timoshenko [5] gave deflection and stresses values for three different cases as like thin plates with small deflection, thin plates with large deflection and for thick plates. In case of large deflections author distinguished between immovable edges and edges free to move in the plane of the plate. This book also gave approximate formulas for uniformly loaded circular plates with large deflections and exact solution for a uniformly loaded circular plate with a clamped edge. Warren C. Y. and Richard G. B. [6] gave different diaphragm stresses or direct stresses for thin plates undergoing large deflection. Author gave various stress and deflection formulas for circular plates under distributed load producing large deflections for centre and edge conditions when the plates clamped at periphery and also for different boundary conditions. Shigley et al. [7] gave information regarding fatigue failure resulting from static and variable loading. Authors discussed different failure theories for the static loading. For variable loading condition authors have given different fatigue life methods like stress life method, strain life method, linear elastic fracture mechanics method. This book also gives endurance limit factors which are needed for fatigue life calculation. The book summarizes various factors which are important for fatigue failure like stress concentration and notch sensitivity, how to characterize the fluctuating stresses, fatigue failure criteria for fluctuating stress. Mahesh L. Raotole et al. [8] gave method of finding the fatigue life of crankshaft under complex loading conditions. Due to the repeated bending and twisting, crankshaft fails, as cracks form in fillet area. Mainly authors have given how to estimate the life of crankshaft using finite element method. In this work dynamic load analysis, FEM and stress analysis and finally prediction of fatigue life for crankshaft was performed, while doing this, authors focused upon accurate prediction of fatigue life to insure safety of components and its reliability.

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They have used the S-N approach to predict the fatigue life of crankshaft. R. Tang and F. Erdogan [9] presented general problem of a rectangular plate clamped along two parallel sides and containing a crack parallel to the clamps is considered. The problem is formulated in terms of a system of singular integral equations and the asymptotic behavior of the stress state near the corners is investigated. Numerical examples are considered for a clamped plate without a crack and with a centrally located crack, and the stresses along the clamp are calculated.

II. SIGNIFICANT PARAMETERS ESTIMATION

The austenitic alloys are most corrosion resistant of all the stainless steels. However they cannot be heat treated to improve their stress. Additional stress from the annealed condition can be obtained only by cold working. The basic alloy is type 304 possessing the minimum of 18% of chromium and 8% nickel combined with maximum of 0.08% of carbon. It is nonmagnetic as are all stainless steels.

- Chemical composition

Table 1. Chemical composition of type 304 is as follows:

Element	C	Mg	P	S	Si	Cr	Ni
%	0.08 max	2 max	0.04 max	0.03 max	1.0 max	18 to 20	8 to 10.5

Table 2. Physical constants for type 304

Specific gravity	8.03
Density	8027 Kg/m^3
Specific heat	0.50 KJ/Kg/K

The mechanical properties of stainless steel type 304 are as given below:

Modulus of elasticity (E) = 1.93 X 10^6 N/mm^2 in tension

Modulus of rigidity (G) = 8.62 X 10^4 N/mm^2 in tension

2.1 Behavior of diaphragm under lateral pressure

The pressure - deflection relationship for a pressure loaded diaphragm is linear for only small deflections. When diaphragm rigidly clamped at the edge is deflected even 10 % of its thickness, tensile stresses begin to appear. As the load continues to increase, the deflection increases at a slower rate and load - deflection relationship becomes nonlinear.

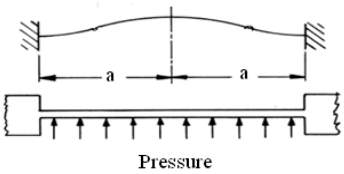

Figure 1. Deflection of rigidly clamped diaphragm under uniform pressure [1]

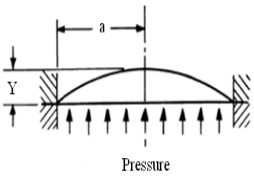

Figure 2. Shape of diaphragm under uniform pressure [1]

Depending upon size and overall dimensions there are 5 types of PRV such as 15NB, 20NB, 25NB, 40NB & 50NB. For study and analysis purpose here considered largest diaphragm which is used in 50NB PRV. The flat diaphragm used in PRV (50NB) having the overall dimensions and other parameters as in Table 3. On the basis of above mentioned data it is clear that the deflection- thickness ratio (Y/h) for existing diaphragm is higher than 5. So simple bending theory (small deflection) is not applicable here because, in case of simple bending theory diaphragm with small deflection (linear range) the ratio (Y/h) must be smaller than 5. In case of simple bending deflection of diaphragm occurs only due to bending stresses, there is no stress acting on middle plane of diaphragm.

Table 3. Parameters used for analysis of flat diaphragm

Material	Stainless steel (SS 304 ASTM -240)
Diameter	203.2 mm
Thickness (h)	0.25 mm
Max. stroke (lift) for diaphragm (Y)	5.0 mm (50 NB).
Max. operating pressure of PRV	17 Bar
Max. operating temperature of PRV	232° C

2.2 Analysis of flat diaphragm as a membrane

In existing flat diaphragm the ratio (Y/h) is greater than 5. So it is considered as membrane for load deflection analysis and used membrane theory for deflection and stress analysis. Again diaphragm consists of centrally placed lower diaphragm (L.D.) pad of 90 mm in diameter along with push rod at upper side of diaphragm. This L. D. pad acts as rigid center for flat diaphragm. So for deflection analysis of diaphragm effect of rigid center is considered. The characteristic equation for the flat diaphragm (membrane) without rigid center has given us the relationship between the pressure on the diaphragm and the corresponding deflection.

2.3 Flat diaphragm (membrane) with rigid center

There are many applications where it is necessary to provide a passive area in the centre of the diaphragm in the form of boss or rigid center. In our application also the lower diaphragm (L.D.) pad acts as rigid center, so it is necessary to consider the effect of L.D. Pad as a rigid center in the analysis of diaphragm for pressure Vs deflection relationship.

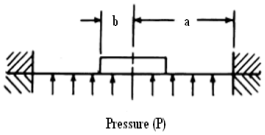

Figure 3. Flat diaphragm with rigid center [1]

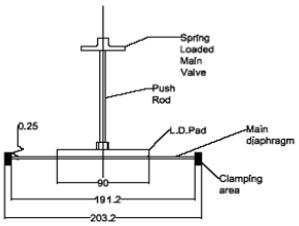

Figure 4. Arrangement of flat diaphragm with L. D. pad & push rod in PRV

The L.D. Pad as a rigid center is located on diaphragm as shown in Figure 3, Where,

a : Radius of diaphragm; b : Rigid center radius; (b/a) : Solidity ratio for flat diaphragm

To be effective the thickness of the rigid center should be at least 6 times the thickness of the diaphragm. The solidity ratio (b/a), it is the ratio of the rigid center radius and diaphragm radius; it is very important parameter in the performance characteristics of the diaphragm with rigid center [1].

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The arrangement of flat diaphragm with L. D. pad & push rod in PRV shown in Figure 4.

2.4 Characteristic equation of a membrane without rigid center

The main diaphragm in the PRV is mounted in between the lower diaphragm chamber and body of the valve. Pressure connection from the pilot valve is supplied to the bottom of diaphragm with the help of contol pipe assembly. The diaphragm gets deflected as per pressure signal coming from the pilot valve through control pipe assembly. This pressure vs deflection relationship in terms of characteristic equation has been given by Andreeva (1946).The characteristic equation of a membrane as developed by Andreeva (1946) is

$$\frac{Pa^4}{Eh^4} = \frac{7-\mu}{3(1-\mu)}\left(\frac{y_0}{h}\right)^3 \quad \text{..... (1)}$$

Where,

P = Pressure acting on diaphragm; E = Modulus of elasticity of material; h = Thickness of diaphragm; a= Radius of diaphragm; y_0= Centre deflection of diaphragm under pressure P.

From above Equation (1), it is clear that the resistance of the diaphragm to an external load increases with the cube of the deflection. There is nonlinear relationship between pressure and deflection [1].

2.5 Stress distribution for membrane without rigid center

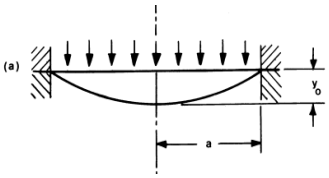

Figure 5. Shape of elastic surface

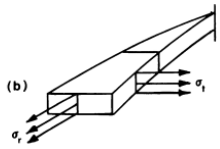

Figure 6. Membrane stresses σ_r & σ_t

When pressure applied on diaphragm which is clamped at periphery, the shape of elastic surface of a membrane under pressure is almost spherical in shape. M. Di Giovanni [1] has given radial stress and tangential stress equations for this clamped diaphragm at peripheries which are given as ,The radial stress at any point r from the centre of diaphragm can be given as,

$$\sigma_r = \frac{E{y_0}^2}{4a^2}\left(\frac{3-\mu}{1-\mu} - \frac{r^2}{a^2}\right) \quad \text{..... (2)}$$

The maximum stress occurs at the centre of diaphragm where r = 0 and it can be given by following equation,

$$\sigma_r = \frac{3-\mu}{1-\mu}\frac{E{y_0}^2}{4a^2} \quad \text{..... (3)}$$

The tangential stress σ_t can be given as,

$$\sigma_t = \frac{E{y_0}^2}{4a^2}\left(\frac{3-\mu}{1-\mu} - \frac{3r^2}{2}\right) \quad \text{..... (4)}$$

The maximum tangential stress is at the centre and is equal to the radial stress. It is important to note that the tensile stresses in a membrane increases with the square of the deflection.

Figure 5 shows, flexible diaphragm shape when it is undergone pressure. The shape of elastic surface after application of pressure is shown in Figure 5.The directions of radial stress and tangential stress are shown in Figure 6. The stresses σr & σt are taken as a membrane stresses because there is large deflection of diaphragm under applied pressure and for the diaphragm the deflection to its thickness ratio is greater than 5 [1]. Solving Equation 3 and 4 for different pressure values, it gives corresponding deflection and stress values. The handbook of M.Di Giovanni giving only the centre stress values but considering the failure location the edge stress values are more important. So for which study undergone through different handbooks as discussed in following sections, which are giving centre stress as well as edge stress values for flat diaphragm (membrane) without rigid center for the corresponding pressure.

2.6 Circular plates under distributed load producing large deflection

Roark's has given formulas for stress and deflection of flat plates for various loading and boundary conditions. These formulas should be used whenever the maximum deflection exceeds half the thickness as like in our case where deflection is much larger compare to the thickness of diaphragm. The following data gives the necessary parameters for the loadings and support conditions for calculation of stress and deflection values.

Let,

h = thickness of plate (mm); a = outer radius of plate (mm); P = unit lateral pressure (MPa); y_0= maximum deflection (mm); σ_b = bending stress (MPa); σ_d = diaphragm stress (MPa);

$\sigma = \sigma_b + \sigma_t$ = maximum stress due to flexure and diaphragm tension combined (MPa)

Then the applying the following formulas: First solving for y_0 in Equation 5 and calculate the stress values from Equation 6.

$$\frac{Pa^4}{Eh^4} = K_1\frac{y_0}{h} + K_2\left(\frac{y_0}{h}\right)^3 \quad \text{..... (5)}$$

$$\frac{\sigma a^2}{Eh^2} = K_3\frac{y_0}{h} + K_4\left(\frac{y_0}{h}\right)^2 \quad \text{..... (6)}$$

For case of fixed and held having uniform pressure q over entire plate constant K_1, K_2 are as follows,

$$K_1 = \frac{5.33}{1-\mu^2}\ ;\ K_2 = \frac{2.6}{1-\mu^2} \quad \text{..... (7)}$$

For getting centre stress value, constant values K_3 & K_4 are as follows,

$$K_3 = \frac{2}{1-\mu}\ ; K_4 = 0.976 \quad \text{..... (8)}$$

To calculate edge stress value, the constants K_3, K_4 are as follows,

$$K_3 = \frac{4}{1-\mu^2};\quad K_4 = 1.7 \quad \text{..... (9)}$$

Solving Equation 5 and 6 for different pressure values [6], it gives corresponding deflection and stress values.

2.7 Timoshenko handbook

The Timoshenko has given stress and deflection formulas for various loading cases and boundary conditions for large deflection of thin plates. The Timoshenko has also given approximate formulas for uniformly loaded circular plates with large deflection and exact solution for uniformly loaded circular plates with clamped edges. For lateral loading of plates Timoshenko has done considerable numerical calculation for getting the stress and deflection values. Considering our case i.e. uniformly loaded circular plates with clamped edges. Let, a circular plate of radius *a* be clamped at the edge and subjected to a uniformly distributed load (UDL) of intensity *q*. Timoshenko assumed that the shape of the deflected surface can be represented by the following Equation 10,

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$$y = y_0(1-\frac{r^2}{a^2})^2 \quad \text{..... (10)}$$

Where,

y = deflection value at distance; from centre of the plate (mm); y_0= centre deflection value (mm); r = radial distance from the centre of the plate (mm); a = outer radius of plate (mm); P= UDL intensity (MPa); σ_r= Radial stress (MPa) ; σ_t =Tangential stress (MPa)

For calculating stress and deflection value Timoshenko has given following formulas [5].

$$\frac{y_0}{h} + A(\frac{y_0}{h})^3 = B\frac{P}{E}(\frac{a}{h})^4 \quad \text{.....(11)}$$

$$\sigma_r = \alpha_r E(\frac{y_0}{a})^2 \quad \text{.....(12)}$$

$$\sigma_{r'} = \beta_r E\frac{y_0 h}{a^2} \quad \text{..... (13)}$$

$$\sigma_t = \alpha_t E(\frac{y_0}{a})^2 \quad \text{..... (14)}$$

$$\sigma_{t'} = \beta_t E\frac{y_0 h}{a^2} \quad \text{..... (15)}$$

Where,

σ_r, σ_t are stresses in middle plane and $\sigma_{r'}$, $\sigma_{t'}$ are extreme fiber bending stresses.α_r, α_t, β_r, β_t, A, B are constant values taken from Timoshenko handbook. The Timoshenko equations have given stresses in middle plane of the plate as well as extreme fiber stresses and square addition of them gives total maximum stress at the centre and edge location using different constants values given by Timoshenko handbook. Solving Equation 11, 12,13,14,15 for different pressure values, it gives corresponding deflection and stress values. Following the numerical results and calculations of Timoshenko, another researcher Nadai has given the approximate solution for uniformly loaded circular plates with clamped edges as follows.

2.8 Nadai approach

Nadai followed the equations of Timoshenko, taking the derivatives of Equation 10; Nadai developed his own equation of flexible membrane. The general equation of such membrane can be given by following Equation 16,

$$y_0 = 0.662\, a\sqrt[3]{\frac{Pa}{Eh}} \quad \text{..... (16)}$$

Above formula shows that, the deflections are not proportional to the intensity of the load but vary as the cube root of that intensity [5]. For the tensile stresses at the centre of the membrane and at the boundary, Nadai has given the following solutions, respectively,

$$(\sigma_r)_{r=0} = 0.423\sqrt[3]{\frac{EP^2a^2}{h^2}} \quad \text{.... (17)}$$

$$(\sigma_r)_{r=a} = 0.328\sqrt[3]{\frac{EP^2a^2}{h^2}} \quad \text{..... .(18)}$$

Where,

E = Modulus of elasticity of plate material (MPa); h = Thickness of diaphragm or plate (mm);

$(\sigma_r)_{r=0}$ = centre stress value (MPa); $(\sigma_r)_{r=a}$ = edge stress value (MPa). Solving Equation 16, Equation 17 and Equation 18 for different pressure values it gives corresponding deflection and stress values.

III. III. FINITE ELEMENT ANALYSIS OF FLAT DIAPHRAGM

3.1 Simulation of flat diaphragm

3.1.1 Model: The model of flat diaphragm shown in Figure 3.7 is drawn using workbench design modeler & imported the same model in ANSYS workbench simulation.

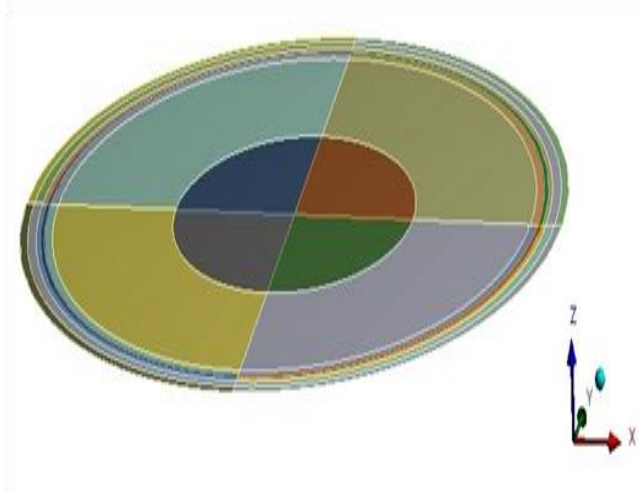

Figure 7. Flat diaphragm model

3.1.2 Mesh: Now, assigned stainless steel material to the model using material directory of ANSYS workbench, and then meshing has been done using mesh option. Element type SHELL 181 is suitable for analyzing thin to moderately-thick shell structures. It is a four-node element with six degrees of freedom at each node: translations in the x, y, and z directions, and rotations about the x, y, and z-axes. It is associated with linear elastic, elastoplastic, creep, or hyper elastic material properties. It is well-suited for linear, large rotation, large strain nonlinear applications. Change in shell thickness is accounted for in nonlinear analyses. It accounts for follower (load stiffness) effects of distributed pressures. The geometry, node locations, and the coordinate system for this element are shown in Figure 9.

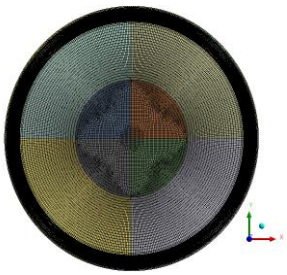

Figure 8. Mesh for flat diaphragm

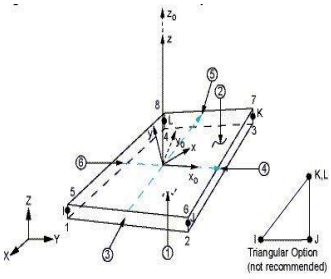

Figure 9. Shell 181 element

3.1.3 Constraints or Boundary Conditions:

The displacements at circular periphery of the geometry are specified as zero, since the diaphragm is clamped at its periphery. Then the pressure is given on the bottom surface of the plate. Once the boundary conditions are specified the model looks as shown in Figure 10.

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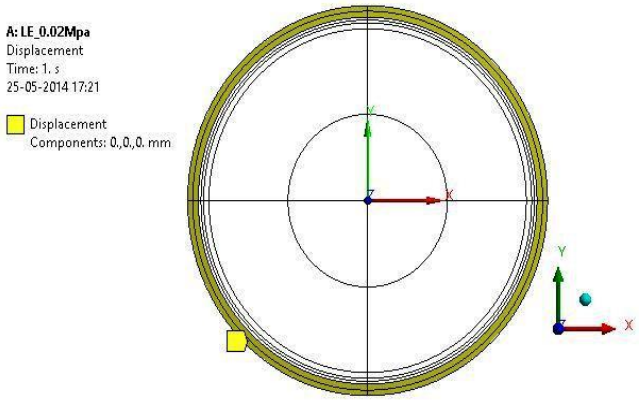

Figure 10. Application of boundary conditions (Clamping at periphery)

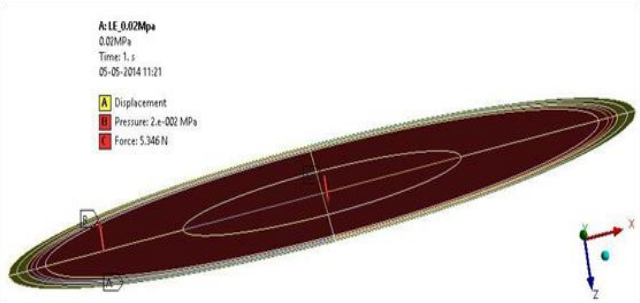

Figure 11. Application of pressure and rigid center weight

3.1.4 Loads:

After fixing the periphery next step is the application of pressure on the diaphragm from bottom side as shown in Figure 11. The L. D. pad & push rod on flat diaphragm is rigid center for our case as mentioned in earlier chapter. Now, the weight of this rigid center converted into force and applied on the flat diaphragm on 90 mm diameter which is opposing to the applied pressure.

3.1.5 Results:

Once the boundary conditions are specified, the model is selected and the solution option solves for all possible results. The result involves the maximum deformation, maximum centre stress, maximum edge stress of the flat circular diaphragm clamped at its periphery as shown in Figure 12 to Figure 17.

3.1.5.1 Flat diaphragm without rigid center results

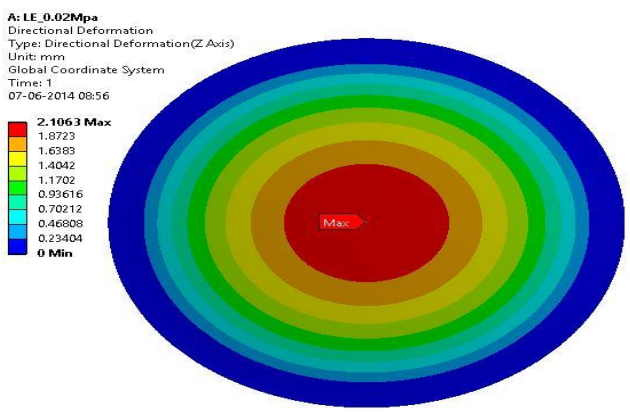

Figure 12. Deformation plot for flat diaphragm without rigid center

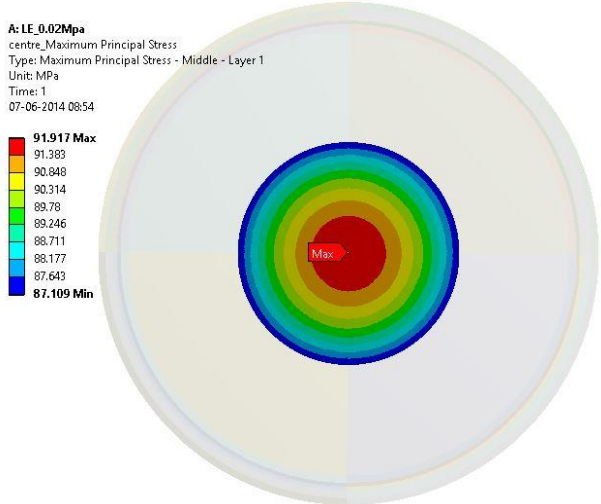

Figure 13. Centre stress plot for flat diaphragm without rigid center

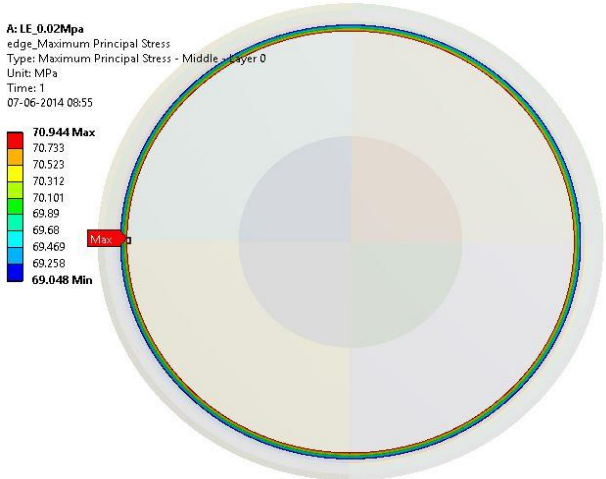

Figure 14. Edge stress plot for flat diaphragm without rigid center

Figure 12, Figure 13, Figure 14 shows, the deflection plot and stress plots obtained for flat diaphragm without rigid center with 0.2 bar as applied pressure. After changing the applied pressure values and solving the same problem, then ANSYS automatically gives the deflection plot and stress plot for that particular pressure.

3.1.5.2 Flat diaphragm with rigid center results

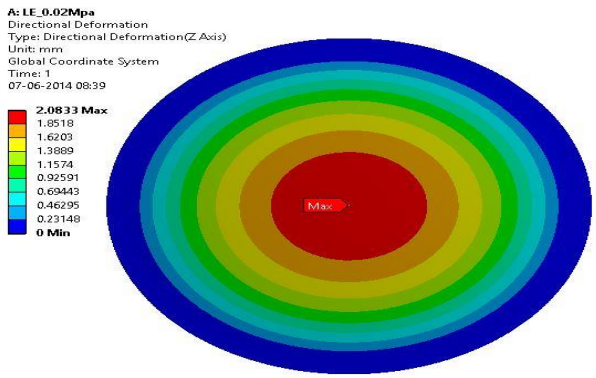

Figure 15. Deformation plot for flat diaphragm with rigid center

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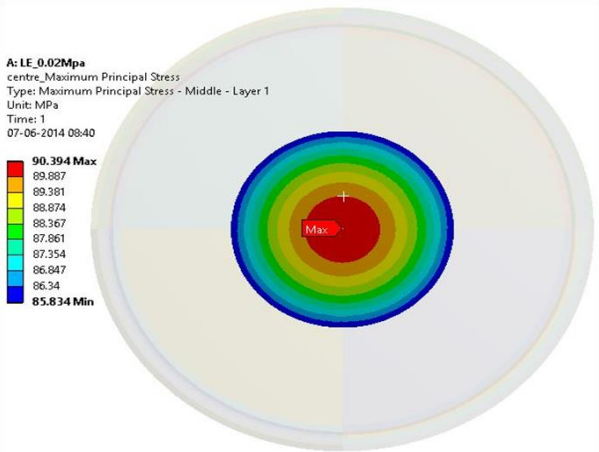

Figure 16. Centre stress plot for flat diaphragm with rigid center

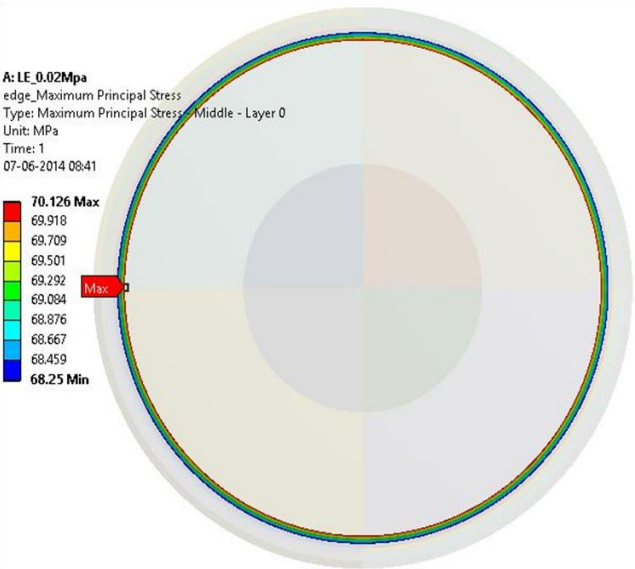

Figure 17. Edge stress plot for flat diaphragm with rigid center

Figure 15, Figure 16, Figure 17 shows, the deflection plot and stress plots obtained for flat diaphragm with rigid center with 0.2 bar as applied pressure. After changing the applied pressure values and solving the same problem, then ANSYS automatically gives the deflection plot and stress plots for that particular pressure. As the weight of the rigid center (545 gram) is much smaller compared to pressure applied from bottom side, it is observed that there is only small amount of reduction occurred for deflection value, centre stress value and edge stress value. As the analytical equations are not available for rigid center on flat diaphragm, FEA simulation results are taken for fatigue life estimation.

IV. SUMMARY

4.1 Deflection values

Conclusion: All the handbooks and FEA results giving the nearly same value of centre deflection for different pressure conditions. It observed that, for all the approaches and FEA results in between 2 bar to 2.5 bar, the diaphragm is giving its maximum stroke or lift of 5 mm. The deflection values given by M.Di Giovanni and Nadai approach are much closer to FEA values compared to all other approaches. The results are shown graphically in the Figure 18.

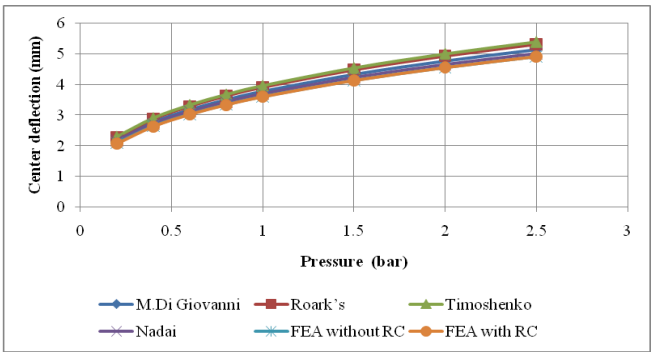

Figure 18. Comparison of results of centre deflection with analytical & FEA method

4.2 Centre stress values

Conclusion: All the handbooks giving the nearly the same value of centre stress for different pressure conditions. The stress values given by Roark's handbook are higher compare to other handbooks. The stress values given by M.Di Giovanni and Nadai are much closer to FEA values compared to all other approaches. The results are shown graphically in the Figure 19.

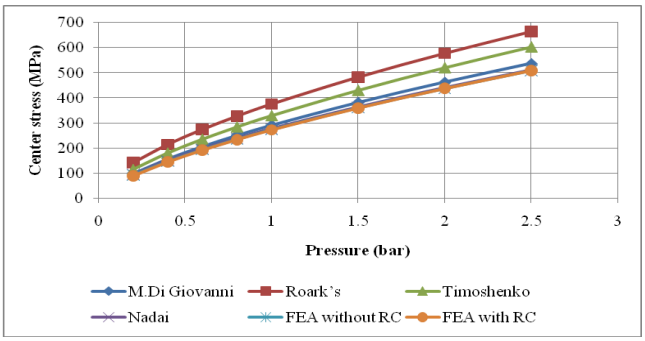

Figure 19. Comparison of results of centre stress with analytical & FEA method

4.3 Edge stress values

Conclusion: The stress values given by Timoshenko and Nadai are closer to each other but they are much lower than Roark's handbook. Because for the Roark's the edges stress value is deflection dependant while for Nadai and Timoshenko approach edge stress value is pressure dependant. Timoshenko and Nadai have given the stresses considering nonlinearity coming into the material but Roark's has given only linear stress value. The results are shown graphically in the Figure 20.

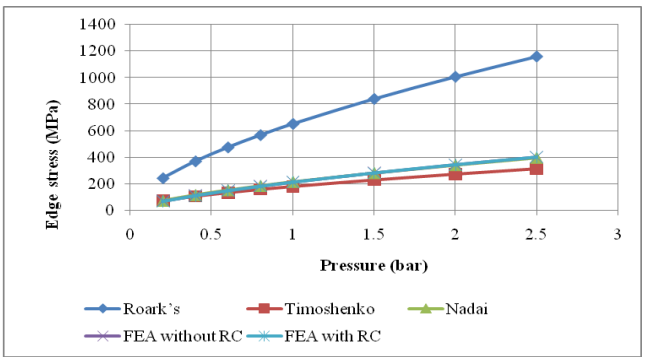

Figure 20. Comparison of results of edge stress with analytical & FEA method

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V. FATIGUE LIFE ESTIMATION

5.1 Endurance limit modifying factors calculation

The fatigue or endurance limit of material is the maximum amplitude of completely reversed stress that the standard specimen can sustain for an unlimited number of cycles without fatigue failure and the fatigue life is the number of stress cycles that standard specimen can complete during the test before appearance of first fatigue crack. Since fatigue test cannot be conducted for unlimited number of cycles, 10^6cycle is considered as sufficient number to define the endurance limit. The rotating-beam specimen used in the laboratory to determine endurance limits is prepared very carefully and tested under closely controlled conditions. It is unrealistic to expect the endurance limit of a mechanical or structural member to match the values obtained in the laboratory. Galling Marin identified factors which considers the effects of surface condition, size, loading, temperature and miscellaneous items. A Marin given the Equation 19 as follows,

$$S_e = k_a k_b k_c k_d k_e S_e' \quad \text{..... (19)}$$

When endurance tests of parts are not available, estimations are made by applying Marin factors to the endurance limit. The stainless steel diaphragm which is clamped at periphery and undergoes pressure cycles, for calculating its fatigue life it is needed to apply here the Marin equation for corresponding operating conditions. When endurance tests of parts are not available, estimations are made by applying Marin factors to the endurance limit. The stainless steel diaphragm which is clamped at periphery and undergoes pressure cycles, for calculating its fatigue life it is needed to apply here the Marin equation for corresponding operating conditions.

Table 4. Endurance limit factors for fatigue life calculation

k_a	k_b	k_c	k_d	k_e	S_e'= 0.5 X Sut	S_e = (MPa)
0.69	0.77	1	0.89	0.62	575	168.574

5.2 Fatigue life calculation

As the diaphragm of PRV undergoing higher number of cycles, for finite life design of diaphragm, used modified Goodman diagram for calculation and based on values of Goodman diagram SN curve is generated this is useful for fatigue life calculation. Goodman diagram is chosen for fatigue stress calculation instead of using Soderberg and Gerber, because Goodman line is safe from design considerations because it is completely inside the failure points. Goodman equation of straight line is simple compared with parabolic equations given by Gerber [7].

5.2.1 Fatigue stress calculation

Table 5. Pressure Vs Centre stress Vs Fatigue stress with analytical & FEA method

Press-ure (bar)	Centre Stress (MPa)			Fatigue stress (MPa)		
	Without RC		With RC	Without RC		With RC
	Analytical	FEA	FEA	Analytical	FEA	FEA
2.5	511.09	510.42	509.78	328.55	328	327.47

Table 6. Pressure Vs Edge stress Vs Fatigue stress with analytical & FEA method

Press-ure (bar)	Edge Stress (MPa)			Fatigue stress (MPa)		
	Without RC		With RC	Without RC		With RC
	Analytical	FEA	FEA	Analytical	FEA	FEA
2.5	396.3	401.74	401.4	239.39	243.38	243.13

Fatigue life of diaphragm for corresponding pressure calculated by fatigue stress given by Goodman diaphragm. It is observed that for lower pressure there is higher number of cycles of diaphragm.

Table 7. Fatigue life for corresponding Fatigue stress with analytical & FEA method for centre stress values

Pressure (bar)		2.5
Without RC (analytical)	M.Di Giovanni	6.15X 10^4
	Roark's	2.07 X 10^4
	Timoshenko	3.41 X 10^4
	Nadai	7.89 X 10^4
Without RC (FEA)		7.94 X 10^4
With RC (FEA)		7.99 X 10^4

Table 8. Fatigue life for corresponding Fatigue stress with analytical & FEA method for edge stress values

Pressure (bar)		2.5
Without RC (analytical)	Timoshenko	7.10 X 10^5
	Nadai	2.63 X 10^5
Without RC (FEA)		2.47 X 10^5
With RC (FEA)		2.48 X 10^5

5.3. Experimental set up

Figure 21 shows ,experimental set up for fatigue testing of flat diaphragm of diameter 203.2 mm ,which is clamped by bolting arrangement in a pressure reducing valve (PRV) having size of 50 NB .

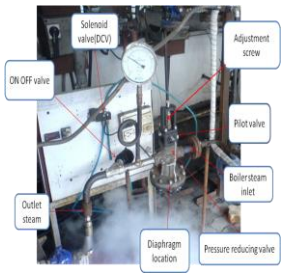

Figure 21.Experimental set up for fatigue testing of diaphragm

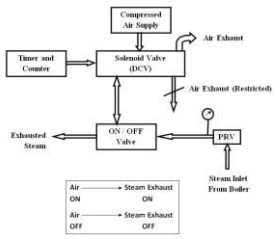

Figure 22. Block diaphragm for experimental analysis

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Table 9 shows, comparison of number of fatigue failure cycles obtained by analytical, FEA and actual testing for 2.5 bar steam pressure.

Table 9. Number of cycles with analytical, FEA and fatigue testing of diaphragm

Pressure (bar)			2.5
Number of cycles	Without RC	Analytical	2,63,000
		FEA	2,47,000
	With RC	FEA	2,48,000
	Plate 1	Testing with RC	2,20,000
	Plate 2	Testing with RC	2,24,000

5.3.1 Failure images of diaphragm

Following Figure 23 and Figure 24 shows, observed cracks at edge location nearer to clamping region after performing the fatigue test.

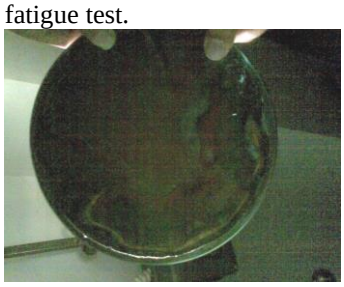

Figure 23. Cracks on diaphragm at edge location (test plate 1)

Figure 24. Cracks on diaphragm at edge location (test plate 2)

VI. RESULTS AND DISCUSSIONS

Fatigue testing of flat diaphragm was conducted for 2.5 bar steam pressure. The number of failure cycles are compared with analytical and FEA results.

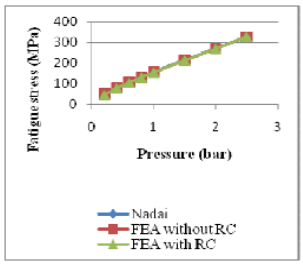

Figure 25. Pressure Vs Fatigue stress at centre ion of plate

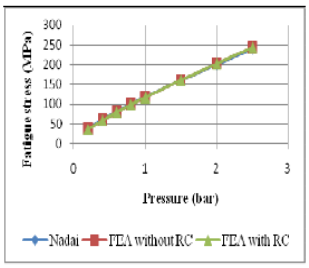

Figure 26. Pressure Vs Fatigue stress at edge location of plate

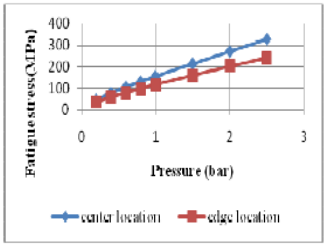

Figure 27. Pressure Vs Fatigue stress comparison at centre and edge location of plate

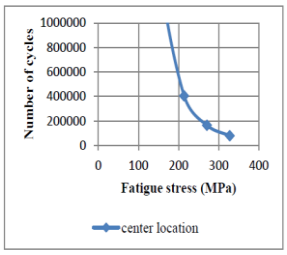

Figure 28. Fatigue stress Vs Number of cycles at centre location of plate

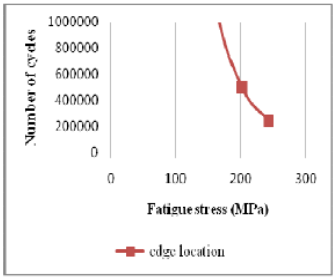

Figure 29. Fatigue stress Vs Number of cycles at edge location of plate

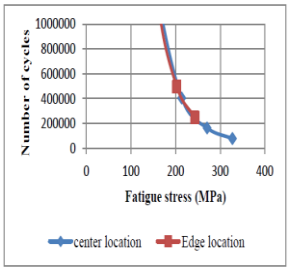

Figure 30. Fatigue stress Vs Number of cycles comparison at centre and edge location of plate

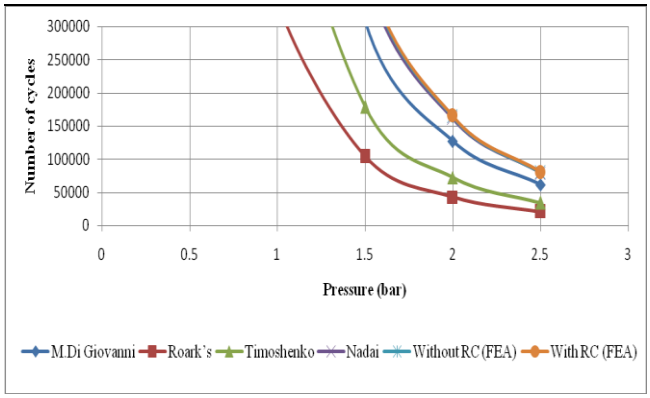

Figure 31. Pressure Vs Number of cycles at centre location of plate

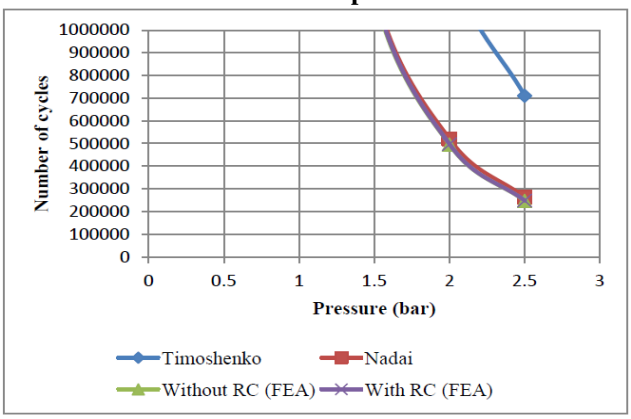

Figure 32. Pressure Vs Number of cycles at edge location of plate

VII. CONCLUSION AND FUTURE SCOPE

7.1 Discussion and conclusions

The project is aimed at estimation of fatigue life of pressure reducing valve diaphragm analytically and experimentally for required pressure inlet and outlet conditions.This chapter summarizes the main conclusions from this work.

- Theoretical pressure deflection relationship for flat diaphragm is established through various approaches suggested by M.Di Giovanni, Roark's, Timoshenko and Nadai.
- Finite element analysis of flat diaphragm for pressure deflection relationship done, results show good agreement with analytical solution.

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- Theoretical pressure stress relationship for flat diaphragm is established through various approaches suggested by M.Di Giovanni, Roark's, Timoshenko and Nadai.
- Finite element analysis of flat diaphragm for pressure stress relationship done, results show good agreement with analytical solution.
- In fatigue life evaluation as go on increasing the reliability, reliability factor K_e decreases, endurance limit of material decreases so number of fatigue failure cycles also decreases.
- In fatigue life evaluation as go on changing the surface finish of the material from ground, machine rolled, hot rolled to as forged endurance limit of material decreases so number of fatigue failure cycles also decreases.
- As the operating temperature increases the temperature modification factor decreases, endurance limit of material decreases so number of fatigue failure cycles also decreases.

7.2 Future scope

In this study the pressure deflection analysis and pressure stress analysis of flat diaphragm has been done. Deflection and stresses in diaphragms are compared by analytical & FEA method. Further future study can be done by changing the material of the flat diaphragm. It is also suggested to check the performance of flat diaphragm by changing the thickness of sheet.The material which is having higher ultimate tensile strength and corrosion resistance can be used as diaphragm material which can increase the fatigue life and performance of pressure reducing valve.

REFERENCES

[1] M. Di Giovanni, "Flat and Corrugated Diaphragm Design Handbook", 1982, Marcel Dekker Inc., New York.
[2] Richard C.Rice, Society of Automotive Engineers Fatigue, "SAE Fatigue Design Handbook", 3rd Edition, (1997).
[3] Dr. Piyush Gohil, Hemant N. Panchal, Siddiqi Mahmud Sohail, Devang V. Mahant, "Experimental and FEA Prediction of Fatigue Life in Sheet Metal (IS 2062)", International Journal of Applied Research & Studies.
[4] Albert E. Macpherson, Walter ramberg and Samuel levy, "Normal Pressure Tests of Circular Plates with Clamped Edges", Report No. 744, National Advisory Committee for Aeronautics.
[5] S. Timoshenko, S. Woinowsky Krieger, "Theory of Plates and Shells", 1959, Second Edition, McGraw-Hill Book Company, pp. 403-404.
[6] Warren C. Y. and Richard G. B, "Roark's Formulas for Stress and Strain", Seventh Edition, McGraw-Hill Book Company, pp. 448-449.
[7] J.E.Shigley, Charles R.mischke, S. krishnamurthy "Shigley's Mechanical Engineering Design", Eighth Edition, Mechanical Engineering, McGraw-Hill Book Company, pp.278.
[8] Mahesh L. Raotole, Prof. D. B. Sadaphale , Prof. J. R.Chaudhari, " Prediction of Fatigue Life of Crank Shaft using S-N Approach" , International Journal of Emerging Technology and Advanced Engineering, Volume 3, February 2013.
[9] R. Tang and F. Erdogan, "Clamped Rectangular Plate Containing Crack", Theoretical and Applied Fracture Mechanics, Volume 4, Issue 3, December (1985), pp. 233-243.

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