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## Evaluating the effect of pretension on the fatigue life of double lap bolted joint by analytical & numerical (FEA)

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### Abstract

The main objective of this work is to improve the fatigue life of double lap bolted joints used in aircraft applications. The fatigue life performance of a bolted joint is dependent on various factors, such as size of the bolt, the number and position of bolts used, the level of preload or tightening torque applied to the bolt, the material plate thickness, coefficient of friction and surface roughness. Among these factors, bolt pretension has been studied. To investigate this an aircraft grade Aluminium alloy 7068 with double lap bolted joints are used. The effect of the bolt pretension is studied and analyzed, both analytical and numerical methods are used to investigate the fatigue behaviour of bolted joint. The numbers of cycles calculated analytically and has been compared with numerical results obtained by FEA. From the result it is seen that fatigue life of double lap bolted joint increased due to compressive stresses induced on the application of bolt pretension. These compressive stresses decrease the stress concentration around discontinuity.

**Keywords:** Bolted joints, fatigue life, stress concentration, Pretension, Double lap bolted joint.

### 1. Introduction

Design of structures and the optimum selection of materials, in industries such as the aerospace industry have always been considered of critical importance if they are to be efficient and safe, and thus resistant to the effects of dynamic loads as occurs during flights. Aircrafts are continually subjected to stresses and harsh conditions, leading to cracking due to corrosion and fatiguing. Mechanical joints, especially bolted joints are most important components in aircraft structures. Aircraft structures are still largely constructed from high strength aluminium alloys, because the low density provides an optimum strength-to-weight ratio material. Connecting structural members is still widely achieved by means of bolted joints. Most of the machine parts bound to dynamic load due to load cycles, inertia effects, reactive forces, vibrations, eccentricity. These dynamic load develops fatigue in aircraft part, structure and ultimately in bolted joint.

Initiation Site	Susceptible to Failure
Bolt, Stud or Screw	108
Fastener hole or other hole	72
Fillet, radius or sharp notch	57
Weld	53
Corrosion	43
Manufacturing defect	27
Scratch, nick or dent	26
Surface or subsurface flaw	6
Improper heat treatment	4
Maintenance-induced crack	2
Wear	2

Table 1 Fatigue initiation sites observed in aircraft

Table 1 highlighted that the study of fatigue behaviour of bolted joints is important. To improve the fatigue performance of component by introducing favourable compressive stresses. This is achieved by bolt pretension in bolted joint.

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B. A. T.N.Chakherlou [1] has presented numerical simulation and experimental results showed that the fatigue life of bolted plates improve because of the compressive stresses created around the plate hole due to clamping force. The life improvement is greatest at the high cycle fatigue life region of the S-N curves. H. T. M. M. A. A. T.N. Chakherlou [2] his study incorporates both of the crack initiation and crack propagation concepts while differentiating between them. The results demonstrate that interference fit has positive influence on improving the fatigue life of both fatigue crack initiation and fatigue crack growth stages. However, the estimation proves that fatigue life improvement at the fatigue crack growth stage is more prominent than at the fatigue crack initiation stage particularly in low load levels. J. V. Jose Maria Minguez [3] in this paper demonstrates how both pretensioning the bolts and the material thickness influence the fatigue life of double lap joints. The tightening torque applied to the bolts results in a compression of the joint plate members, which causes friction between them, and so prevents their relative slipping. Thus the bearing of the bolt against the hole edges is avoided, and load transmitted through the joint by friction. So the load is distributed over a larger area around the hole, the SCF is becomes negligible and so the fatigue performance of the joint is much improved. M. D. A. N. V. D. Crococo [4] have presented every maintenance operation (loosening and tightening) provides a loss in preloading force, which is particularly evident in the presence of dry conditions: wear pattern indicates that coating is progressively peeled off. Tightening torque should be increased with the number of tightening in order to guarantee the same preloading force and coupling pressure between the parts assembled by means of the clamp.

J. C. B. L. T.-G. K. Hong-Chul Lee [5] has presented experimental and finite element results which shows that the cold expanded plates have longer fatigue life compared to as drilled plates in double shear lap joints and the life enhancement is more for lower alternating longitudinal loads. U.A. Khashaba, H.E.M. Sallam [6] has presented the bolt bearing strength increased as the tightening torque increased. The results in this figure indicate that, in the range of the investigated tightening torques, the bolt bearing strength increases with increasing the tightening torque. A. Benhamena [7] said that the wear mechanism and contact surfaces degradation depends on the magnitude of tightening torque. The adhesive wear dominates at tightening torque and fretting fatigue cyclic number lower while the abrasive wear dominates for tightening torque and fretting fatigue cyclic number higher. The size of the adhesion and slip zones on contact zone is related at the magnitude of tightening torque. This means that the wear mechanism is related to the torque value. In other words, the increasing of torque leads to an increasing of the frictional stresses and to reduce the relative slip at the interface of bolted assembly. Babak Abazadeh, T.N. Chakherlou [8] says that bolt clamping and interference fitting of a plate containing a hole in a joint create pre-stresses around the hole. In addition, the clamped mating surfaces of the specimen (plate) and connector parts introduce resistant shear stress in the double shear lap joint. N. Eliaz, G. Gheorghiu [9] represents that the bolts in the helicopter main rotor drive-plate assembly failed in overload mechanism. The main objective of this project is to investigate the effect of bolt torque tightening, thus achieving different levels of joint preloading. When the bolt is not torque tightened there is no load to cause compression of the

members as the plates press together. The joint members can freely move over each other and load is transferred from bolt to the plates. So the stress concentration increases at the hole & it may fail. Now if the certain amount of torque is applied on the bolt, friction occurs at the contact surfaces. So the slipping of members can be avoided. Then the load will be transmitted through the joint directly from the member to the plate members by friction, instead of transferred through the bolt to the plate members hole. In this way stress concentration is minimized. Practically it is not possible to measure amount of bolt tension in a joint in service. To ensure that a desired preload has been achieved with a bolt, it is more practical to use a torque wrench to apply the load to the bolt through the nut.

## 2. Materials & Methods

First all the analysis is done on plain specimen of BSL165, with a centrally drilled hole, because these have a well defined stress concentration factor. These specimens will be referred to as plain specimens and were used for bench mark comparison purposes.

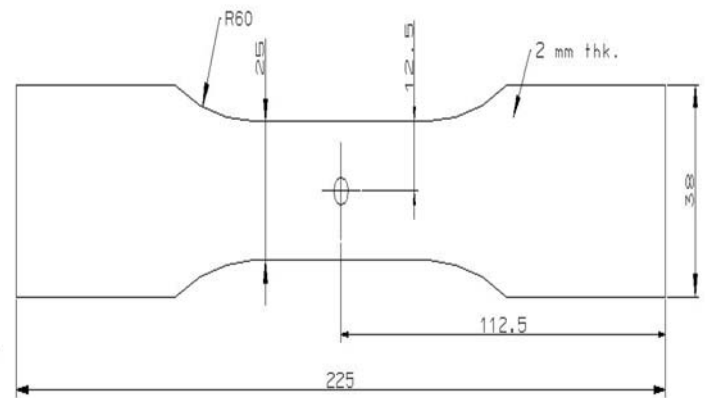


Fig 1. Plain Specimen as per standard (ASTM E466-82)

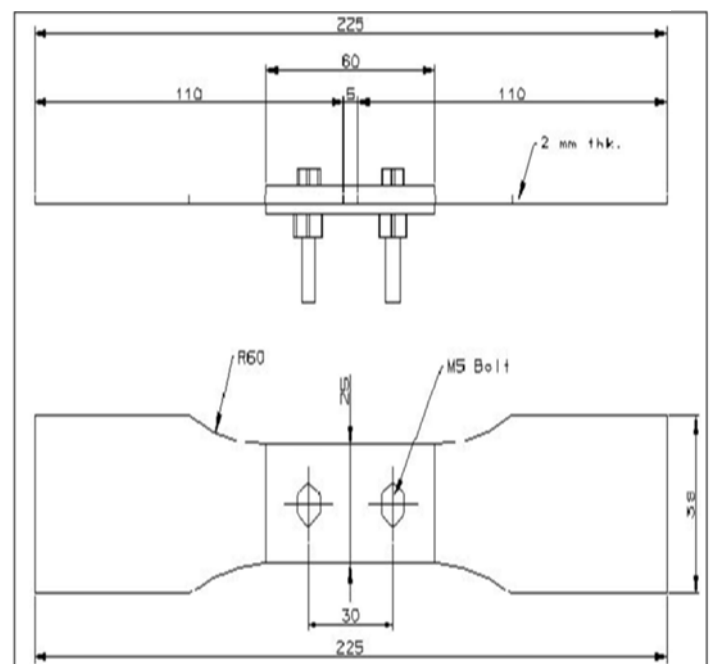


Fig 2. Double lap joint Specimen

The joints constructed using M5 set screws, washers and aluminium ny-lock nuts. The ny-lock nuts used to prevent relaxation of preload during fatigue testing. During joint assembly the nuts are run down to 1 mm above the washer using a spanner, before the final run-down torque is applied. Finally, the required torque loading is achieved using a torque wrench. This is carried out for each specimen to ensure similar torque loading on the nuts is achieved. The fatigue life of plain specimen & double lap bolted joint has been calculated using two methods.

1. Analytical method using S-N curve
2. Numerical method using Ansys.

The above two methods are elaborated below.

**1. Analytical method using S-N curve (Plain specimen)**

Endurance limit of actual component,

$$Se = Ka \times Kb \times Kc \times Kd \times Se'$$

$$Ka = 0.6825$$

$$Kb = 1 \quad (\text{for axial loading})$$

$$Kc = 0.868 \quad (\text{for 95\% Reliability})$$

$$Kd = 0.47 \quad (\text{from charts in } q=0.75, Kt=0.2)$$

Endurance limit of standard specimen (bending test specimen)

$$Se' = 0.85 \times 0.3 \times 483 \quad (S_{ut} = 483 \text{ MPa for 2014-T6})$$

$$Se' = 123.2 \text{ MPa}$$

Test 1- Tensile Load on specimen 8200N and 200 N

Corresponding stresses are,

$$\sigma_{max} = \frac{F}{(w-d) \times t} = \frac{8200}{40} = 205 \text{ MPa}$$

$$\sigma_{min} = \frac{F}{(w-d) \times t} = \frac{200}{40} = 5 \text{ MPa}$$

$$\sigma_{mean} = \frac{\sigma_{max} + \sigma_{min}}{2} = 105 \text{ MPa}$$

$$\sigma_{amp} = \frac{\sigma_{max} - \sigma_{min}}{2} = 100 \text{ MPa}$$

Endurance limit for specimen,

$$Se = Ka \times Kb \times Kc \times Kd \times Se'$$

$$Se = 0.6825 \times 1 \times 0.868 \times 0.47 \times 123.2$$

$$Se = 34.34 \text{ MPa}$$

$$Sf = S_{ut} \times \left( \frac{\sigma_a}{S_{ut} - \sigma_m} \right)$$

$$Sf = 127.78 \text{ MPa}$$

By using S-N curve

$$a = \log_{10}(0.9 \times S_{ut}) = 2.6382$$

$$b = \log_{10}(Sf) = \log_{10}(127.78) = 2.1065$$

$$d = \log_{10}(Se) = \log_{10}(34.34) = 1.5358$$

No. of cycles can be calculated by using below formula,

$$(a-d)/(b-d) = 3/(6-N)$$

$$N = 4.4469$$

$$\text{No. of cycles} = 10^N = 27983.37$$

Similarly other results with different loads were calculated and given in table no.2.

P <sub>max</sub>	P <sub>min</sub>	Cycles
8200	200	27983
8700	200	22686
11000	400	9797

Table 2 Analytical Results of Plain specimen

**1. Analytical method using S-N curve (Double lap joint)**

Endurance limit of actual component,

$$Se = Ka \times Kb \times Kc \times Kd \times Se'$$

$$Ka = 0.6825$$

$$Kb = 1 \quad (\text{for axial loading})$$

$$Kc = 0.868 \quad (\text{for 95\% Reliability})$$

$$Kd = 0.47 \quad (\text{from charts in } q=0.75, Kt=0.2)$$

Endurance limit of standard specimen (bending test specimen)

$$Se' = 0.85 \times 0.3 \times 710 \quad (S_{ut} = 710 \text{ MPa for Al 7068-T6})$$

$$Se' = 181.05 \text{ MPa}$$

Test 1- Tensile Load on specimen 8.7 KN and 0.4KN

Corresponding stresses are,

$$\sigma_{1max} = \frac{F}{(w-d) \times t} = \frac{8700}{40} = 217.5 \text{ MPa}$$

$$\sigma_{1min} = \frac{F}{(w-d) \times t} = \frac{400}{40} = 10 \text{ MPa}$$

$$\sigma_{1mean} = \frac{\sigma_{max} + \sigma_{min}}{2} = 113.75 \text{ MPa}$$

$$\sigma_{1amp} = \frac{\sigma_{max} - \sigma_{min}}{2} = 103.75 \text{ MPa}$$

Endurance limit for specimen,

$$Se = Ka \times Kb \times Kc \times Kd \times Se'$$

$$Se = 0.6825 \times 1 \times 1 \times 0.47 \times 181.05$$

$$Se = 58.07 \text{ MPa}$$

For constant preload,

$$\sigma_{2amp} = \frac{8000}{25 \times 6} = 53.33 \text{ MPa}$$

Double lap joint is subjected to Biaxial stress,

By distortion energy theory,

$$\sigma_{r_m} = 113.75 \text{ MPa}$$

$$\sigma_{r_a} = 89.86 \text{ MPa}$$

$$Sf = S_{ut} \times \left( \frac{\sigma_a}{S_{ut} - \sigma_m} \right)$$

$$Sf = 107 \text{ MPa}$$

By using S-N curve

$$a = \log_{10}(0.9 \times S_{ut}) = 2.8055$$

$$b = \log_{10}(Sf) = \log_{10}(107) = 2.0294$$

$$d = \log_{10}(Se) = \log_{10}(50.41) = 1.7025$$

No. of cycles can be calculated by using below formula,

$$(a-d)/(b-d) = 3/(6-N)$$

$$N = 5.19$$

$$\text{No. of cycles} = 10^N = 154881.66$$

Similarly other results with different pretension were calculated & tabulated as below.

Max. Load(N)	Pretension	Fatigue life
8700	8	154881
8700	6	131825
8700	4	117489

Table 3 Analytical Results of Double lap bolted joint

**2. Numerical solution by using Ansys.**

Finite Element Analysis is performed using ANSYS workbench 14.5. The model of Plain specimen & Double lap bolted joint specimen are generated and used for Finite Element Analysis. The modal analysis of Plain specimen and Double lap bolted joint specimen are carried out to determine the effect of pretension on the fatigue life. The material properties for Plain specimen as follows, Young’s Modulus of elasticity (E) =73.1GPa,Poisson ratio ( $\nu$ ) =0.33, Density ( $\rho$ ) =2800 Kg/m<sup>3</sup>.The material (Al 7068) properties as follows,  $S_{yt}$  =648MPa, $S_{ut}$  =710MPa,Density( $\rho$ ) =2851 Kg/m<sup>3</sup>.  
FEA of Plain specimen

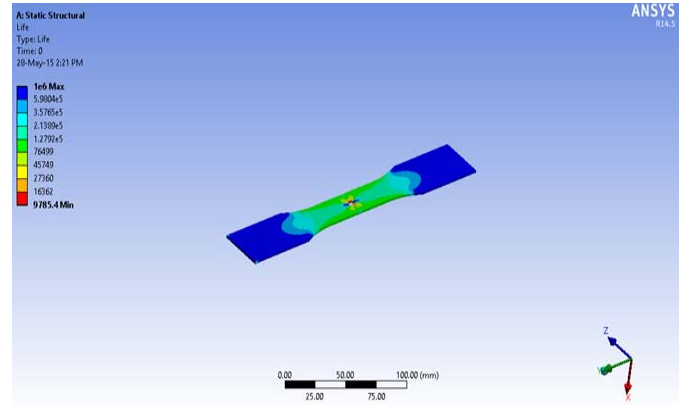


Fig 5. FEA result for Plain specimen with 11 KN load.

$P_{max}$	$P_{min}$	Cycles
8200	200	28214
8700	200	22588
11000	400	9785

Table 4 life of plain specimen calculated numerically (FEA)

Above table shows fatigue life of plain specimen obtained from ANSYS. Numerical results show that, increase in tensile load will induce high tensile stresses in the component leading fatigue failure of the component. In the similar way double lap bolted joint analysed by using FEA with pretension of double lap bolted joint.

FEA of double lap bolted joint

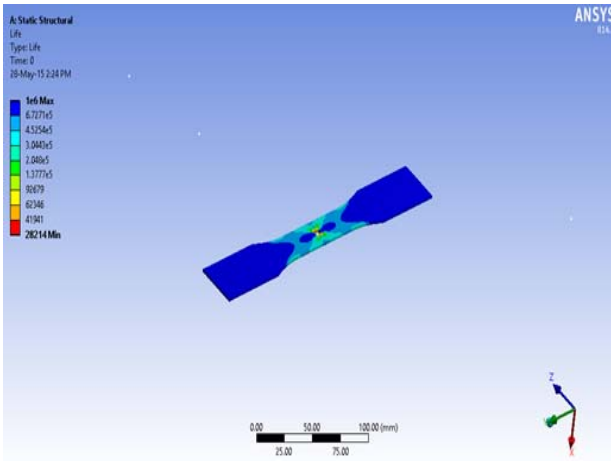


Fig 3. FEA result for Plain specimen with 8.2KN load.

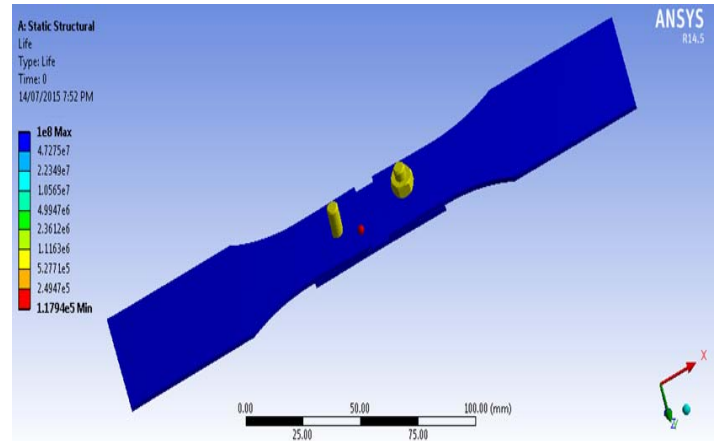


Fig 6. Load=8700 N, Bolt pretension=4000 N

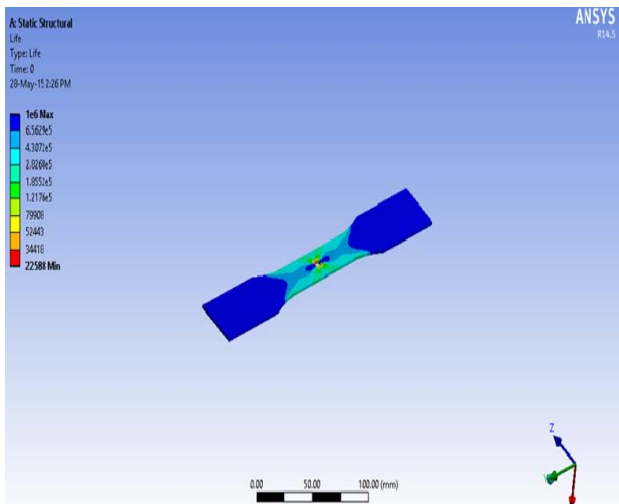


Fig.No.4. FEA result for Plain specimen with 8.7KN load.

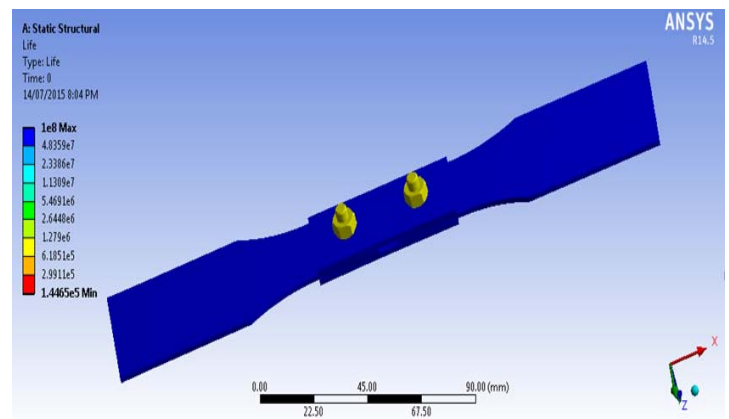


Fig.No.7. Load=8700 N, Bolt pretension=6000 N

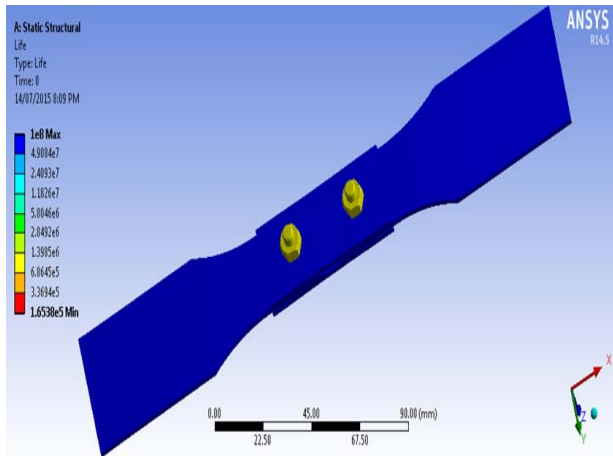


Fig 8. Load=8700 N, Bolt pretension=8000 N

Max. Load(N)	Pretension	Fatigue life
8700	8	1.6538e5
8700	6	1.4465e5
8700	4	1.1794e5

Table 5 Numerical (FEA) results of Double lap bolted joint

Above table shows fatigue life of double lap bolted joint specimen obtained from ANSYS. It shows that, increase in pretension, fatigue life of the component also increases.

### 3. Results & Discussion

Earlier, the material use in aircraft application have higher weight and lower strength which leads to fatigue failure results in less number of Fatigue life cycle, so to avoid this failure it is necessary to increase the strength of material to withstand for such heavy application which can be done by changing the material and material properties. So we have decided to use aluminum alloy 7068-T6 which has high tensile stress and low weight. Fatigue life cycles obtained by analytical & Numerical methods are compared with each other & % variation is calculated as in below table.

P max (N)	Life Analytical	Life Numerical	% variation
8200	27983	28214	0.82
8700	22686	22588	0.43
11000	9797	9785	0.12

Table No.6 % variation in results (Plain specimen)

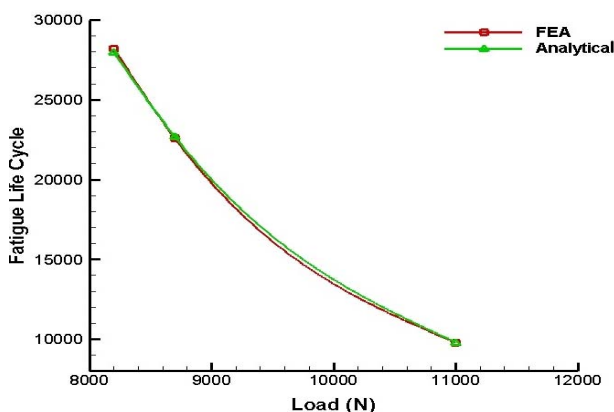


Fig.No.9 Comparison of results (Plain Specimen)

From Table No.6 it is observed that as the tensile force on plain specimen increases, fatigue life is decreases. It is also observed that the % variation in results is below 1% which is accepted. From Fig.No.9, it is clear that FEA & Analytical results are good in match.

Pretension (Nm)	Life Analytical	Life Numerical	% variation
4	117489	1.1794e5	0.38
6	131825	1.4465e5	9.27
8	154881	1.6538e5	6.55

Table No.7 % variation in results (Double lap bolted joint)

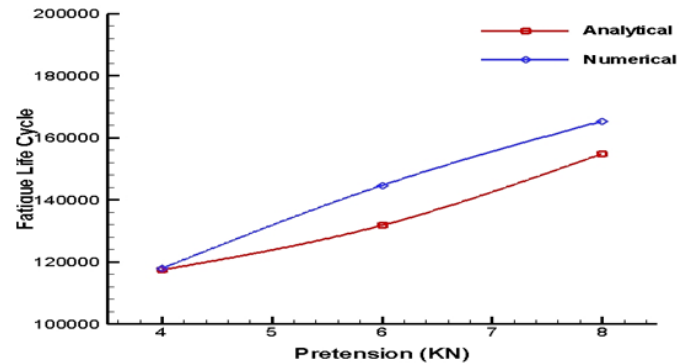


Fig.No.10 Comparison of results (Double lap bolted joint)

From Table No.7, it is observed that % variation in the results for double lap bolted joint is below 10% which is acceptable. From Fig.No.10, it is seen that all results are in good match.

### 4. Conclusion

The effect of the bolt pretension was studied and analysed. The following observations are made:

It is seen from the results that as the value of bolt pretension increases, the life cycles of double lap bolted joint increases. When torque application at the bolt increases then it results in a compression of joint members, which causes friction between them and hence very high proportion of shear load may be transmitted through the joint by friction, hence load is distributed over large area around the hole and hence stress concentration factor is vanished so fatigue performance of joint is much improved.

- From Fig.No.9&10, it is seen that all the Numerical results (FEA) & analytical results are in good match.
- We can conclude that as the tensile force on the specimen increases the fatigue life is decreases due to increase in stress concentration at hole.
- But by application of pretension on the double lap bolted joint, as pretension increases then fatigue life also increases.

So the pretension is the best way to increase the fatigue life of double lap bolted joint.

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