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AN EXPERIMENTAL STUDY OF TIGHTENING EFFECTS ON THREADED FASTENERS WITH ENLARGED THREAD FILLET RADIUS

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Abstract - The uniform thread engagement of nut -bolt makes the stress distribution between the threads more uniform and thus the nut-bolt assembly becomes more reliable. But due to the manufacturing variables the stresses in the nut and bolt assembly during tightening and the load sharing of individual mating threads remain non-uniform in general.Studies have confirmed that the threads of nut-bolt assembly experience stresses being geometrically distributed, initiating from maximum at the first thread which is near to bearing surface and then reducing subsequently. A large fillet radius at the thread root which is more than the standard is proposed and to realise the effects, undercutting is done within the thread root profile of ISO standard M24 bolts so as to induce elasticity in threads and changing the stress distribution from geometric to simple arithmetic. Four specimens with different undercuts are investigated experimentally for proving the effectiveness of the proposed modifications. The effect of tightening torque is observed on one experimental set up. The results are compared between the existing and corrected thread profiles and the corrected thread profile has given better results.

Keywords - Stress Distribution, Fillet Radius, Undercutting, Tightening Force.

I. INTRODUCTION

The most commonly applicable fastening method is to use threaded fasteners since they have ease of bulk manufacturing, interchange ability due to standard sizes, easy disassembly for maintenance purpose and reliable life. The reliability of a complex mechanical assembly hence depends upon the reliability of threaded fasteners used within it. A nut-bolt pair is said to be failed if there is a loss of clamping load that holds two components together. The loss of clamping load is due to non-uniform nature of stress distribution between the mating threads of nut and bolt pair. A complex failure pattern is observed in a nut-bolt assembly. Many researchers in the past have studied the stress distribution pattern between the mating threads of nut-bolt assembly.

N. E. Joukowsky[1] proved that the stress distribution pattern in a nut-bolt pair is of non-uniform nature and follow geometric progression. The very first mating thread is the most heavily loaded thread and carries around 33% of the total axial load. Fig.1 shows the graphical geometric distribution pattern of axial load within the mating threads.

D.G. Sopwith [2] analysed the nut-bolt pair on the basis of strains developed between the mating threads which he assumed to be cantilever beams and also suggested the means to make the stress distribution more uniform within the mating threads of a nut-bolt pair and also a turnbuckle. The maximum intensity of stress is at the bearing face of nut.

A.F.C Brown and V.M. Hickson [3] came up with their research to prove some theoretical results of Sopwith through the photo-elastic techniques. They investigated the effects of nut with a tapered body tightened over a screw having a recess.

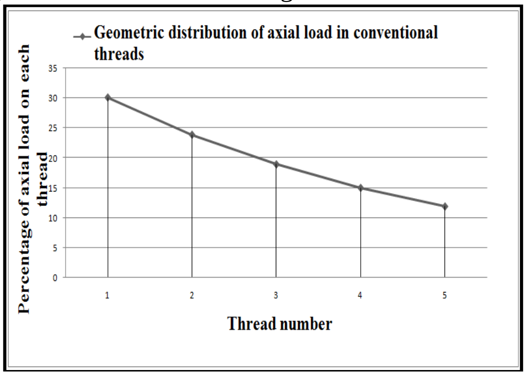


Fig.1- Geometrical Distribution of Axial Load in the Conventional Threads. (Thread Number Counted from Bearing Surface)

The results were discussed on the basis of graphs plotted between the stress concentration factor at thread root and the number of threads in contact from the free end. David L. Miller, Kurt M. Marshek and Mohammad R. Naji [4] developed another analytical method to determine the load distribution in a pair of nut-bolt assembly. Simulation of the threaded components with that of an assembly of springs properly connected so as to represent the desired loading conditions of compression and tension cases were discussed. Second order differential equations were given for the spring model analysis. Researchers have proposed that the models could be adaptable for different thread profiles.

B. Kenny and E.A. Patterson [5] reported that the previous studies considered the load distribution between the mating threads of nut-bolt pair indirectly by measurements of either stress concentrations or deformations in nut.

E.A. Patterson and B. Kenny [6] optimised the design of nuts so as to reduce the maximum stress value by tapering the threads of nut. Tapering the threads of a nut by two degree reduces the maximum stress value by 41 percent for 5.5 threads of ISO standards. The experiments are based on the photo elastic analysis done by researchers.

E.A. Patterson and B. Kenny [7] investigated the modifications to the external shape of nut so as to observe the effect on the stress levels of an axially loaded bolt. A circumferential groove is cut over the external face of nut and also a bevel at the load bearing face. A reduction of 26 percent in the maximum stress in the bolt is observed through three-dimensional photo elastic techniques.

Niels Leergaard Pedersen [8] suggested that it is the thread root profile that controls the stress concentration in the threaded connections. A super elliptical thread root profile is proposed against the standard circular profile.A reduction of 8.9 percent in the stress concentration factor is observed.

The literature suggests that the researchers have been making keen attempts for many decades in order to realize the practical stress distribution conditions between the engaged threads of a nut-bolt pair. Generally, the nuts are manufactured by some softer material as compared to the material of bolt so as to provide the elastic and surface bearing effects for the stresses generated when the nut is tightened over the bolt [9]. Many attempts have been made to uniformly distribute the load in the threaded connections by making modifications in the thread profile of nut and also the overall design of a nut [10].

II. LOAD DISTRIBUTION IN THE TIGHTENED THREADS

The geometric distribution of load in the tightened threads can be visualised in Fig.2

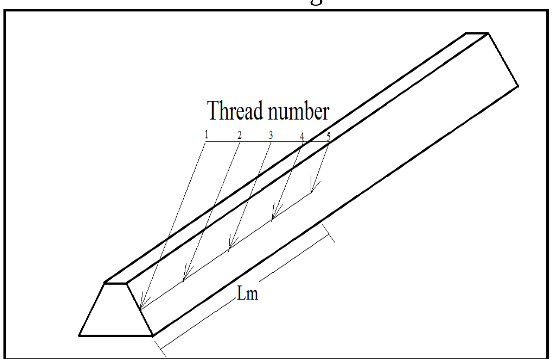


Fig.2- Axial Load Distribution (In Geometric Progression) for A Continuous Unwound Conventional Thread.

Here, L_m = Effective length = $2\pi.R_m$. nWhere, R_m = Mean radius and n = thread number, which is 5 in this case.

The strains developed in thread are of complex nature due to complex geometry. The first thread distorts maximum due to initial compression. The elastic shape changes due to compressive stress wave that evens out at the n^{th} thread where the shape change is negligible. The thread is considered to be acontinuous element and therefore the bulk modulus (K) may be preferred instead of elastic modulus (E) and an effect of strain energy comes into effect.

A general expression for relation between the young's modulus and the bulk modulus is given as

E = 3K(1 -2□□

Where, □ = Poisson's ratio (0.25-0.33 for isotropic engineering materials)

Keeping the value of Poisson's ratio = 0.29 (average of 0.25 and 0.33) in the above relation, we get

E = 1.26 K

Now a two-dimensional view of the load distribution pattern can be shown in Fig.3

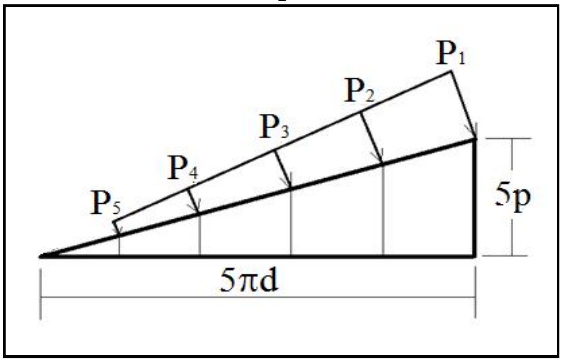


Fig.3- 2D Pattern for Load Distribution in a Continuous Thread of Five Pitches

Considering that screw has five threads then it can be represented as the base of right triangle with its measurement as 5πd and the height of triangle is 5p, whereas the load distribution pattern is shown by P_1, P_2, P_3, P_4, P_5, over each pitch respectively. The hypotenuse of this right triangle is inclined at an angle equal to helix angle of thread.

The load distribution pattern according to the literature review is in geometric distribution which can be explained below:

$P_1 = P$

$P_2 = P / \Phi$

$P_3 = P / \Phi^2$

$P_4 = P / \Phi^3$

$P_5 = P / \Phi^4$

Where,

1 / Φ, is the common ratio of between each successive and preceding load.

Now,

$P_1 + P_2 + P_3 + P_4 + P_4 + P_5 = P_{ax}$

Where, P_{ax} = Total axial load on bolt

Therefore,

$P + P / \Phi + P / \Phi^2 + P / \Phi^3 + P / \Phi^4 = P_{ax}$

$P\,[1 + \frac{1}{\Phi} + \frac{1}{\Phi 2} + \frac{1}{\Phi 3} + \frac{1}{\Phi 4}] = P_{ax}$

If, Φ = 1.26, which is a factor relating the elastic modulus to that of bulk modulus then the following results are obtained

$$\frac{Pax}{3.320} = P$$

From the above relation the values of load shared by

each thread in geometric series is given as
P_1 = 30.12% of P_{ax}
P_2 = 23.91% of P_{ax}
P_3 = 18.97% of P_{ax}
P_4 = 15.05% of P_{ax}
P_5 = 11.95% of P_{ax}
The first load takes maximum portion of the load while the last thread takes the minimum portion of the load. Thus, the load distribution is not uniform and each thread shares a different load and so different stresses and elastic deformations occur. It causesuneven local deflections and possibility of plastic deformation and then failure.
The current work is to propose a method for uniform distribution of load amongst the mating threads.The proposal is based on the redistribution of load from geometric proportion to arithmetic proportion through undercutting within the thread root of a bolt and some experimental work.Fig.4 shows an example of proposed thread profile.

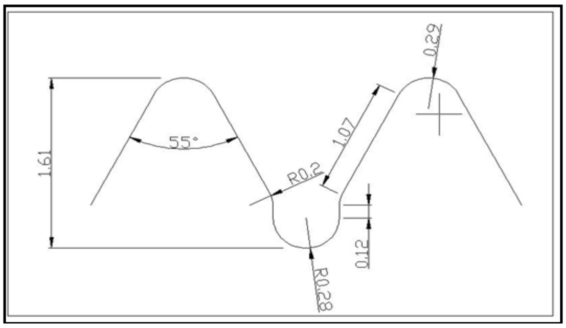


Fig. 4– Correction in The Root of The Thread by Undercutting. (One Example)

III. EXPERIMENTAL WORK

One experimental set-up has been adopted for the experimental study of the effect of undercutting on tightening force requirements of specimens and the standard bolt. The drawing of set-up is shown in Fig.5.

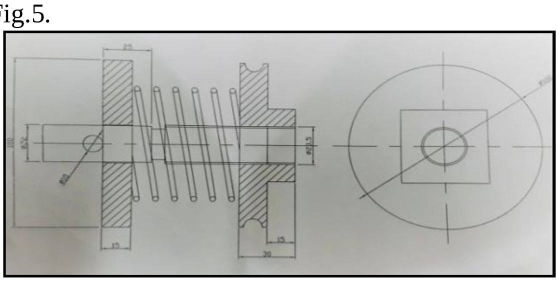


Fig.5- Drawing of the experimental set-up.

The bolts are tightened in the experimental set-up by the input load of fine sand bag on the pulley string. The sand is utilised for the load application so that a slow, small and continuous increment in load application can be observed. The pulley is having internal threading to mate with the male threads. As the tightening is done by applying the weight, the deflection is observed in the spring. The constant spring stiffness (15.38 N/mm) and the deflection in the spring give the output load. The spring load as output load and the sand load as input load are compared. A snug fit is given prior to the load application for further tightening since it is to be ensured that the threads are mating with each other and the incremental effects of load would show only the resistance of threads for tightening. The spring between the two discs acts as clamping parts. It is to be noted here that there is no component of surface bearing friction in the experimental set-up.

The ISO standard M24 bolts are machined on CNC turning lathe. The four specimens with an undercut of 0.2, 0.3, 0.5, and 1 mm, respectively in their thread root profile.Fig.6 shows the prepared specimens.

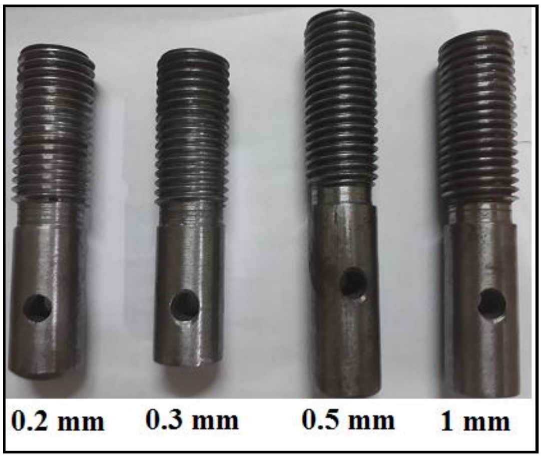


Fig.-6 Specimens Having Undercut Within the Thread Roots.(Depth of Undercut Shown Below the Specimen)

A close front view of threads of bolt with an undercut of 1mm is shown in Fig. 7.

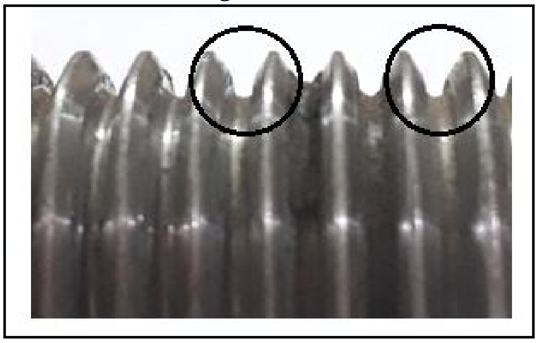


Fig. 7- Front View of Bolt Specimen with 1mm Undercut. (Undercut is shown within Circle)

IV. RESULTS AND DISCUSSION

For each specimen the results are tabulated below:

Undercut depth (mm)	Applied load (N)	Spring deflection (mm)	Clamp load (N)	Efficiency of tightening
Nil	92.2	3	46.14	50.05 %
0.2	85.5	3	46.24	53.98 %
0.3	79.5	1.5	23.07	29.03 %
0.5	71.5	3.2	49.22	68.85 %
1	83.1	3.8	58.44	70.50%

Table no. 1- Input loads v/s Output loads

Various trials were made to get best results from the experimental set up and it was so decided that the undercut from the available root profile be given as per the ratio of 0.12-0.14p.
Where p = thread pitch.
The graphical representation of the table is shown in Fig.8.

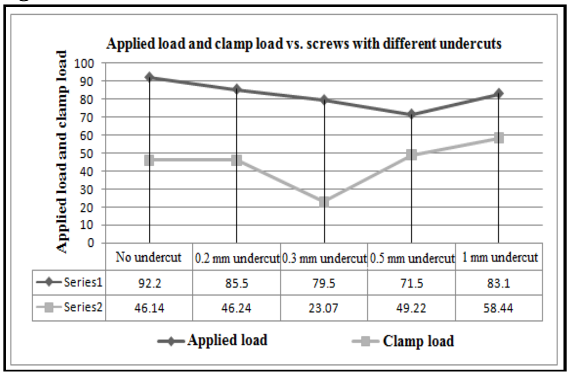


Fig.8- Applied Load and Clamp Load(In Newton) Vs. Threads with Different Values of Undercut.

The frictional characteristics are not altered and no lubrication is applied. It can also be seen that the bolt with an undercut of 1 mm is also quite effective but the applied load is also increased.Initially with no undercut the applied load is maximum and the clamping load is minimum. Any amount of correction in the thread is more efficient than without corrected threads. The percentage change of applied load for all specimens is calculated with respect to load measured for bolt without undercut. 22.45 % reduction in the load for bolt of 0.5mm undercut is observed, which is a maximum reduction. This shows that for different thread profiles and different diameters the correction may vary and needs to be evaluated. The reason may be attributed to the resultant change in the geometry due to induced elasticity of thread. One acrylic model was prepared to see the effect of undercutting on the thread matching of nut –screw pair. The Fig.9 shows with clarity the regular pattern of thread matching after correction in root profile.

Fig.9 - One Acrylic Model to Check The Pairing

It appears that the all the threads are in proper contact. The load distribution of mating threads in the geometric progression is non-uniform but as per the observation in the model the load sharing is uniform and it can be assumed due to the correction in the root profile the load distribution is in arithmetic progression and linear. The comparison is shown in Fig.-10

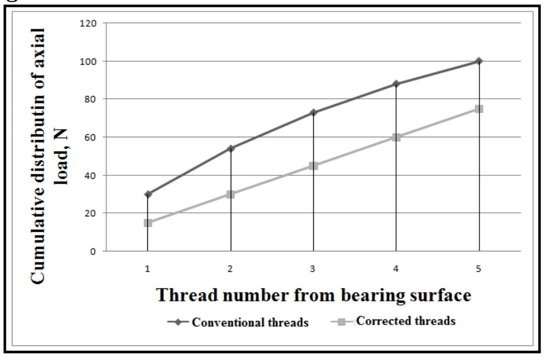


Fig.10 – Comparison of Load Sharing Amongst Thread Pairs.

On the basis of the tabulated data and the graph, it is observed that the load required for tightening is reduced for bolts having corrected profile. The graph shows that corrected fastener compresses the component [in this case spring] more as compared to the uncorrected threads. That is to say that the deformation / frictional resistance between the threads is effectively reduced and utilisation of applied load is increased. There is local elastic adjustment in threads due to change in physical geometry in threads. A vivid picture of load distribution on the basis of reduced applied load and increased clamp load can be drawn with reduction of 25% in applied load and the clamping load been distributed arithmetically. This is shown by Fig.11.

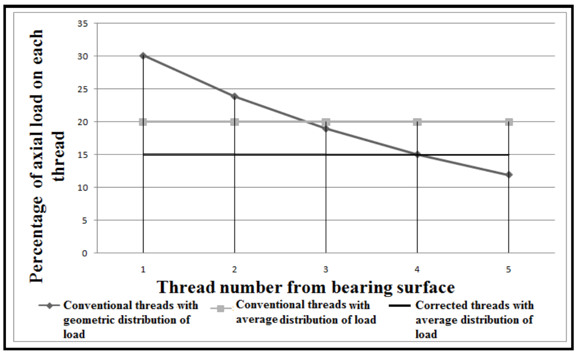
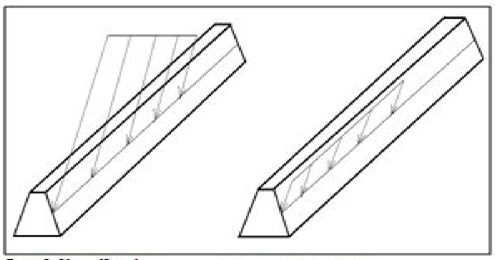


Fig.11 – Comparison of Load Sharing Amongst Thread Pairs.

The average load on corrected screw is reduced to around 22% as compared to the conventional screw. The tightening torque will also be less and wear of the threads also will be uniform, as an outcome of corrected threads. The comparison is pictorially made as per Fig.12.



Load distribution over conventional thread profile

Load distribution over corrected thread profile

Fig. 12- Comparison of Load Distribution.

CONCLUSION

It can be deduced from the experimental data that due to correction in root profile of the screw the tightening load value gets reduced and the load distribution becomes uniform. The undercut for the corrected root profile is found to be in the range of 0.12 to 0.14times thread pitch.

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