OPTIMIZATION OF BOLTED JOINT OF A GAS TURBINE CASING USING DOE

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Abstract: In this work, a sector model of a bolted flange is modeled in the Design Modeler of ANSYS Workbench. Model is parameterized so that by changing its geometric parameters, an updated geometric model is auto generated. The FE model is given necessary contacts, BCs,structural and thermal loads. Since a sector is considered, zero displacements are given in its hoop direction and on one cut face of sector. An axial pull is given on the other cut face of the sector. Bolt pre-load is applied along the axis of the bolt and a uniform temperature is presumed. Now the model is solved and the stresses are plotted.In the Design Explorer of workbench, optimization is carried out through the concept of Design of Experiments and Response surface. In the design explorer input and output parameters are defined and bounds(upper and lower) are specified to the defined input parameters. Within the specified bounds, DOE generates the best possible design points along with the corresponding output parameter values. A response surface is then generated which depicts the variation of o/p parameter with changing i/p parameter values. Then, three most promising design candidates are being generated by the optimization cell of which one best design candidate is chosen as per the goal of optimization. The selected design candidate is solved and the results are evaluated. It's observed that the maximum stress values have been declined after design exploration.

IndexTerms – Bolted joint, Pre-load, Design of Experiments, i/p&o/p parameters, stress.

I. INTRODUCTION TO BOLTED JOINTS

When a connection is desired that can be disassembled without destructive methods and that is strong enough to resist external tensile loads, moment loads, and shear loads, or a combination of these, then the simple bolted joint using hardened-steel washers is a good solution.

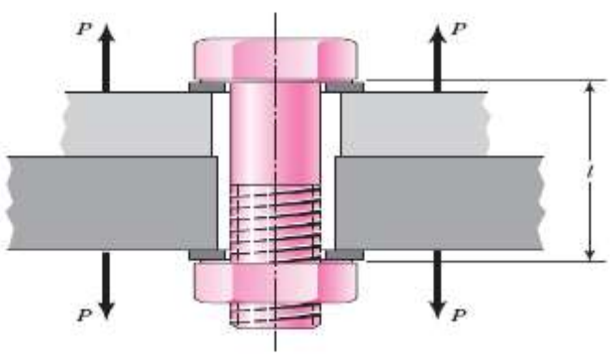

Fig 1: Bolted joint

Twisting the nut stretches the bolt to produce the clamping force. This clamping force is called the pretension or bolt preload. It exists in the connection after the nut has been properly tightened no matter whether the external tensile load *P* is exerted or not. Of course, since the members are being clamped together, the clamping force that produces tension in the bolt induces compression in the members

II. LITERATURE REVIEW

Research on bolted joints is being carried out from many years. Current literature explains the research done on bolted joints and optimization concept in Design Explore module of ANSYS workbench, where it helps in understanding different types of experimental and analytical methods used to find how these bolted joints work efficiently. The bolted joint is very widely used assembly method, therefore a very precise analysis of parts of the joint is required in order to have high accuracy in the global mechanical system

In a journal R.M.El-Zahry [1] Optimum design of preloaded bolted joint under harmonic excitation The dynamic response of a preloaded bolted joint is investigated when subjected to harmonic excitation. The equations of motion of the joint are derived on the basis of a realistic linear mathematical model, by using simple spring-mass system analysis. Solutions for the joint response are obtained in the frequency domain in terms of the force and motion transmissibility ratios between joint plates. Although the main emphasis of this study is focused on the dynamic response of preloaded bolted joint as a criterion of design, the data on the joint design based on yield and fatigue criteria are also discussed.

In a journal S.H. Ju et.al [2] Finite element method is used to study the structural behaviour of bolted joints and its numerical results are verified with AISI specification. The similarity was found to be satisfactory during the loading stages. The bolt nominal forces obtained from FE Analysis are almost linearly proportional to the bolt arranged in connection. For the cracked plate in a bolted joint structure, the relationship between K_I and the applied load is near linear , in which non linear part is only about one tenth of the total relationship. This means the linear elastic fracture mechanics can still be applied to the bolted joint problem for the major part of loading.

Nomesh Kumar et.al [3] "3-D Finite Element Analysis of Bolted Flange Joint of Pressure Vessel" In this paper it was found that, the stresses in the bolts of the bolted flange joint of the pressure vessel so that bolts/studs should not be failed during proof pressure test. Bolted flange joints perform a very important structural role in the closure of flanges in a pressure vessel. It has two important functions: (a). to maintain the structural integrity of the joint itself, and (b). to prevent the leakage through the gasket preloaded by bolts. The preload on the bolts is extremely important for the successful performance of the joint. The preload must be sufficiently large to seat the gasket and at the same time not excessive enough to crush it. The flange stiffness in conjunction with the bolt preload provides the necessary surface and the compressive force to prevent the leakage of the gases contained in the pressure vessel. Due to flange opening , bending has been noticed in the bolt. Due to existence of Preload, internal pressure and bending moment at a time, the bolt behavior is nonlinear. 3-Dimensional finite element analysis approach is only the technique which shows some satisfactory result.

G. Gerbert et.al [4] "Centrically Loaded Bolt Joints" Externally loaded bolt joints of different design are investigated. The bolt load is substantially lower than that predicted by the mounting stiffnesses of bolt and abutment. A load application fraction n is introduced in VDI 2230 and some thumb rules are suggested. New fractions are determined which are much lower than the suggested ones. It appears that the application fraction n is almost independent of the location of the external load in practical design but it is influenced by the layout of the bolt joint.

III STRESS ANALYSIS OF A BOLTED JOINT

The input geometric parameters for optimization are taken as the following

1. Fillet radius of left flange 2. Fillet radius of right flange 3. Width of left flange and 4.Width of the flange

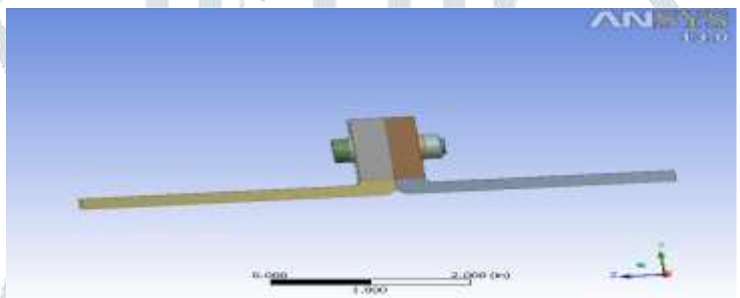

Fig 2: Design Modeler workspace

The following are the contacts given in the joint:

1. frictional contact between the two flange surfaces-1
2. frictional contact between washer surface and flange surface-2
3. frictional contact between bolt head surface and washer surface-1
4. frictional contact between nut surface washer surface-1
5. bonded contact between shank outer surface and nut inner surface

The frictional contacts are given where the two contact surfaces slide against each other and bonded contact is given where the two contact surfaces are not supposed to have any sliding motion against each other. In the current project, bonded contact is given to the interface of bolt shank outer surface and nut inner surface. All the remaining contacts are given as frictional with a friction coefficient 0.1

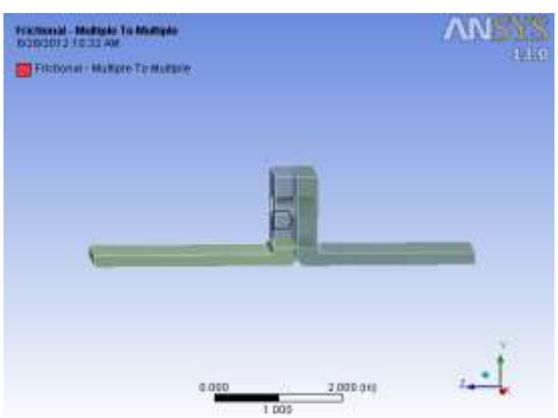

Fig3: Contact between two flange surfaces

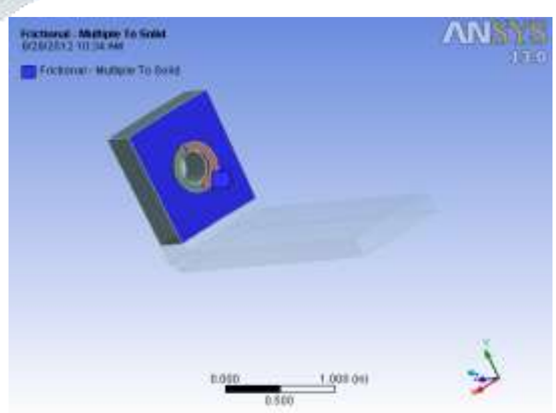

Fig 4: Contact between left flange and washer

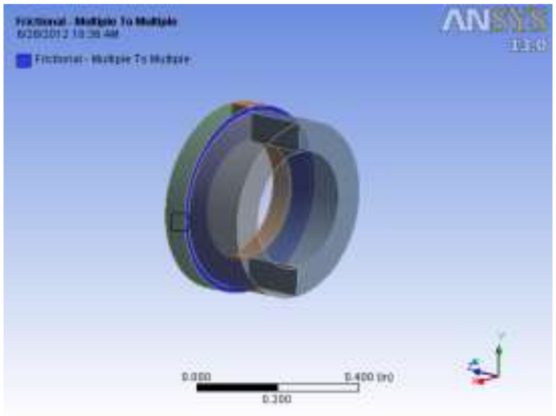

Fig 5: Contact between nut and washer surfaces

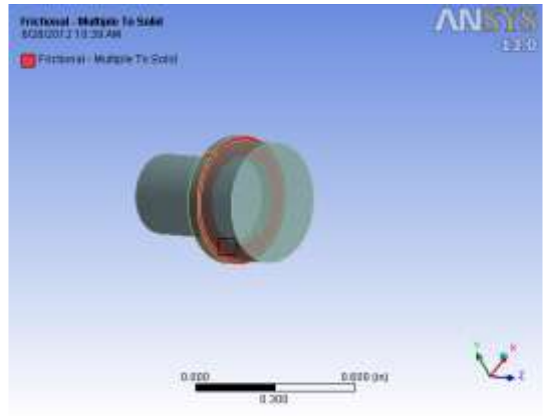

Fig 6: bolt head and washer surfaces

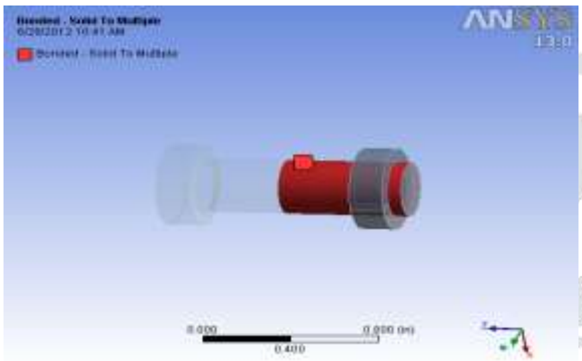

Fig 7: Bolt shank and nut surface

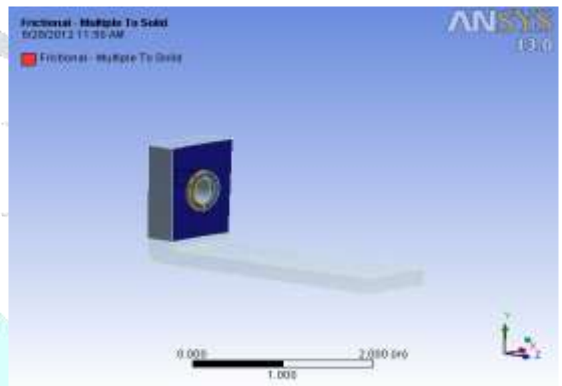

Fig 8: Right flange and washer surfaces

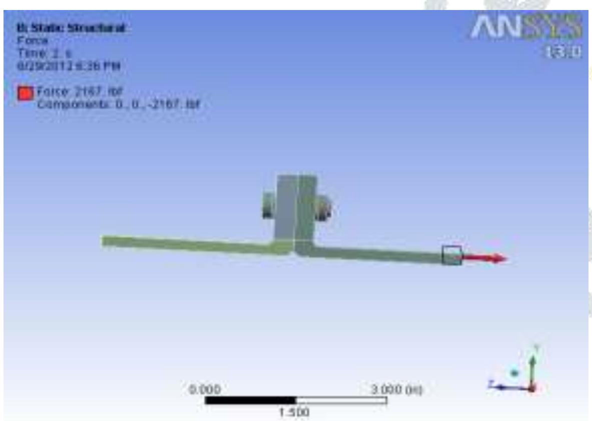

Fig 9: Force in the axial direction

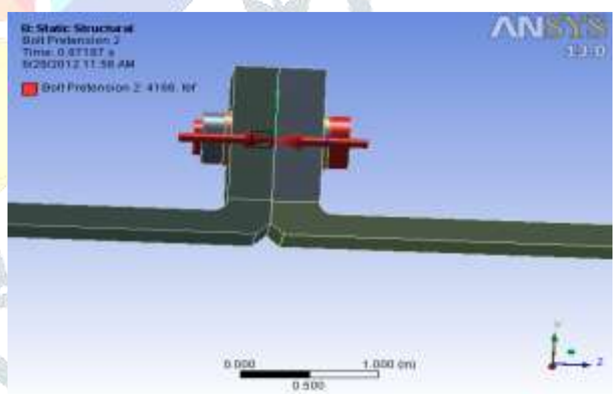

Fig 10: Bolt Pre-tension

IV RESULTS & DISCUSSION

The loading is given totally in two time steps. In the first time step, only the Bolt pre-load is considered and in the subsequent time step the axial load is considered besides the same bolt pre-load.

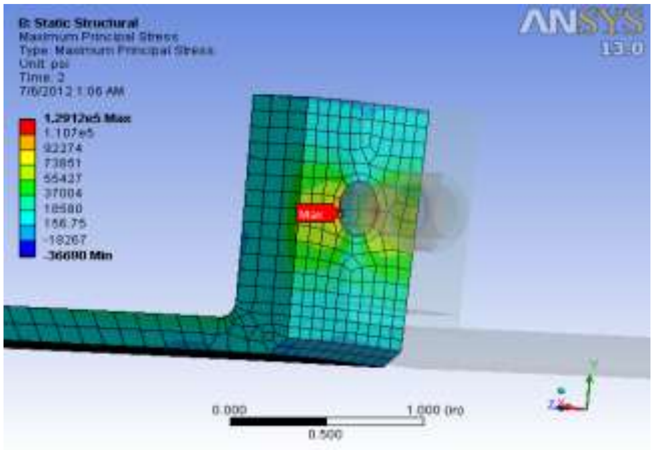

Fig 11Stress pattern in the right flange

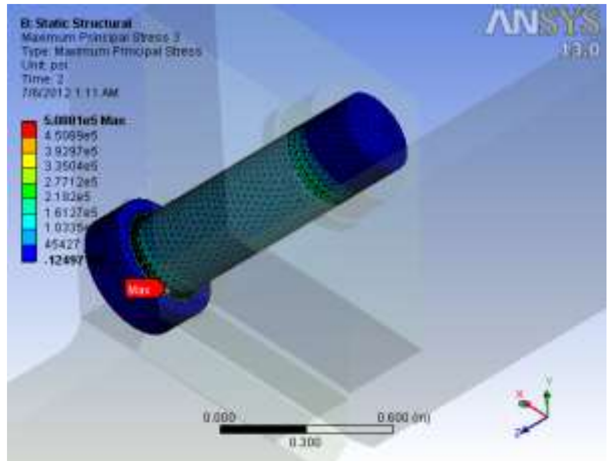

Fig 12 Stress pattern in the bolt

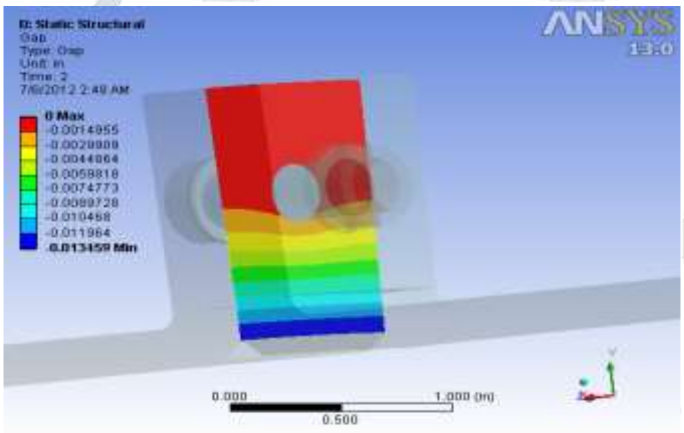

Fig 13 Contact Gap between the flanges

It's been observed that the maximum stress is induced at the circumference of the bolt hole in case of the flanges and for the bolt, maximum stress is at the fillet part. The contact gap between the two flanges also is plotted and is observed that the two flange surfaces stick to each other at their top surfaces and the gap starts just below the bolt hole of the flanges and increases towards the bottom of the two contact surfaces.

4.1 Design exploration

Design exploration describes the relationship between the design variables and the performance of the product by using Design of Experiments (DOE), combined with Response Surfaces. Once the variation of the performance with respect to the design variables is known, it becomes easy to understand and identify all changes required to meet the requirements for the product.

4.2 Design Points

Design Points are created by design exploration for instance, when processing a Design of Experiments or a Correlation Matrix, or refining a Response Surface. Note that the output parameter values are not copied to the created Design Point since they were calculated by design exploration and are, by definition, approximated. Actual output parameters will be calculated from the Design Point input parameters.

4.3 Design of experiments

Design of Experiments is a technique used to determine the location of sampling points and is included as a part of the Response Surface, Goal Driven Optimization, and Six Sigma systems. These techniques locate the sampling points such that the space of random input parameters is explored in the most efficient way, or obtain the required information with a minimum of sampling points. By default the deterministic method uses a central composite design, which combines one center point, points along the axis of the input parameters, and the points determined by a fractional factorial design.

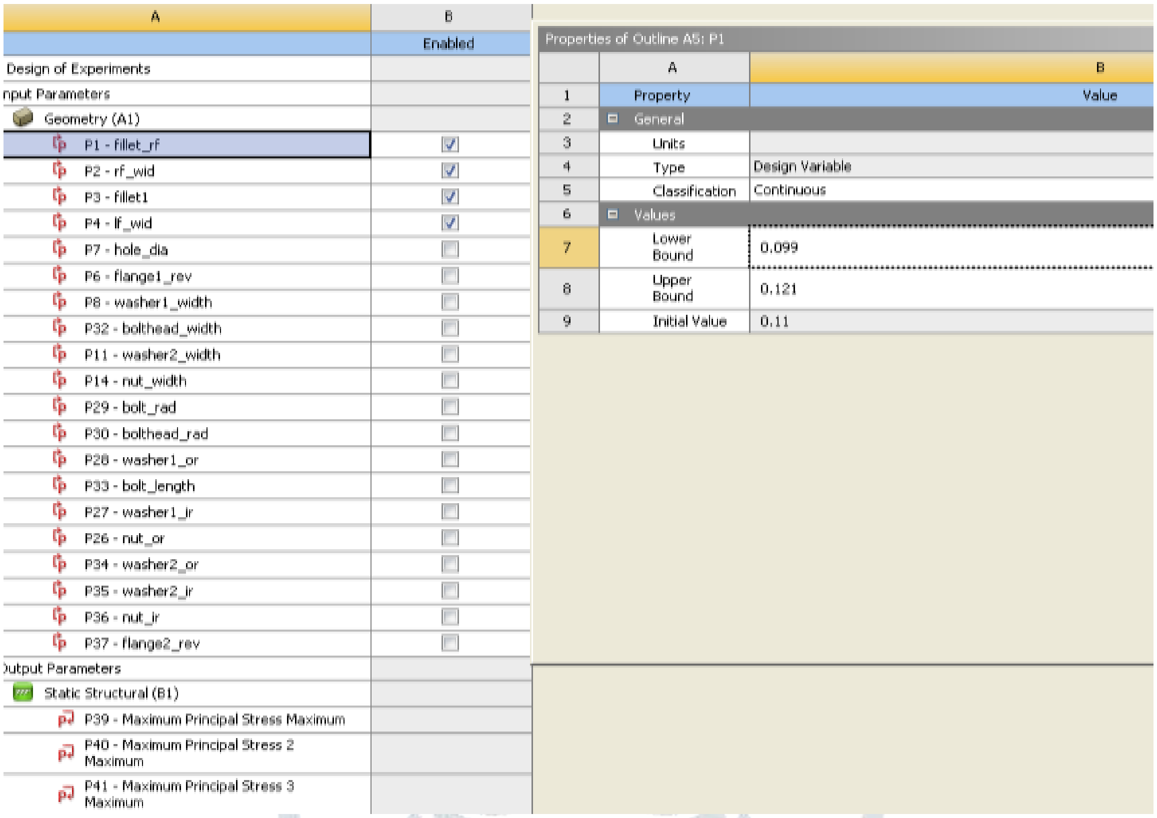

Fig 14 Defining i/p & o/p parameters for DOE

In DOE, the input parameters and output parameters are specified initially and then the bounds are given to all the input parameters. Then, a DOE type is specified and DOE is conducted. Once, the DOE run is finished successfully, it generates all the possible design points within the limits of input parameters. Once the optimization is updated successfully, it generates three promising design candidates. Figure 14 depicts
a) Defining the input parameters and output parameters
b) Enabling the required input parameters to optimization and
c) Giving bounds (upper and lower) to a selected input parameter.
This picture shows

- ➢ List of all the possible design points that are generated by conducting the design of experiments process and
- ➢ The corresponding output parameter values of all the generated design points

	Name	P1 - fillet_rf	P2 - rf_wid	P3 - fillet1	P4 - lf_wid	P39 - Maximum Principal Stress Maximum (psi)	P40 - Maximum Principal Stress 2 Maximum (psi)	P41 - Maximum Principal Stress 3 Maximu
2	1	0.11	0.3	0.1	0.35	1.2912E+05	1.3886E+05	5.0881E+05
3	2	0.099	0.3	0.1	0.35	1.2749E+05	1.6787E+05	5.2071E+05
4	3	0.121	0.3	0.1	0.35	1.2781E+05	1.6877E+05	5.1891E+05
5	4	0.11	0.27	0.1	0.35	1.242E+05	1.9532E+05	5.3264E+05
6	5	0.11	0.33	0.1	0.35	1.3132E+05	1.4453E+05	5.1198E+05
7	6	0.11	0.3	0.09	0.35	1.2803E+05	1.6749E+05	5.1994E+05
8	7	0.11	0.3	0.11	0.35	1.2707E+05	1.6731E+05	5.1971E+05
9	8	0.11	0.3	0.1	0.315	1.5204E+05	1.6428E+05	5.5853E+05
10	9	0.11	0.3	0.1	0.385	1.0944E+05	1.6068E+05	5.0211E+05
11	10	0.10225	0.27887	0.092958	0.32535	1.3034E+05	1.8498E+05	5.3274E+05
12	11	0.11775	0.27887	0.092958	0.32535	1.363E+05	1.8458E+05	5.3204E+05
13	12	0.10225	0.32113	0.092958	0.32535	1.4314E+05	1.491E+05	5.2002E+05
14	13	0.11775	0.32113	0.092958	0.32535	1.4109E+05	1.498E+05	5.193E+05
15	14	0.10225	0.27887	0.10704	0.32535	1.4081E+05	1.8479E+05	5.3259E+05
16	15	0.11775	0.27887	0.10704	0.32535	1.3941E+05	1.8438E+05	5.3191E+05
17	16	0.10225	0.32113	0.10704	0.32535	1.4568E+05	1.4903E+05	5.1994E+05
18	17	0.11775	0.32113	0.10704	0.32535	1.4428E+05	1.4878E+05	5.1905E+05
19	18	0.10225	0.27887	0.092958	0.37465	1.0085E+05	1.8848E+05	5.051E+05
20	19	0.11775	0.27887	0.092958	0.37465	1.0705E+05	1.8818E+05	5.0393E+05
21	20	0.10225	0.32113	0.092958	0.37465	1.1322E+05	1.527E+05	4.9047E+05
22	21	0.11775	0.32113	0.092958	0.37465	1.1106E+05	1.5255E+05	4.8913E+05
23	22	0.10225	0.27887	0.10704	0.37465	1.1111E+05	1.8836E+05	5.0509E+05
24	23	0.11775	0.27887	0.10704	0.37465	1.096E+05	1.8804E+05	5.0387E+05
25	24	0.10225	0.32113	0.10704	0.37465	1.1558E+05	1.5266E+05	4.9043E+05
26	25	0.11775	0.32113	0.10704	0.37465	1.1371E+05	1.5247E+05	4.8870E+05

Figure 15: Design points generation from Design of Experiments

Amongst the generated design candidates, three most promising design candidates are taken out by the optimization cell of Design of Experiments as per the goal of optimization. Out of the three , one design candidate is selected and is solved to verify the results of optimization.

Figure 16: Schematic Optimization

Table of Schematic C4: Optimization

	A	B	C	D	E	F	G	H
1		P1 - Fillet_rf	P2 - rf_wid	P3 - fillet1	P4 - ff_wid	P39 - Maximum Principal Stress Maximum (psi)	P40 - Maximum Principal Stress 2 Maximum (psi)	P41 - Maximum Principal Stress 3 Maximum (psi)
2	Optimization Study							
3	Objective	No Objective	No Objective	No Objective	No Objective	Minimize	Minimize	Minimize
4	Target Value							
5	Importance	Default	Default	Default	Default	Default	Default	Default
6	Candidate Points							
7	Candidate A	0.12099	0.32435	0.090293	0.30472	1.0475E+05	1.4891E+05	4.7714E+05
8	Candidate B	0.11403	0.32704	0.10924	0.30114	1.008E+05	1.477E+05	4.8543E+05
9	Candidate C	0.11439	0.32109	0.096402	0.30330	1.0041E+05	1.5006E+05	4.9309E+05

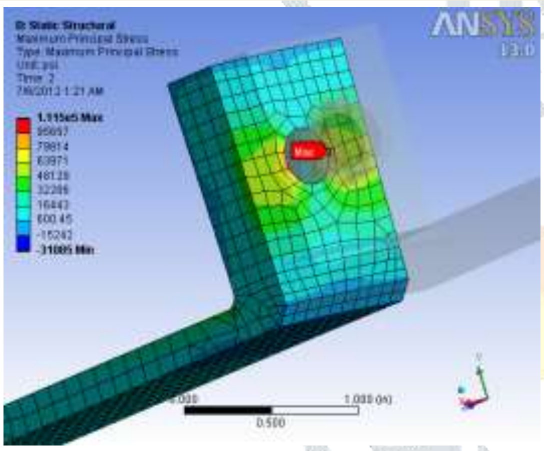

Figure 17: Stress Pattern in the Flange

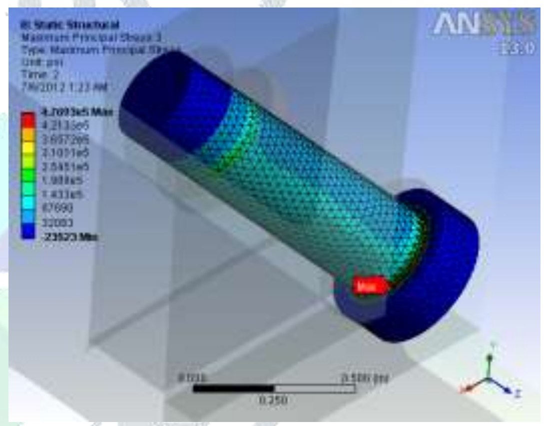

Figure 18: Stress Pattern in the Bolt

V. CONCLUSION

In the first part of this paper, the stresses are evaluated in the usual way for the bolted joint in ANSYS workbench. After that, Design of experiments is conducted to the parametric model through design explorer module of ANSYS workbench. DOE generates list of all possible design candidates and simultaneously solves the stress values for all those possible design candidates. By default, central composite design technique is used by the design explorer to generate all feasible design points. One best design candidate is selected and the stresses are observed for that design candidate. When the results after performing optimization are compared to the results before the optimization, a noticeable decrease in the maximum principal stress values are observed. This work mainly aimed at performing DOE through the design explorer module of ANSYS workbench.

For FLANGE:

The maximum principal stress has been reduced from 890.25 MPa to 796.34 MPa

For BOLT:

The maximum principal stress has been reduced from 3.5 GPa to 3.2 GPa

VI. REFERENCES

[1] R.M.El-Zahry, “Optimum design of preloaded bolted joint under harmonic excitation” Journal of Sound and Vibration, volume 108,Issue 3, 8 Aug 1986, Pages 455-470.

[2] S.H. Ju, C.Y. Fan and G.H. Wub, “Three-dimensional finite elements of steel bolted connections” Engineering Structures, Vol. 26, (2004), pp. 403–413

[3] Nomesh Kumar, P.V.G. Brahamanandam and B.V. Papa Rao, “3-D Finite Element Analysis of Bolted Flange Joint of Pressure Vessel” MIT International Journal of Mechanical Engineering, Vol. 1, No. 1, Jan 2011, pp. 35-40

[4] G. Gerbert and Hakan Bastedt “Centrically Loaded Bolt Joints ASME, Journal of Mechanical Design, Volume 115, Issue 4, Research Paper.

[5] FEA and Design exploration of bolted joints by Luhana Piscan, Nicole Predentiae and Nicolae Pop, MIT International Journal

titled 3-D FEA of Bolted flange joints.
[6] A limit analysis study to interpret ultimate behavior of Bolted Joints by R. Accti, G. Ballio and l. Corradi.
[7] Parameter Optimization In Radial Basis Function Response Surface Approximation. Wang, Bo Ping. Denver : AIAA, 2002.
[8] Rao, Singiresu S. Engineering Optimization. Third.sl: Wiley Interscience Publications, 1996. 0-471- 55034-5.
[9] Majid, K.I. Optimum Design of Structures. New York : Wiley, 1974.
[10] Carmichael, D G. Structural Modeling and Optimization. West Sussex : Chichester, 1981.
[11] Kirsch, U. Optimum Structural Design. New York : McGraw-Hill, 1981.
[12] Jing Jin, Xinyue Wu, Analysis and optimization on the explosive impact response of bolt joint structure, Reliability, Maintainability and Safety, 2009. ICRMS 2009. 8th International Conference