IMPACT OF NOTCH GEOMETRY ON THE FATIGUE LIFE OF BS 970-4 GRADE 349S52 AUSTENITIC STAINLESS STEEL

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Abstract - *The current research work deals with finite element analysis of the impact of notch geometry on the pressure bearing capacity of BS 970-4 grade 349S52 austenitic stainless steel under fatigue loading. Fatigue testing specimen prescribed in ASTM E606 standard is modelled in Creo 3.0 and exported to Ansys Workbench 18.1. Thereafter, the finite element model is prepared to perform fatigue analysis using Soderberg's method of the stress-life approach. The pressure to be applied so as to induce an equivalent (Von-Mises) stress of around 546 MPa so that the fatigue life of around 21,500 is found out on an iterative approach. Experimentation is conducted as per Taguchi method and optimal parameters are identified. It is found that the depth of notch plays a vital role in the deterioration of the pressure bearing capacity of the material when compared to width and notch central angle.*

Key Words: ANSYS Fatigue Tool, ASTM E606 standard, Finite Element Analysis, Fatigue specimen.

1.INTRODUCTION

Components in engineering applications often have discontinuities and abrupt change in cross sections which may be present owing to their functional requirements like oil holes, grooves, keyways etc. This would result in localization of high stresses when these components are subjected to loading. Situation becomes more hazardous to the material if the loading is not static and varying in magnitude with time. This fatigue phenomenon reduces the resistance of the material under fluctuating stresses. Fatigue can be defined as a failure taking place by the formation and growth of cracks due to repeated stresses. Fatigue design is considered to be complex as the failure sometimes occurs abruptly without any indication about the initiation of the failure. It is evident from experience that, around 80% of structural failures is due to insufficient fatigue design. Prodigious work is done in the field of fatigue design but there is still lot of scope in this area. C.S.Yen et.al. reviewed lot on the literature and concluded that, the fatigue notch-sensitivity of a metal member depends upon three different factors namely, the basic material characteristics, the degree of material homogeneity, and the geometry of the member [1]. M.Makkonnen showed that when the notch gets sharper, the magnitude of the plastic portion of the strain starts to play an important role in the fatigue crack initiation, and the fatigue limit is lower than that prediction is by statistical and geometric size effects. In those cases, fatigue limits should be arrived, by assuming the notch to be an initial crack with the notch depth being considered as the depth of the crack and the fatigue limit is computed to this crack by banking upon linear elastic fracture mechanics and the stress intensity factor range threshold [2]. Yoshiaki Akiniwa et. al. demonstrated that, for specimens with circumferential notch, fatigue fracture starts from the surface or very near the surface. The slip deformation is often responsible for the crack initiation in high cycle and very high cycle regimes [3]. A.J.McEvily et.al. proved that, for holes of radii less than 1 mm in the steel investigated, the notch fatigue factor, K_F, is dependent upon crack closure and for holes of radii in the range of 1–5 mm in the steel investigated; the analysis indicates that K_F is constant and dependent upon the ratio σ_{max}/σ_y [4]. M. Zehsaz et.al. showed that the volumetric approach gives good results in predicting the fatigue life of the notched specimens. The effect of notch radius for different notched specimens was investigated to observe the stress concentration factor, notch strength reduction factor, and fatigue life of the specimens [5]. G.H.Majzoobi et.al. demonstrated that notch geometry has profound effect on fatigue life of materials. For high strength steel this reduction is roughly about 50%. For low strength -steel alloy, however, the reduction depends on fatigue life and varies from 20% for low cycle fatigue tests up to 75% for high cycles fatigue tests. The maximum and minimum fatigue life reduction occurs for the V-shape and U-shape notches, respectively [6]. Baohua Nie et. al. concluded that fatigue life improves with the increase in the crack initiation depth. The scatter of the fatigue property should be carefully considered in fatigue design [7]. M.L. Aggarwal et. al. developed a numerical model using stress approach to predict the fatigue life of a shot-peened mechanical component [8].

The current research work aims at finding the pressure that the selected material can sustain in tension at a fixed stress level and fatigue life. Specimens are fabricated with different geometries of notches on them. The width, depth and central angle of the notch are varied and the fatigue life is found out using finite element method [9, 10]. The design of experiments using Taguchi L9 orthogonal array is selected to know the impact of each parameter on the pressure bearing capacity and thereby on the fatigue life. Numerous finite element method runs are conducted in an iterative approach using ANSYS 18.1 to get the target values of stress and fatigue life.

2. MATERIALS AND METHODS

2.1. Material

BS 970-4 grade 349S52 austenitic stainless steel is selected for the analysis due to its wide application in manufacturing industry. Table 1 depicts the chemical composition and Table 2 shows the mechanical properties of the above said material.

Table 1: Chemical Composition of BS 970-4 grade 349S52

Fe	Cr	Mn	Ni	C	N	P	Si	S
62.09- 66.99	20 - 22	8 - 10	3.25 - 4.5	0.48 - 0.58	0.38 - 0.5	0 - 0.04	0 - 0.25	0 - 0.04

Table 2: Mechanical Properties of BS 970-4 grade 349S52

Poisson's Ratio	Tensile Yield Strength (MPa)	Hardness (BH)	Impact Strength (J)
0.3	622	444	14

2.2. Design Criterion

In a conservative fatigue design approach proposed by Soderberg, yield point is the criterion for a ductile material to undergo failure. In the current research, fully reversible loading is applied for all the cases and Soderberg's method of stress-life approach is undertaken. The specimen for fatigue testing is varied in terms of the geometry by creating different types of notches at the centre of the gauge length. The notch is varied in width, depth and the notch central angle (popularly known as notch length). A typical notch provided on the gauge length of the specimen is shown in Figure 1.

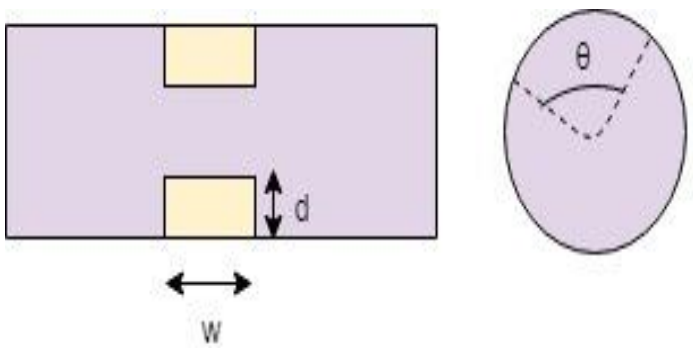


Figure 1: Typical Notch geometry

Following variations are chosen as parameters in the current investigation (Table 3).

Table 3: Parameters for fatigue testing in ANSYS 18.1

S.No	Parameter	Units	Range		
			Low	Medium	High
1	Width (w)	mm	1	1.25	1.5
2	Depth (d)	mm	1	1.25	1.5
3	Notch Central Angle (θ)	degrees	120°	240°	360°

2.3. Specimen Geometry

Fatigue specimen proposed by ASTM E606 standard is selected to perform the finite element method runs. The drawing of the specimen is depicted in Figure 2;

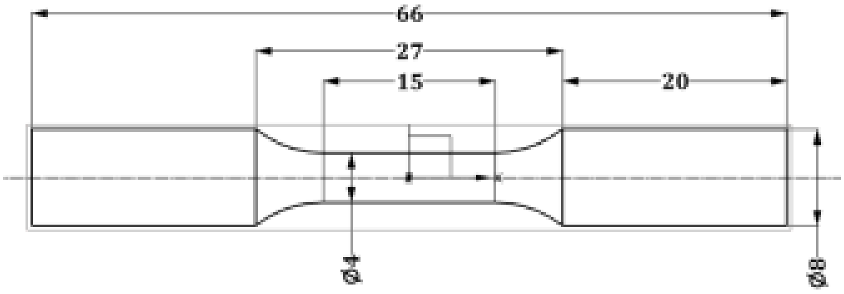


Figure 2: Drawing of the specimen

2.4. 3-D Model for Experimentation

The 3-D model of the specimen is modelled using Creo 3.0 and the schematic view of the same is shown in Figure 3.

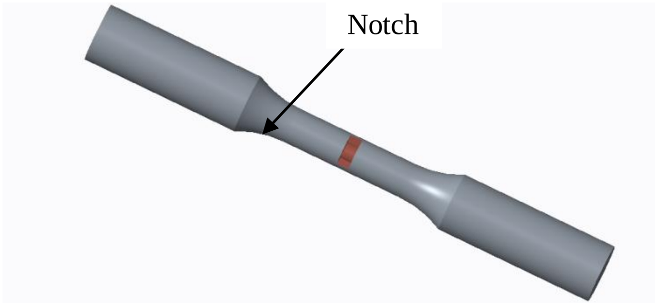


Figure 3: 3-D model of the specimen

The Creo 3-D model is exported as IGES file and imported to Ansys 18.1. Discretization is performed for all specimens using Ansys 18.1 mesh tool with refinement level of 3 so that there are around 6150 elements on an average for all specimens.

2.5. Taguchi Method

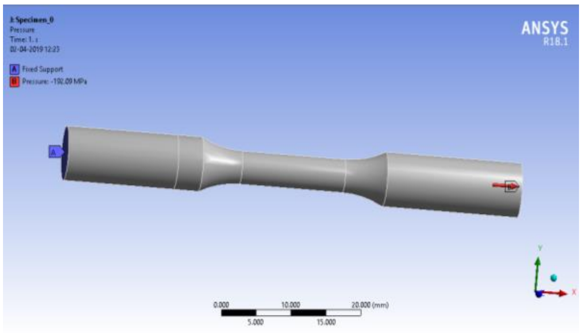
Taguchi philosophy in the design of a product or process for its robustness is done by adopting the design parameters in order to achieve the target. The orthogonal array (OA) serves the purpose and L9 OA is undertaken for the current investigation (Table 4) [11, 12] and the analysis was carried out using Minitab 17.

Table 4: Taguchi orthogonal array (L9)

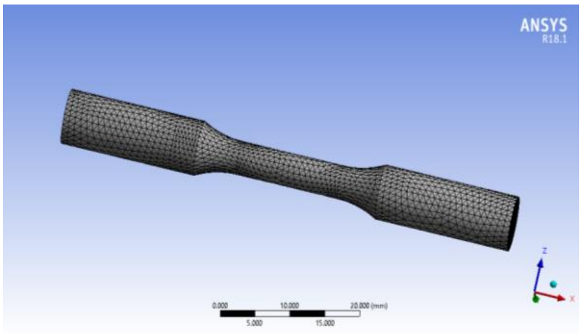
S.No	Width (mm)	Depth (mm)	Notch Central Angle (degrees)
1	1	1	120^o
2	1	1.25	240^o
3	1	1.5	360^o
4	1.25	1	120^o
5	1.25	1.25	240^o
6	1.25	1.5	360^o
7	1.5	1	120^o
8	1.5	1.25	240^o
9	1.5	1.5	360^o

3. RESULTS & ANALYSIS

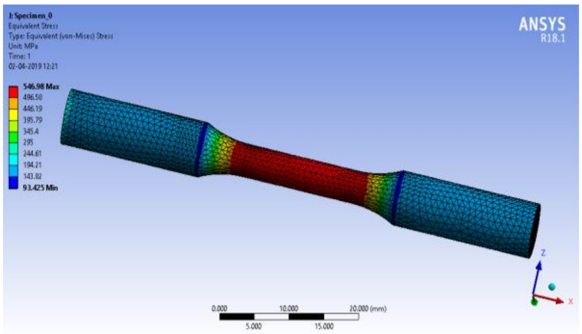
The results of the specimen without any notch are shown in the Figure 4 and tabulated in Table 5.



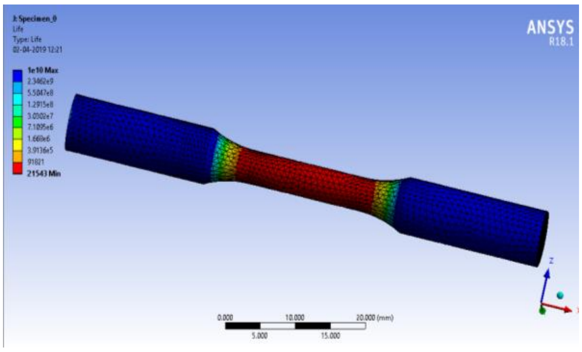
(a) Boundary conditions



(b) Finite Element Model after meshing



(c) Von-Mises stress



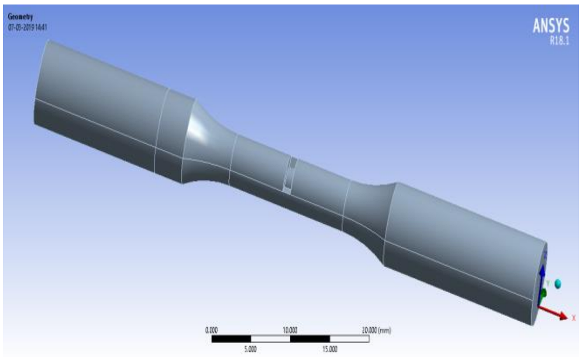
(d) Fatigue Life

Figure 4: FE Analysis of specimen without notch

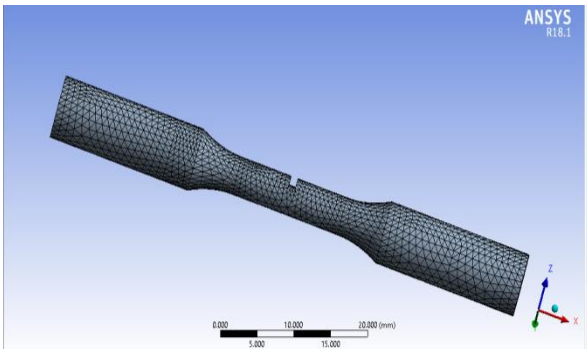
Table 5: Fatigue results of specimen without notch on it

Fatigue Life (cycles)	Von-Mises (Equivalent) Stress (MPa)	Pressure Required (MPa)
21543	546.98	192.09

The results of the specimen with notch (Run 1 in Table 6) are depicted in Figure 5 and Table 6 encapsulates the results for all the scenarios.



3-D model imported into ANSYS 18.1



Finite element model after meshing

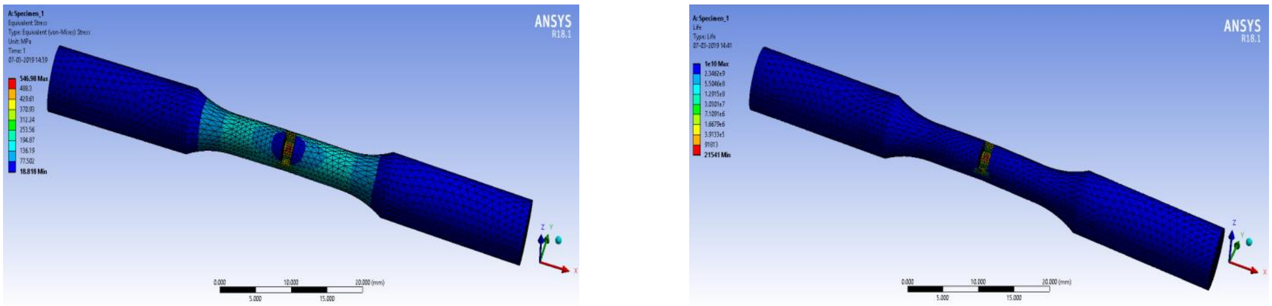


Figure 5: FE analysis of specimen mentioned in Run 1 (1-1-120°)

The experimentation is carried out on different specimens (Table 4) in Ansys 18.1 with an objective to attain the pressure to be applied as the boundary condition for a fixed value of stress induced (546MPa) at which the fatigue life is around 21500 cycles. The stress induced is fixed at about 546MPa (88% of the yield point) so that the loading condition is aggressive. Safe design as per Soderberg's theory shall be below the yield point [13, 14].

Table 6: FEA results for L9 OA matrix

Run No	Width (mm)	Depth (mm)	Notch Central Angle (degrees)	Fatigue Life (cycles)	Equivalent Stress (MPa)	Pressure Required (MPa)
1	1	1	120	21541	546.98	51.01
2	1	1.25	240	21575	546.92	20.84
3	1	1.5	360	21553	546.96	15
4	1.25	1	120	21582	546.91	47.21
5	1.25	1.25	240	21592	546.89	15.266
6	1.25	1.5	360	21555	546.96	17.18
7	1.5	1	120	21612	546.85	57.25
8	1.5	1.25	240	21589	546.89	23.12
9	1.5	1.5	360	21593	546.89	17.532

It can be seen from the Table 6 that, as the width of the notch is increasing, the pressure bearing capacity is also increasing while, as the depth and central angle of the notch are increasing, the trend is reversed. Variation of pressure with the parameters undertaken is plotted as shown in Figure 6. Optimum parameter selection is undertaken on the basis of higher the better criteria as suggested by Taguchi design of experiments [15]. It is found that, when the width is at higher level, depth and notch central angle are at lower levels, the optimal parameters are arrived at, in the current investigation.

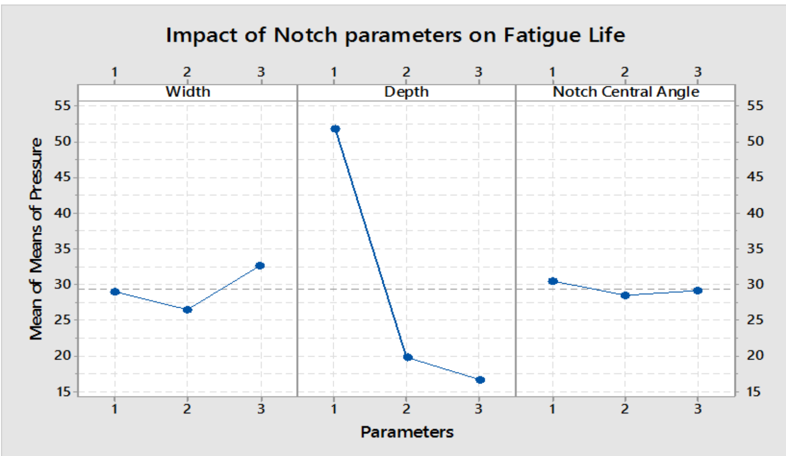


Figure 6: Variation of Pressure w.r.t. the parameters

The analysis of variance was carried out and the results are tabulated in Table 7. It is found that the depth of notch is predominantly playing a major role in governing the applied pressure for the specified stress and fatigue life to an extent of 96.29%, while the notch central angle is affecting insignificantly to an extent of only 0.24%. From literature it is evident that, the fatigue life reduces with the increase of crack depth for BSL65 aluminium alloy [16]. The results obtained for the material undertaken in the current research are in tandem with the above findings.

Table 7: Analysis of Variance

Source	Degrees of Freedom	Sum of Squares	Mean Square	Fisher Ratio	Contribution (%)
Width	2	56.31	28.16	2.18	2.38
Depth	2	2282.02	1141.01	88.34	96.29
Notch Central Angle	2	5.66	2.83	0.22	0.24
Error	**2**	**25.83**	**12.92**		
Total	**8**	**2369.83**			

4. CONCLUSIONS

From the above research, it can be concluded that the pressure bearing capacity of the material reduced by around four times to thirteen times due to the introduction of the notches specified in Table 4. It can also be concluded that the depth of notch has major role to play to an extent of 96.29% in deterioration of pressure bearing capacity and the notch central angle is affecting insignificantly to an extent of only 0.24%.

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