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Study of Stress Concentration Factor in Crankshaft by FEM

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ABSTRACT: Stress concentration factor is one of the key factors in estimating the life of a crankshaft. The accuracy life estimated depends on the accuracy with which the stress concentration is determined. Present work aims at determining the stress concentration factors in crankshaft subjected to bending using finite element method (FEM). A parametric solid model develops in SOLID EDGE 3D solid model software and a computer program using ANSYS Parametric Design Language (APDL) in ANSYS Finite Element Analysis (FEA) software augment the analysis. The results obtained are validated by comparing the experimental results obtained by Arai.

Keywords: ANSYS, Stress concentration

I. INTRODUCTION

Design analysis techniques are extending in two directions. Firstly, they are becoming more sophisticated. The conflicting demands of modern engine design, in particular the quest for high standards of refinement and high ratings, without penalties in other areas, requires more in-depth analysis than hitherto. Less sweeping assumptions must be made and calculations more nearly approximating the actual engine operating conditions must be carried out. Secondly, in addition to this increased level of sophistication, the breadth of design and analysis effort is increasing there is an increasing disparity of design variants, with a compressed time scale for the design and development of each. This results in a demand for more rapid solutions, which directly opposes the demand for more in-depth analysis. These trends are particularly apparent for the crankshaft, which consumes more analysis effort than any other single engine component. Many highly sophisticatedcrankshaft analysis methods have been reported in the past.

Table 1: Parameters of Crankshaft.

Sr.No	Parameters	Range
1	Web Thickness to Diameter of crankpin(t/d)	0.36, 0.40, 0.44, 0.48, 0.52, 0.56
2	Fillet Radius to Diameter of crankpin (r/d)	0.0625, 0.08, 0.1, 0.12
3	Centre distance between crankpin and journal to Diameter of crankpin (s/d)	-0.3, -0.2, -0.1, 0.1, 0.2, 0.3

This study discusses a complete range of tools required to satisfy the whole spectrum of demand. The scope of this work is to investigate the effect of web thickness, fillet radius and distance between the centers of crankpin and journal of crankshaft subjected to bending. The details of the range of the parameters are specified as in below table

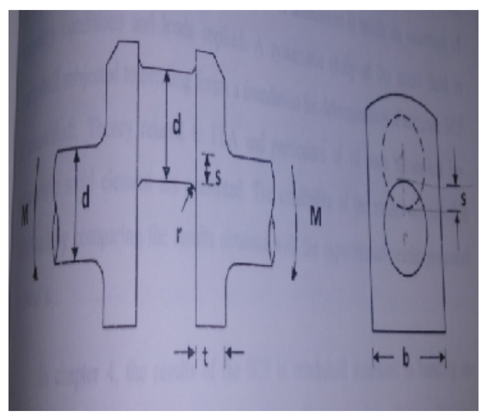

Fig. 1. Parameters of Crankshaft.

II. LITERATURE SURVEY

The maximum stresses in the fillets of the pin and journal of a series of crankshafts in bending were determined by use of the mechanical strain gauge method by Aria[1]. Design parameters were systematically varied in a comprehensive manner involving 178 tests. The stress concentration factor is defined as $\sigma_{max}/\sigma_{min}$,
Where
σ_{max}=M(d/2)/I = M/(πd^3/32)

Stahl [2] and Pfender [3], Amedick and Sonntag made use of extensometers to determine stresses in crankshafts. Here, transducers are temporarily attached to the specimen surface. Under loading, the change in length between two points in the surface is measured and this is then converted to a stress. Extensometers tend to be bulky and are not ideally suited to measuring stresses in small fillets.

Fessler & Sood [4] utilized the technique of photo-elasticity. Frozen stress photo-elasticity requires that a three-dimensional epoxy model, with the desired physical and optical properties for photo-elasticity analysis, be made of the crank throw. The model is then loaded and subjected to a stress-freezing process, after which slices through various planes of interest are removed. Slices are examined in a transmission polar scope and the complete stress intensity in the plane of the slice can be obtained. The results from a carefully executed test can be considered to be the most precise and informative of the four experimental methods because continuous stress intensity is obtained. However, photo-elasticity requires that the model is sliced up after loading, there by necessitating a new model for each test.Frozen stress photo-elasticity is therefore not an ideal technique for investing a wide range of model parameters.

G.C. Volcy [5] talks of Bureu Vertas contribution to the augment put to IACS (International of Classification Societies) for the adoption of the rules proposed by CIMAC. Bureu Veritas carried out FE analysis of a crankthrow and concluded that FE calculations were laborious, expensive and the results were disappointing.

III. NEED FOR PRESENT STUDY

Significant studies are reported in the literature dealing with analysis of crankshaft. However a limited number of works have been carried out to investigate the SCF in crankshafts in simplest way. Therefore it is necessary to develop a simple and straight forward method of the determining the SCF in crankshafts. The present study focuses on the geometric finite element analysis of the following crankshafts parameters, which are presented in our study such as r/d ratio, s/d ratio, t/d ratio

IV. MODEL PREPARATION AND IMPORTING IN ANSYS

The various 3D half model of crankshaft are created in solid edge software with the dimensions given in table.2 to ANSYS environment through IGES file format.

A. Meshing

The geometry of the model in finite element analysis is reoriented by the collection of finite elements used, known as a mesh. Most automatic meshes create tetrahedral elements in solid volumes. Tetrahedral elements are less accurate than the brick elements. But for structural dynamic analysis tetrahedral mesh is sufficient. Our geometric model is meshed with SOLID92 element.

Table 2: Parameters.

SL.NO	R	S	t	b
1	1.5	-7.5	9	33.25
2	1.5	-5	9	33.25
3	1.5	-2.5	9	33.25
4	1.5	2.5	9	33.25
5	1.5	5	9	33.25
6	1.5	7.5	9	33.25
7	2	-7.5	9	33.25
8	2	-5	9	33.25
9	2	-2.5	9	33.25
10	2	2.5	9	33.25
11	2	5	9	33.25
12	2	7.5	9	33.25
13	2.5	-7.5	9	33.25
14	2.5	-5	9	33.25
15	2.5	-2.5	9	33.25
16	2.5	2.5	9	33.25
17	2.5	5	9	33.25
18	2.5	7.5	9	33.25

B. Particulars of Element

The element has quadratic displacement behavior and is well suited model irregular meshes. The element is defined by ten nodes having three degrees of freedom at each node: translations in the nodal x, y and z directions. The element also has plasticity, creep, swelling, stress stiffening, large deflection, and large strain capabilities.

The assumptions of SOLID92 element are,

-The element must not have a zero volume.

An edge with a removed mid side node implies that the displacement varies linearly, rather than parabolic ally, along that edge.

The restrictions of SOLID92 element are,

-The damp material property is not allowed.

-Fluence body loads are not applicable.

-The only special feature allowed is stress stiffening.

C. Material Properties

The linear elastic isotropic material is assigned with young's modulus 2.1*10^5 N/mm^2 and poison's ratio is equal to 0.29.

D. Boundary Conditions

The model is meshed with solid 92 elements. Symmetric boundary conditions are applied on the mid plain of crankpin and the moment of 12500 N-mm is applied as shown in fig and solved to get maximum stress induced represented in fig

Fig. 2. Meshed Model with Boundary Conditions.

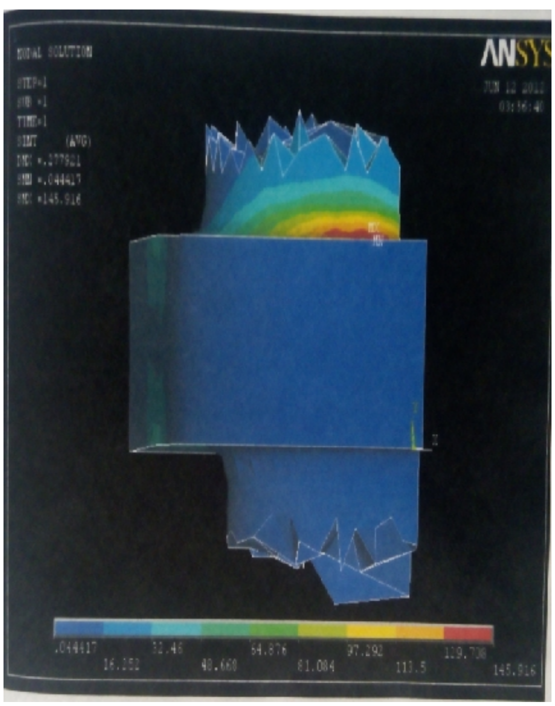

Fig. 3. Stress Intensity in Model.

E. Convergence Test

An analysis with an initial smart mesh size of 7 is performed first and then reanalyzed by decreasing the size. The solutions are compared. If the results are close to each other then the initial mesh configuration isconsidered to be adequate. If there is substantial difference between thetwo, the analysis should continue with a more-refined mesh and a subsequent comparison until convergence is established.

In fig.4 maximum stress Vs smart mesh size for a typical case are plotted. In this case it is observed that the finite element results of maximum stress converge at smart size equal to 2. The percentage of difference between this trial and the immediate next trial is 0.8. Mesh size, maximum stress and percentage variation are given in Table.3

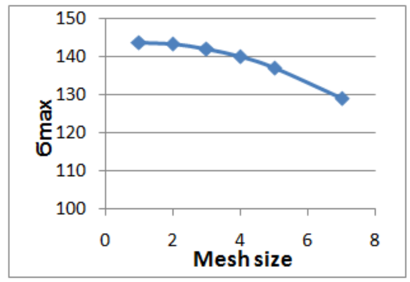

Fig. 4. Maximum Stress Vs Mesh Size.

Table 3.

Trial no	Mesh size	Ϭmax	variation
1	7	129	
2	5	137	5.8
3	4	140	2.1
4	3	142	1.4
5	2	143.2	0.8
6	1	143.6	0.3

RESULTS AND DISCUSSION

This section consists of four sub sections. The first section deals with calculation of nominal stress and stress concentration factor while second section deals with effect of t/d ratio with different s/d and r/d ratios on stress concentration factor. In the third section effect of r/d with different s/d ratios on stress concentration factor are dealt. In the fourth section effect of s/d with different r/d ratios on stress concentration factor are dealt.

A. Nominal Bending Stress

The nominal bending stress can be completed using the relation given in equation (1)

$$\sigma_{nom} = M(d/2)/(\pi d^4/64) \qquad (1)$$

Where,

M- Bending moment = Force*Distance

d- Diameter = 25 mm

σ_{nom}= Nominal bending stress

y= d/2

A force of 1000 N is considerfor analysis. It is applied at nodes on the periphery of the circular (diameter= 25 mm) section. This amounts to applying a moment of 25 kN (2*Force* Radius) as shown in fig.3

Therefore,

σ_{nom}= 2500*(25/2)/(π*25^4/64)

= 16.3 (approx)

Stress concentration factor is given by,

$K_t = \sigma_{max} / \sigma_{nom}$

Where,

σ_{max}= maximum stress induced in the crankshaft due to bending load

B. Effect of Stress Concentration Factor Vs Various Parameters

Effect of t/d ratio on stress concentration factor . When t/d ratio with various r/d and s/d were plotted against stress concentration factor for dimensions given in table.4 the result obtained are shown in table.4 and represented in fig.5

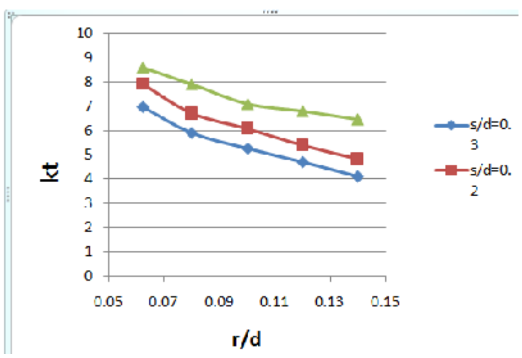

Fig. 5. K_t Vs t/d.

Table 4.

Sr. no	t/d	kt for r/d= 0.06 s/d=-0.1	kt for r/d= 0.06 s/d=-0.1	kt for r/d= 0.06 s/d= 0.3	kt for r/d= 0.06 s/d= 0.3	kt for r/d= 0.1 s/d= 0.3	kt for r/d= 0.1 s/d= 0.3
1	0.3	8.94	8.6	6.86	6.7	5.15	5.05
2	0.4	7.51	7.35	6.06	5.95	4.59	4.5
3	0.4	6.33	6.35	5.36	5.27	4.15	4.04
4	0.4	5.5	5.6	4.84	4.75	3.71	3.65
5	0.5	4.87	5	4.38	4.35	3.38	3.3
6	0.5	4.41	4.5	4.11	3.95	3.13	3

From fig.5 it is observed that the SCF increases as the thickness of the web decreases. SCF increases as r/d ratio decreases. SCF increases as s/d decreases.

Effect of r/d ratio with different s/d ratio on stress concentration factor. When r/d ratio with various s/d were plotted against stress concentration factor for dimensions given in table.5the result obtained are shown in table and represented in fig.6

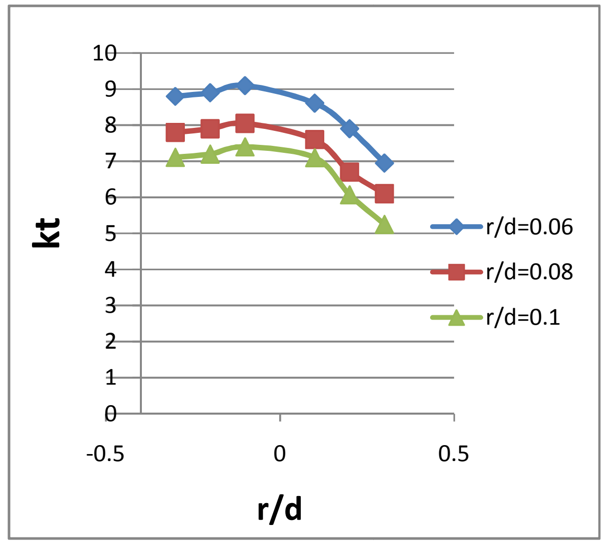

Fig. 6. Kt Vs r/d.

Table 5.

Sr. no	t/d	kt for r/d=0.06 s/d=-0.1	kt for r/d=0.06 s/d=-0.1	kt for r/d=0.06 s/d=0.3
1	0.3	8.94	8.6	6.86
2	0.4	7.51	7.35	6.06
3	0.4	6.33	6.35	5.36
4	0.4	5.5	5.6	4.84
5	0.5	4.87	5	4.38

From fig.6 it is observed that the SCF increases as the fillet radius decreases. SCF increases as s/d ratio decreases.

Effect of s/d ratio with different r/d ratios on stress concentration factor. When s/d ratio with various r/d were plotted against stress concentration factor for dimensions given in table.6 the result obtained are shown in table.8 and represented in fig.7.

From fig.6 it was observed that effect of s/d with different r/d ratios on stress concentration factor was nonlinear and varying.

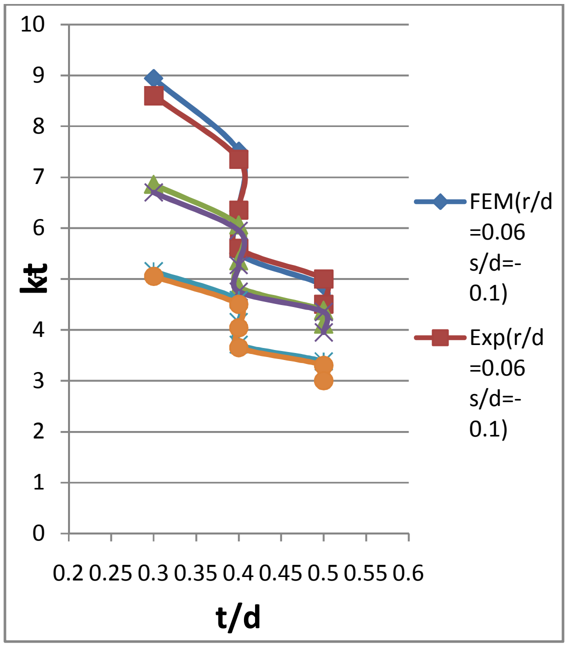

Fig. 7. Kt Vs s/d.

Table 6.

Sr. no	s/d	kt for r/d=0.06	kt for r/d=0.08	kt for r/d=0.1
1	-0.3	8.8	7.8	7.11
2	-0.2	8.9	7.9	7.2
3	-0.1	9.1	8.05	7.4
4	0.1	8.61	7.6	7.1
5	0.2	7.9	6.7	6.07

CONCLUSION AND FUTURE SCOPE

The maximum stresses are found at the fillet area of the crankshaft. The raise in the stress is due to abrupt change in the cross sectional area between the web and crank. It is true for the junction between web and journal.

Parametric half 3D solid models of crankshafts are created to automate the process of creating geometry model of crank shaft for various parameters. This FEA Analysis was carried out to determine concentration factor. The result of FEA are validated by comparing the result of the present work with the expected results available in literature conducted by Arai.

REFERENCES

[1]. Arai, J., “The bending stress concentration factor of solid crankshaft “, *Bulletin of JSME,* Vol. **8**, 1965, p.322

[2]. Staul, G., “Der Einfluβ der form auf di Spannungen in Kurbelwellen”. Konstruction, Vol.10, Heft 2, 1958

[3]. Pfender, M., Amedick, E., and Sonntag.,“Einflu β der Formgebung auf die soannungsverteilung in kurbelkropfungen”, M.T.Z., Vol.**27**, 1966,p.225

[4]. Fessler, H,.andsood, V.K., “Stress distribution in some diesel engine crankshaft”, Trans. ASME Diesel and engine Power Division conference, Toronto, Paper No.71-DGP-1,1971

[5]. Volcy, G.C., Discussion of paper by Donath, G., Seidemann, H., “Design of Marine Diesel Engine Crankshafts: comparision of measured and calculated stresses using the proposed CIMAC rules”, *Trans. Of the Institute of Marine Engineers*, Vol. **97**, Paper No.25 1985, p.14