Design & Analysis of Crankshaft Bending Test Rig for Actual Engine Condition

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***Abstract-* The crankshaft is one of the most critically loaded components as it experiences cyclic loads in the form of bending and torsion during its service life. Its failure will cause serious damage to the engine so its reliability verification must be performed. This paper deals with fatigue strength assessment of crankshaft in automobile industry. The topic was chosen because of increasing interest in higher payloads, lower weight, higher efficiency and shorter load cycles in crankshaft equipment.**

The aim of this work is to design bending test fixture for crankshaft for load ratio R=-0.2 which is an actual engine condition. This paper consists of design of test fixture, 3-D model generation of test fixture and stress analysis of crankshaft & test fixture using CAE tool in order to minimize the time during physical test.

Keywords- Crankshaft, Fatigue, Actual Engine Condition, Bending Test Rig, FEA

I. INTRODUCTION

Crankshaft is a large component with a complex geometry in the engine, which converts the reciprocating displacement of the piston to a rotary motion with a four link mechanism. Since the crankshaft experiences a large number of load cycles during its service life, fatigue performance and durability of this component has to be considered in the design process. Design developments have always been an important issue in the crankshaft production industry, in order to manufacture a less expensive component with the minimum weight possible and proper fatigue strength and other functional requirements. These improvements results in lighter and smaller engines with better fuel efficiency and higher power output.

The rapid developments in numerical simulation techniques, faster computing ability, and greater Memory capacity, are allowing engineers to create, and test industrial equipment in virtual environments. Through finite element analysis (FEA), these sophisticated simulations provide Valuable information for designing and that are easy to manufacture, and which make the most economic use of developing new products, as well as perfecting existing Ones. Manufacturers have found this method eminently useful, as it helps them to achieve better productivity at a lower cost per unit, and develop engineering components their materials Physical testing is an important part of any component validation process. However, physical tests require the component. Since the test is carried out after the component has been designed and the manufacturing process established the results can only have a limited impact on the immediate design and manufacturing process. It is incumbent on a supplier to develop an expertise in CAE where the feasibility of the component under the bending fatigue test setup can be estimated with confidence before the test setup is finalized and long before the manufacturing process of bending fatigue test fixture design has been established.

II. LITERATURE REVIEW

A. Analytical Models

Dejan Ninic and Hugh L. Stark [2] presented a method for multiaxial fatigue damage function, fatigue performance and durability of this component has to be considered in the design process. In detail the fatigue testing procedure of six cylinder diesel engine crankshaft & load cycles during its service life, fatigue performance, analysis etc was done by Paswan and Goel [6]. Do-Hyun Jung et.al [3]. In 2003 have described the method of increased quality of fatigue testing reliability prediction of the fatigue life of a crankshaft & bending test of crankshaft with load ratio R=-1.Also the analytical method for V-block three-point method & most conventional design, for a V- block is a 90 degree included angle was done by Eiki Okuyama et.al [4].

B. Finite Element Models

Chien W.Y.et.al In 2004[1] have described the method of the fatigue analysis of crankshaft sections under bending with consideration of residual stress with most widely used codes are finite element tools. In detail investigating torsional fatigue with a novel resonant testing fixture by using finite element model was done by Fabricio Tonon Joaquim et.al [5]. Yung-Li Lee et.al [7] presented a method for finite element analysis for the fatigue testing and analysis of crankshaft & bending test fixture at actual engine conditions.

III. METHODOLOGY TO ACHIEVE LOAD RATIO:

Fatigue is a localized damage process of a component produced by cyclic loading. It is the result of the cumulative process consisting of crack initiation, propagation, and final fracture of a component. During cyclic loading, localized plastic deformation may occur at the highest stress site. This plastic deformation induces permanent damage to the component and a crack develops. As the component experiences an increasing number of loading cycles, the length of the crack increases. After a certain number of cycles, the crack will cause the component to fail.

During fatigue testing, the test specimen is subjected to alternating loads until failure. The loads applied to the specimen are defined by either a constant stress range (σ_r) or constant stress amplitude (σ_a). The stress range is defined as the algebraic difference between the maximum stress (σ_{max}) and minimum stress (σ_{min}) in a cycle:

$$\sigma_r = \sigma_{max} - \sigma_{min}$$

The stress amplitude is equal to one-half of the stress range:

$$\sigma_a = \frac{\sigma_r}{2} = \frac{(\sigma_{max} - \sigma_{min})}{2}$$

Typically, for fatigue analysts, it is a convention to consider tensile stresses positive and compressive stresses negative.

The mean stress (σ_m) is defined as

$$\sigma_m = \frac{(\sigma_{max} + \sigma_{min})}{2}$$

Actual structural components are usually subjected to alternating loads with a required mean stress.

The stress ratio is defined as the ratio of minimum stress to maximum stress:

$$R = \frac{\sigma_{min}}{\sigma_{max}}$$

When load ratio R=-1 then tensile & compressive stresses are same as shown in fig. 3.1

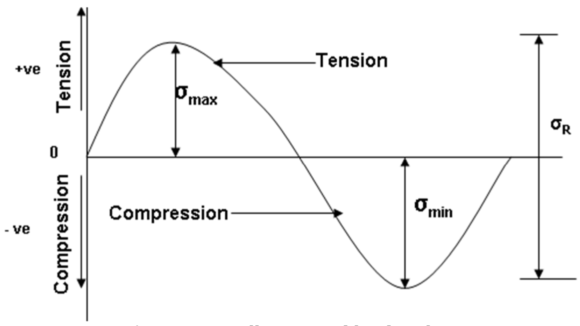


Figure 3.1: *Fully reversed load cycle*

But when load ratio R=-0.2 then stresses are as shown in fig. 3.2.

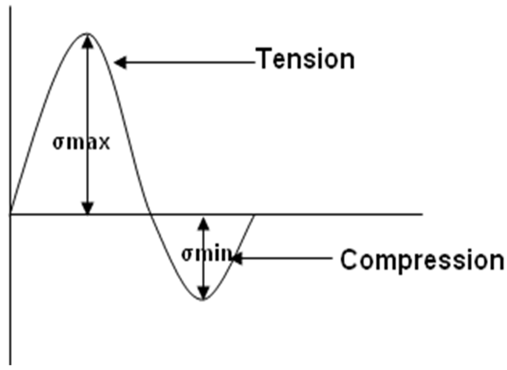


Figure 3.2*: Variable load cycle*

Suppose min. stress is Y & max. Stress is X then,

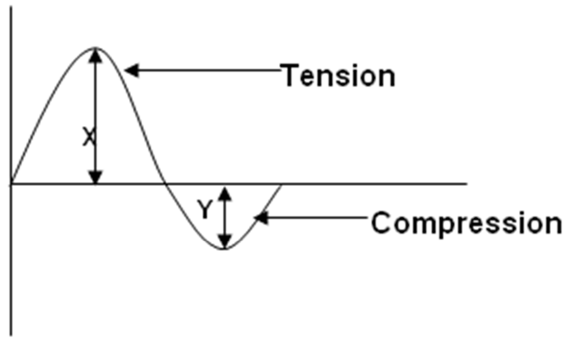


Figure 3.3: *Variable load cycle*

Load ratio=Y/X=-0.2

X=5Y

Range=X+Y=6Y

Load ratio calculation:

The maximum load amplitudes under service conditions for the crank pin radius of the crankshaft and the load ratio R were calculated based on the peak combustion pressure, the maximum continuous over speed and on the rotating and oscillating masses.

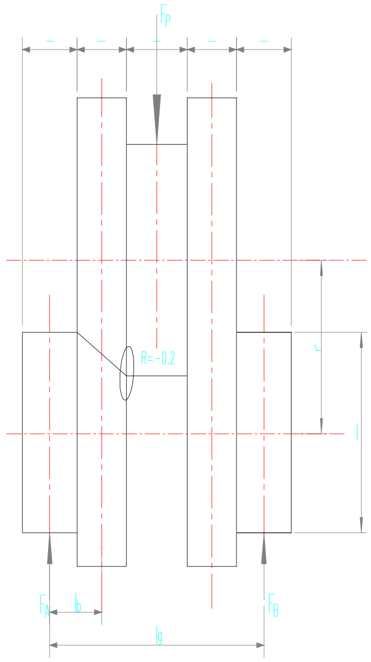


Figure.3.4: *Forces on crankshaft*

The formulas for the calculation are given bellow,

$$F_A = F_B = \frac{1}{2} * F_P$$

F_P: Con-rod load

- **Connecting rod tension load:**

$$Ft = F_{ozt} + F_{rot}$$

F_t: tension load

F_{osz}: Load due to oscillating mass.

F_{rot}: Load due to rotating mass

$$F_{rot} = m_{rot} * r * w^2 = m_{rot} * r * (\pi * n / 30)^2$$

M_{rot}: rotating mass

r: crank radius

n: speed in min^{-1}

$F_{osz} = m_{osz} * (1 + \lambda) * r * \omega^2 = m_{osz} * (1 + \lambda) * r * (\pi * n/30)^2$

m_{osz}: oscillating masses

λ: connecting rod ratio r/l

L= connecting rod length

F_{max}=-$F_t/2$

- **Connecting rod compression loads:**

F_G = load due to combustion

$F_G = p * \pi/4 * d^2$

P= peak firing pressure

d=Bore diameter

$F_c = F_G - (F_{rot} + F_{osz}) = F_G - F_t$

F_{min}=-$F_c/2$

$$Load\ Ratio = R = \frac{F_{\min}}{F_{\max}}$$

IV. FIXTURE MODEL DESIGN

We design bending test fixture for single through & double through crankshaft testing for load ratio=-0.2. In fixture we use the following part.

1. T Slotted table
2. Hydraulic actuator
3. V Block
4. Frame
5. Actuator plate

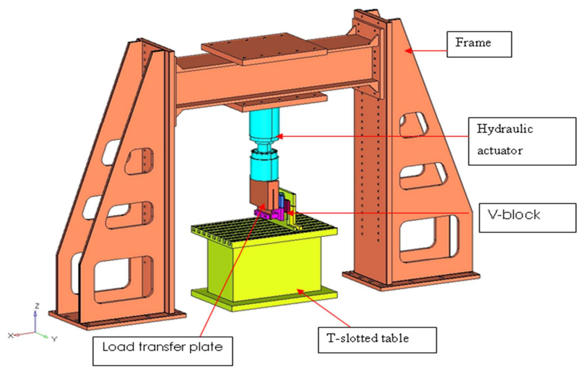


Fig.4.1 Fixture model

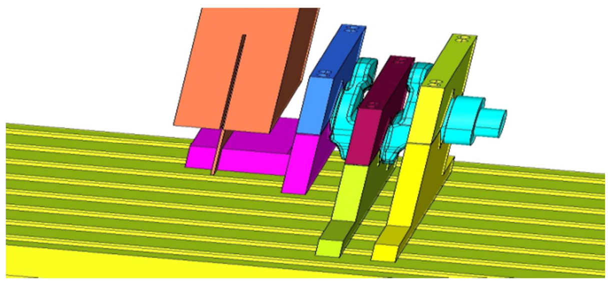


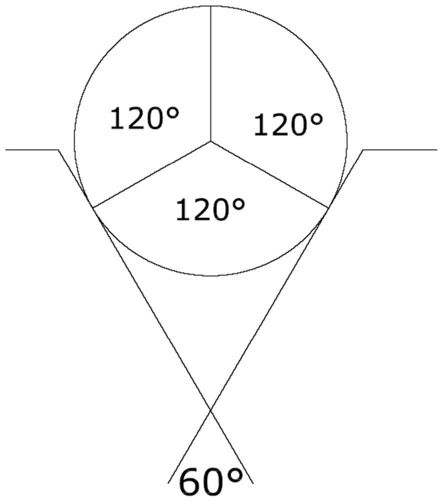
Fig.4.2 Detailed view of fixture

V-block:

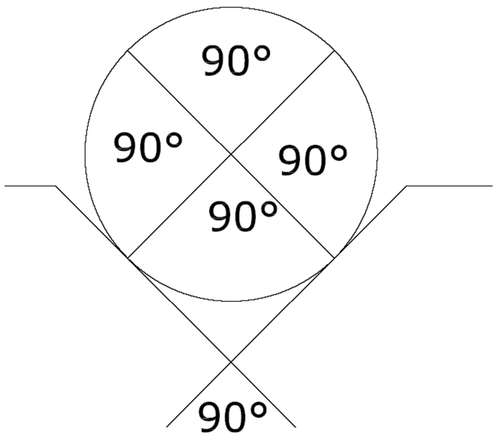
In V-location, work piece having circular profile is located by means of vee block. The v-block should be used correctly so that the variations in work piece size are not detrimental to location. Vees can be used for both locating & clamping a work piece.

A monolithic kinematic vee block is the most widely used design. There are several variations possible. The included angle between the two plane prismatic surfaces can be varied to achieve a desired result.

A 60 degree included angle of the prismatic faces will yield a 120 degree angle of contact with the ball. This angle will give the very best repeatability, but it will have very low load carrying capacity, due to the very high vector forces that result from this steep angle. Any angle steeper than 60 degrees will start to cause wedging and sticking of the ball in the vee.

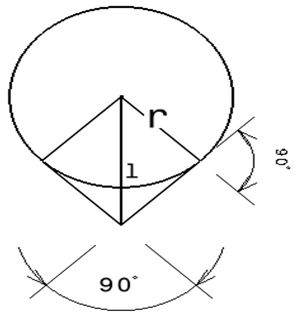


The most conventional design, for a vee block is a 90 degree included angle. This angle will give good location accuracy, and reasonable load carrying capacity.



So we take the angle for v-block is 90 degree.

When we decide depth of V-block we use the formula,



Suppose r is radius of crank pin or journal which will clamp in v-block then depth of V-block is,

L=r/sin45

Now we are clamping crank pin having diameter 100mm.

L=50/sin45

L=70.71mm.

And we clamping journal having diameter 120mm.

L=60/sin45

L=84.85

Now to give some gap between upper and lower V-block and also to use standard V-block we had taken depth of V- block for clamping crankpin as 65mm. and fo

V. FINITE ELEMENT ANALYSIS

The bending test set-up was modeled in CATIA V5R16 and meshed in the Hypermesh- 9.0 software and solved in ANSYS-10.0 with tetrahedron elements. The stress concentration areas in the fillets are fine meshed with minimum four rows of elements.

A. Crankshaft details under testing

Type: Six Cylinder Crankshaft

- Weight of Single Throw Crankshaft: 21.5 Kg

B. Material Properties

TABLE I

MATERIAL PROPERTIES

Yield Strength	1150MPa
Ultimate Tensile Strength	1400MPa
Poissons Ratio	0.3
Modulus of Elasticity	2.1 e5 MPa
Hardness	170HB

VI. RESULT & DISCUSSION

Then, for load ratio =R=-0.2

Minimum bending moment= $M_{b,\ min}$ =3000Nm.

Maximum bending moment= $M_{b, max}$ =15000Nm.

A.Tensile Load Step

$M_{b, max}$=F*L

15000=F* 0.210

F=71.43kN.

Constraint and load applied to model is as shown in fig.6.1 below.

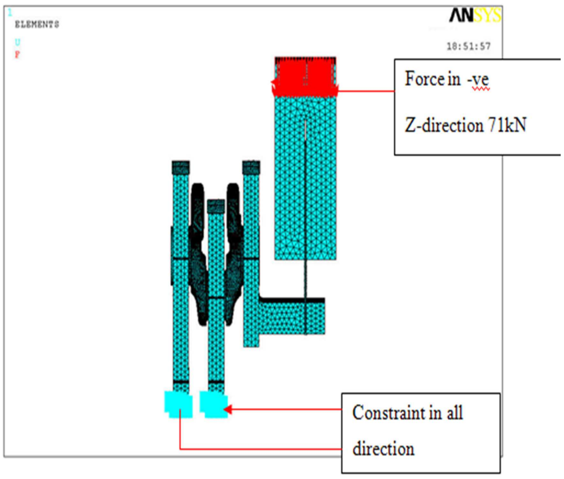


Fig.6.1 Boundary condition For tensile load

We give the contact in the area shown in fig.6.2

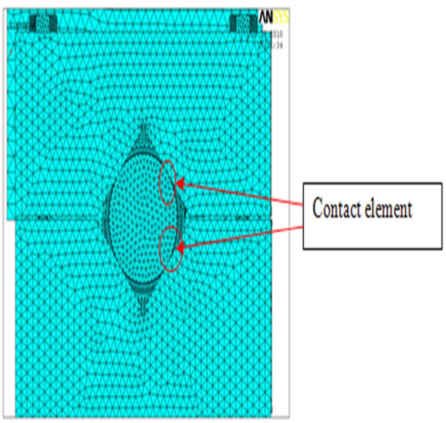


Fig.6.2 Contact element between V-block & crankshaft

Due to contact element these problem goes in nonlinear analysis and As we know that Non-linear analysis take no. of iteration to converge force value so force convergence criteria for these force is shown in graph below.

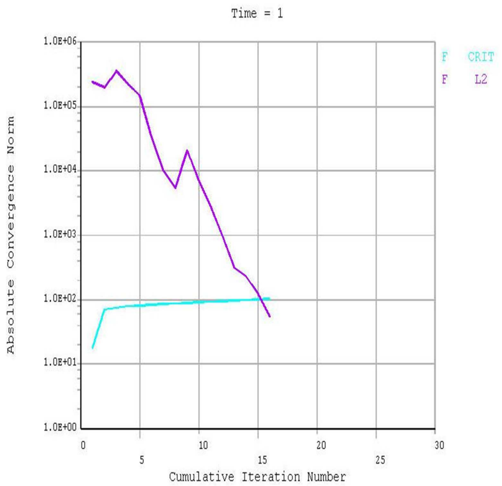


Fig.6.3 Force convergence criteria

This problem takes 16 iterations to converge force value.

Now in tension test we observe 1^{st} principle stress and 1^{st} principle strain.

1. *Crankshaft:* - When we applying load our main purpose is to introduce high stress in crankshaft pin fillet. A stress in crankshaft is as shown in fig.6.4

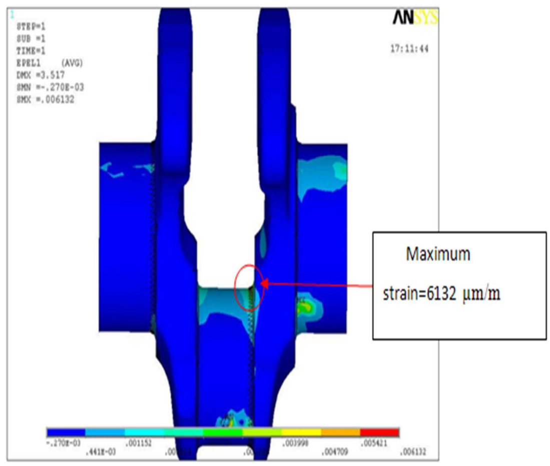


Fig.6.4 Tensile strain in crankshaft

The maximum strain in the crankshaft is as shown in fig. We also see that maximum strain in these test is at the pin fillet is about 6132 µm/m.

2. *V-block*

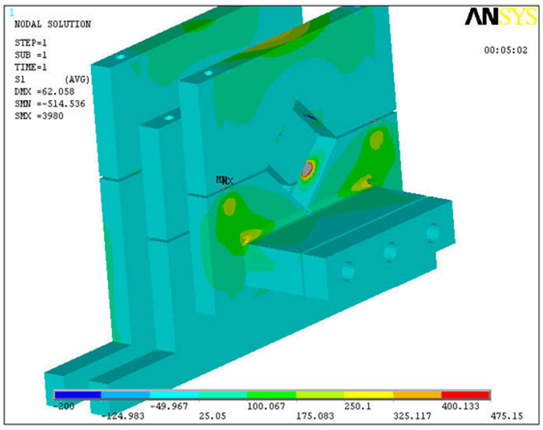


Fig.6.57 First principle stresses in V-block

We are using SN8 material for v-block the yield strength of this material is about 600Mpa. In our design maximum stress in V-block is about 475MPa which is attached to load transfer plate because bending moment in this region is maximum. Therefore our design is safe.

3. Load transfer plate

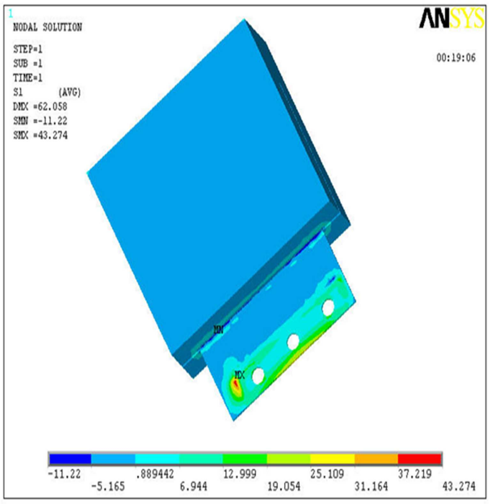


Fig.6.6 First principle stresses in load transfer plate

Material used for load transfer plate is alloyed steel is having yield strength 575Mpa. We see that stresses in load transfer plate are much below the yield strength.

Before completion of this analysis Bharat Forge test same crankshaft outside the company lab due to unavailability of instruments for this test.

Now to validate our CAE result we are comparing with this actual testing result.

TABLE II

COMPARISON OF PHYSICAL & FEA RESULTS

FOR BENDING MOMENT 15000 N-M

	Physical Test Value	FEA Values	Variation (%)
Stress (MPa)	1171.96	1139	2.81%
Strain(*mm*/m)	4996.18	4870	2.52%

Above table-II gives comparisons of physical test results and FEA results .We see that the difference between our results with experimental result variation is about 2-3%. This is good correlation with experimental result.

B Compressive Load Step

$M_{b,\ min}$=F*L

3000=F* 0.210

F=14kN.

The constraint and load applied to model is as shown in fig.6.7 below.

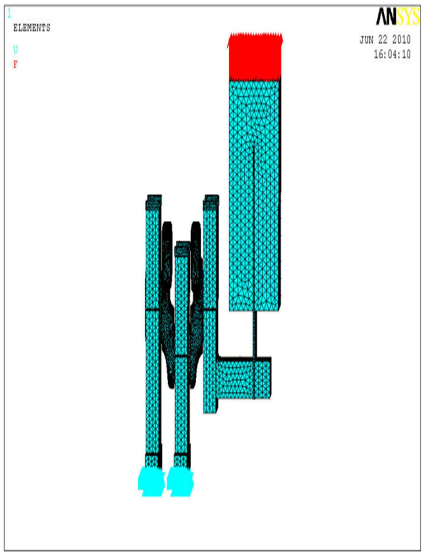


Fig.6.7 Boundary condition for compressive load

As we know that Non-linear analysis take no. of iteration to converge force value so force convergence criteria for these force is shown in graph below.

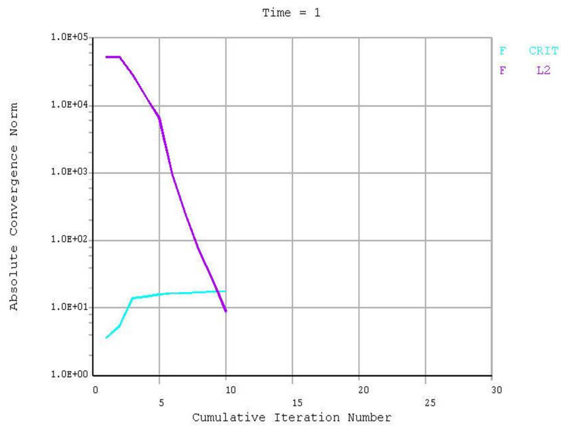


Fig.6.8 Force convergence criteria

This problem takes 9 iterations to converge force value. Now in compression test we 3rd principle stress and 3rd principle strain.

1. Crankshaft: - When we applying load our main purpose is to introduce compressive stress in crankshaft pin fillet. A stress in crankshaft is as shown in fig.6.9

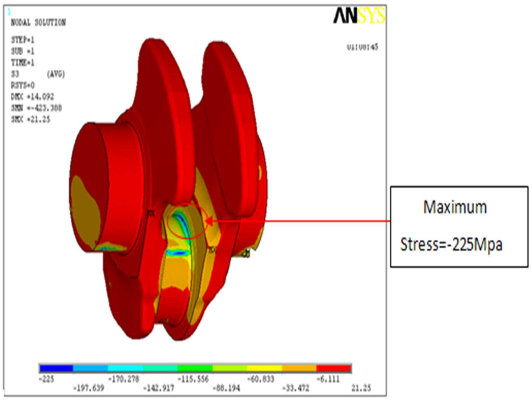


Fig.6.9 Compressive stresses in crankshaft

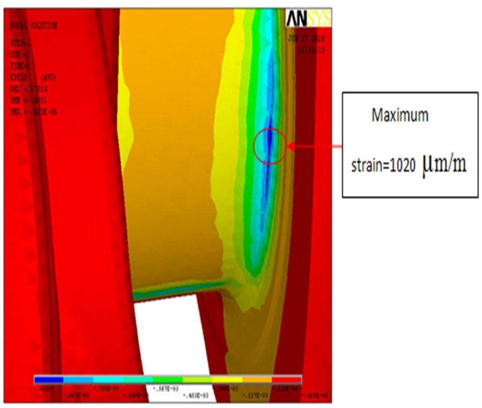


Fig.6.10 Compressive strain in crankshaft

The maximum strain in the crankshaft is as shown in fig. We also see that maximum strain in these test is at the pin fillet is about 1020 μm/m.

2. V-block

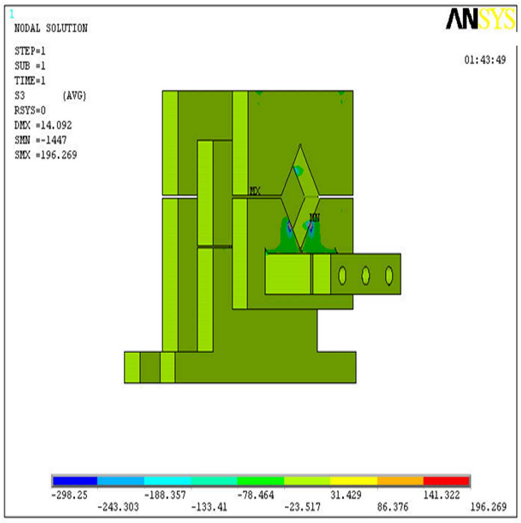


Fig.6.11 Third principle stresses in V-block

As shown in fig. above maximum stress is in that v-block which is attached to load transfer plate because in this region maximum bending moment.

Maximum compressive stress for this load case is 298.25Mpa. This value is safe according design criteria.

3. Load transfer plate

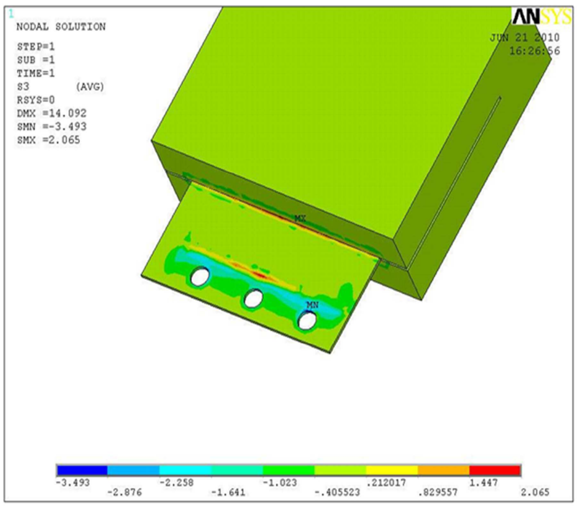


Fig.6.12 Third principle stresses in load transfer plate

Material used for load transfer plate is alloyed steel which is having yield strength 575Mpa. We see that Max. Stress value in load transfer plate are much below the yield strength

Before completion of this analysis Bharat Forge test same crankshaft outside the company lab due to unavailability of instruments for this test.

Now to validate our CAE result we are comparing with this actual testing result.

TABLE III

COMPARISON OF PHYSICAL & FEA RESULTS FOR BENDING MOMENT 3000 N-M

	Physical Test Value	FEA Values	Variation (%)
Stress (MPa)	-231.54	-225	2.82%
Strain(mm/m)	1050	1020	2.85%

Above table-III gives comparisons of physical test results and FEA results .We see that the difference between our results with experimental result variation is about 2-3%. This is good correlation with experimental result. According to above results minimum stress in crankshaft at -3,000 Nm bending moment =-225MPa. Maximum stress in crankshaft at 15,000 Nm, bending moment =1139MPa.

$$\text{Load ratio}= R = \frac{\sigma_{\min}}{\sigma_{\max}}$$

$$R = \frac{-225}{1139}$$

R=-0.197

So we can say that we achieve the load ratio -0.2 of stress in fillet.

VII. CONCLUSION

1. Test fixture FE analysis shows lower stresses in various components such as V-block, load transfer arm etc as compared to yield strength of material hence design is safe.

2. Crankshaft bending test on above fixture at load ratio R=-0.2 induces high tensile stresses in pin fillet as compared to load ratio R=-1 which is reality & it is as per industry requirement.

3. Crankshaft physical test results (tested at other agency) & FE analysis results matches with 90-95% confidence level hence this design is validated.

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