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Analysis of stress on crankshaft of an I.C Engine

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Abstract:
In this work, a four-cylinder four-stroke engine's crankshaft was statically analyzed. Finite element analysis was used to determine the variation in stress magnitude at important places. A three-dimensional model of the crankshaft was generated using PRO/E software. The crankshaft was subjected to finite element analysis using ANSYS. The load was then applied to the FE model, with boundary conditions based on the engine's mounting circumstances in ANSYS. The analysis is performed to identify crucial locations in the crankshaft. The investigation looked at stress variation during the engine cycle as well as the effect of torsion and bending strain. Von-mises stress is estimated theoretically and numerically using the FEA software ANSYS. The stresses obtained from these analyses using the analytical approach and the FEA results are compared. The results obtained from the aforementioned analysis were utilized to compare the analytical and FEA results. The value of Von-Misses Stresses obtained from the analysis was significantly lower than the material yield stress, indicating that the design was safe and that optimization could be performed to minimize material and hence cost.
Keywords-Finite Element Analysis, ProE, ANSYS, Crankshaft.

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I. Introduction:

The crankshaft is a big component with complex geometry in the engine that converts the piston's reciprocating displacement to rotary motion via a four-link system. This investigation was carried out on a four-cylinder four-stroke cycle engine. Because the crankshaft is subjected to a considerable number of load cycles over its service life, fatigue performance and durability must be considered during the design process. Design innovations have always been a major concern in the crankshaft-manufacturing sector, in order to produce a less expensive component with the least amount of weight possible while meeting fatigue strength and other functional requirements. These advancements lead to lighter and smaller engines that consume less fuel and produce more power. The crankshaft must be sturdy enough to withstand the downward force of the power stroke without undue bending. So the crankshaft's strength determines the dependability and life of the internal combustion engine.

This study examined the stress on the crankshaft of a four-cylinder, four-stroke engine. The load is then applied to the model in Pro-E, with boundary conditions based on the engine mounting conditions. The crankshaft was subjected to finite element analysis using ANSYS. The stresses obtained from these analyses using the analytical approach and the FEA results are compared. Because crankshafts have complicated structures, integral crankshafts are examined for finite element analysis. We looked into how stress varied throughout the engine cycle and how torsion and bending load affected the results. ANSYS FEA software is used to calculate von-Mises stress theoretically. The aforementioned analysis's results were compared to the FEA's results analytically, and further optimization might be done to maximize both the production cost and weight. The crankshaft is made of 40Cr4Mo3.

II. Literature Review :

For a single cylinder, four stroke engine, Farzin H. Montazersadgh and Ali Fatemi[1] choose for forged steel and a cast iron crankshaft. A CMM machine was used to digitize both crankshafts. The engine's ADAMS modeling was used to verify the results of the load analysis. Geometry and production cost optimization were carried out at the following stage. Von Mises stress at the critically stressed point is unaffected by torsional load in the overall dynamic loading conditions. There was only a 7% discrepancy between the experimental stress and FEA values. Due to strong stress gradients, all of the crankshaft's critical places are found on the fillet sections. The forged steel weighs 18% less as a result of geometry improvement. Crankshaft fatigue strength is increased by 165% as a result of fillet rolling, which creates compressive residual stress in the fillet sections. K. Thriveni and Dr. B. Jaya Chandraraiah[2] examined the static analysis of a single-cylinder, four-stroke I.C. engine's crankshaft. CATIA-V5 software is used to construct the crankshaft model. Using the ANSYS software and the boundary conditions, finite element analysis (FEA) is used to determine the fluctuation of stress at important crank shaft points. Subsequently, the outcomes are determined. The crankshaft experiences 15.83 MPa of von-misses stress and 8.271 MPa of shear stress induction. Von Misses stress is 19.6 MPa and shear stress is 9.28 MPa, according to the theoretical results. Theoretical and FEA findings of Von-misses stress and shear stress are compared with the model validation, and both are within tolerances. It can also be expanded to include various materials, dynamic analysis, and crank shaft optimization. The crankpin neck surface's center is where the greatest distortion is seen, according to the author's conclusion. The fillet areas between the crankshaft journal and crank cheeks, as well as the vicinity of the central point journal, exhibit the highest level of stress. Yongqi Liu, Ruixiang Liu, and Jian Meng[3] talked about the modal analysis and stress analysis of a four-cylinder crankshaft. The crank throw's vibration modalities, distortion, and stress state were examined using the FEM program ANSYS. The crankshaft modal analysis provided an explanation for the link between vibration modal and frequency. This offers a useful theoretical framework for engine design optimization and advancement. The crankpin neck surface's center shows the most distortion. The fillet between the crankshaft journal and the crank cheeks, as well as close to the central point journal, is where the greatest stress is found. Under the lower frequency, the crankshaft deformation was primarily bending deformation. The point of maximum deformation was found at the intersection of the crankpin, crank cheeks, and main bearing journal. Thus, the region where bending fatigue cracks are likely to form. Researchers Sanjay B. Chikalthankar, V. M. Nandedkar, and Surender Kumar Kaundal[4] look at the stresses that arise from dynamic loading on the crankshaft. This study used a dynamic crankshaft simulation, and finite element analysis was used to determine how the stress magnitude varied at key places. Pro-ENGINEER software was used to produce the 3D model of the crankshaft, which was then loaded into ANSYS software for examination. We may now conclude that, for more accurate results, we can run the dynamic analysis on the same system. At the center of the crankpin neck surface is where the largest distortion is seen. The fillets between the crankshaft journal and crank cheeks, as well as the area next to the central point journal, are where the greatest stress is found. The main journal's edge is a high-stress location. While static analysis overestimates results, dynamic loading analysis of the crankshaft yields more realistic stresses. Precise stresses are essential for fatigue study and crankshaft optimization. Momin Muhammad Zia Muhammad Idris[5] reports the findings of a PRO/E and ANSYS software-assisted strength analysis performed on the crankshaft of a single-cylinder, two-stroke gasoline engine. Crankshaft strength study was conducted using a three-dimensional model created in PRO/E and loaded into ANSYS. Torsion stress, which is typically disregarded, is included in the analysis in this paper. To validate the model, a computation method is employed. In order to lower the mass of the crankshaft, the report also suggests changing its design. Additionally, the improved design is analyzed. This study presents the results of a strength analysis of the crankshaft of a two-stroke, single-cylinder gasoline engine. A suggested design change is made in light of the result analysis. The analysis also took into account the torsion stress. It is discovered that the crankpin fillet and journal fillet are the weakest parts of the crankshaft. Crankshaft model and crank throw were generated by Pro/E software and subsequently loaded into ANSYS software, according to Rinkle Garg and Sunil Baghla [6]. The outcome demonstrates that the crankshaft's strength has improved as total deformation, strain, and stress maximums have all decreased. The crankshaft's weight is decreased. decreases the force of inertia as a result. The I.C. engine's performance will improve and the crankshaft's cost will go down as its weight decreases. V. Mallikarjuna Reddy and T. Vijaya Devi[8] address the issue that arose with the crankshaft of a six-cylinder, four-stroke engine. It consists of a static structural analysis of the crank shaft of a six-cylinder engine. It uses modeling and simulation approaches to identify and address the problem. Unigraphics-NX7.5 is the program used to model the crankshaft. Crankshaft structural analysis is done using ANSYS. The crankshaft was then optimized using the information gleaned from the previously described investigation. The improvements in crankshaft strength as the upper limits of stresses are demonstrated in this paper's results. Our design is safe since the von-misses stresses obtained from the analysis are far smaller than the material yield stress.Additionally, there is a 5% weight reduction in the crankshaft. decreases the force of inertia as a result. Reduced weight on the crankshaft will result in lower crankshaft costs and improved engine performance. A three-dimensional model of the crankshaft, produced by SOLID

WORKS software and loaded into ANSYS, has been examined by Jaimin Brahmbhatt and Prof. Abhishek Choubibey [10]. Crankpin neck surface center is where the crankshaft highest deformation is visible. Between the crankshaft journals and crank cheeks, as well as in close proximity to the central point journal, are the locations where the greatest stress is found. High stress is concentrated at the main journal's edge.

III. Design:

Force analysis

3.1 Parameters of Engine

The crankshaft is for car. The technical specifications are given in Table 3.1

Table 3.1:- Technical Specifications of engine of a car.

SPECIFICATIONS	**VALUES**
Number of Cylinders	4 Cylinder, SDE Common Rail
Type of Engine (Inline / 'Vee' engine)	1248 cc, Inline Diesel, 475IDI Engine
Bore / Stroke (D/L)	69.6 / 82
Power @ speed	75 PS (55 KW) @ 4000 rpm
Toque @ speed	190 Nm @ 1750 RPM
Compression Ratio	17.6:1
Engine type	Compressor Ignition (CI) Engine
Ratio of r/l –λ	0.31

The material of crankshaft is 40Cr4Mo3. The mechanical properties are given in Table 3.2

Table 3.2:- The Mechanical Properties of the Crankshaft Material 40Cr4Mo3

PROPERTIES	**VALUES**
Tensile Strength	83 Kg/mm^2MIN
Yield Strength	55 Kg/mm^2MIN
% Elongation	14 % MIN
% R.A.	53 % MIN
Impact	34 J MIN

3.2 When the crank is at dead center : At this crank position, the piston's maximum gas pressure will transfer its maximum force to the crankpin in the crank's plane, only bending the shaft. Only bending moment will be applied to the crankpin and crankshaft ends. Therefore, the twisting moment is zero and the bending moment on the shaft is maximal when the crank is at dead center.The following equations indicate the different forces acting on the crank shaft through the connecting rod (Fp), the horizontal and vertical responses on the shaft, and the resultant force at bearings 2 and 3.Now the piston force

P_{max} = P x no of cylinders/1248 x 10^{-6} x 4000

= 55 x 4/1248 x 10^{-6} x 4000

= 44.07 KN

Piston force, Fp = π/4 x D^2 x P_{max}

= π/4 x $(69.6)^2$ x 44.07

= 167.67 KN

The distance between the bearing 1 & 2 as b = 110 mm and b1 = b2 = b/2 = 55 mm.

We know that due to piston gas load, there will be two equal horizontal reactions H1 & H2 at bearings 1 & 2 respectively.

i.e H1 = Fp/2 = 167.67/2 = 83.835 KN = H2

The length of bearing to be equal i.e. c1=c2=c/2.

We know that due to weight of flywheel acting downwards, there will be two vertical reactions V2 & V3 at bearings 2 & 3 V2 = V3 = W/2 = 100 x 9.8/2 = 490 N

Since, the belt is absent in engine, neglecting the belt tension exerted by belt. i.e. $T_1 + T_2 = 0$.

3.2.1 Crank pin :

Crankpin is also subjected to shear stress due to twisting moment. Thus we can calculate bending moment at centre of crankpin and twisting moment on crank pin and the resultant moment.

Let,

d_C = Diameter of crank pin in mm l_C = length of crank pin

σ_{allow} = allowable bearing stress for crank pin = 83

kgf/mm^2 =814.23 N/mm^2.
Bending moment at the centre of crank pin is,
Mc = H1 x b2= 83.835 x 55 = 4610.925 x 103 N-mm
Now,
Diameter of crank pin, d_C = 60 mm
The length of crank pin, lc = 38 mm

3.2.2 Left hand crank web

The crank web is for eccentric loading. There will be two stresses acting on the crank web, one is direct compressive stress and the other is bending stress due to piston gas load (Fp).
We know that the thickness of crank web is
t = 19 mm.
Also width of crank web is,
w = 64 mm
The maximum bending moment on crank web is
M_{max} = H1 (b2 –lc/2-t/2) = 83.835 (55 - 38/2 - 19/2)
= 2221.6275 x 10^3 N-mm
Section modulus, Z = 1/6 x w x t^2 = 1/6 x 64 x 19^2
= 3850.67 mm^3
Bending stress, σ_b = M/Z
σ_b= 576.946 N/mm^2

The compressive stress acting on crank web are
σ_c = H1 / (w x t) = 83.835 / (64 x 19) = 68.943 N/ mm^2
The total stress acting on crank web is
$\sigma_T = \sigma_b + \sigma_c$ = 645.889 N/ mm^2
Thus total stress on crank web is less than allowable bending stress of 814.23 N/mm^2

3.3When the crank is at an angle of maximum twisting moment:

The twisting moment on the crankshaft will be maximum when the tangential force on the crank (F_T) is maximum. The maximum value of tangential force lies when the crank is at angle of 30º to 40º for constant pressure combustion engines (i.e. diesel engines). When the crank is at angle at which the twisting moment is maximum, the shaft is subjected to twisting moment from energy or force stored by flywheel. Considering this, we have the various forces acting on crankshaft at different twisting angles.

Consider a position of crank at an angle of maximum twisting moment. If P´ is the intensity of pressure on the piston at this instant, then the piston gas load at this position of crank.
We know that piston gas load,
Fp = π/4 x D^2 x P´ = π/4 x $(69.6)^2$ x 44.07
= 167.67 x 10^3 N
To find thrust in connecting rod F_Q, find out angle of inclination of connecting rod with the line of stroke (Φ).
We know that

$$\text{Sin}\ \Phi = \frac{Sin\ \theta}{\frac{1}{r}} = \text{Sin}\ 35^\circ \text{ x } 0.31 = 0.1778$$

Φ = 10.24˚
Thrust in connecting rod

$$F_Q = \frac{F_p}{cos\emptyset} = \frac{167.67}{\cos 10.24} = 170.384 \text{ x } 103 \text{ N}$$

Tangential Force acting on the Crankshaft
F_T = F_Q Sin (θ + Φ) = 170.384 x 10^3 Sin (35˚ + 10.24˚)
= 120.983 x 10^3 N
And Radial Force on the Crankshaft
F_R = F_Q Cos (θ + Φ) = 170.384 x 10^3 Cos (35˚ + 10.24˚)
= 119.974 x 10^3 N
Reactions at bearings 1 and 2 due to Tangential force (F_T) is given by` H_{T1} = H_{T2} = F_T / 2 = 120.983 x 10^3/ 2 = 60.4915 x 10^3 N
Reactions at bearings 1 & 2 due to Radial force (F_R) is given by
H_{R1} = H_{R2} = F_R / 2 =119.974 x 10^3 / 2 = 59.987 x 10^3 N

3.4 Crank Pin :
Let, d_C = Diameter of crank pin in mm
= 60 mm
Bending moment at the centre of crank pin is
M_C = H_{R1} x b_2 = 59.987 x 10^3 x 55 = 3299.285 x 10^3 N mm
Twisting moment on the crankpin is
T_C = H_{T1} x r = 60.4915 x 10^3 x 57 = 3448.0155 x 10^3 N mm

From this we have equivalent twisting moment on the crankpin

$$Te = \sqrt{\boldsymbol{Mc^2 + Tc^2}}$$
$$= \sqrt{(3299.285 \times 10^3)^2 + (3448.0155 \times 10^3)^2}$$
$= 4772.221 \times 10^3$ N-mm

We also know that equivalent twisting moment on the crankpin

$$Te = \frac{\pi}{16} \times d^3 \times \tau$$

Shear stress

$\tau = 112.522\ \mathrm{N/mm^2}$

Crankpin against fatigue loading

According to distortion energy theory, the Von-Mises stress induced in the crank-pin is

$$M_{ev} = \sqrt{(K_b \times M_c)^2 + \frac{3}{4}(K_t \pm T_c)^2}$$

Here, K_b = Combined shock and fatigue factor for bending (Take K_b =2)
K_t = Combined shock and fatigue factor for torsion (TakeK_t =1.5)

$$M_{ev} = \sqrt{(2 \times 3299.285 \times 10^3)^2 + \frac{3}{4}(1.5 \times 3448.0155 \times 10^3)^2}$$

$M_{ev} = 7975.18 \times 10^3$ N-mm

Also know that,

$$M_{ev} = \frac{\pi}{32} \times d^3 \times \sigma v$$
$$7975.18 \times 10^3 = \frac{\pi}{32} \times 60^3 \times \sigma v$$

$\boldsymbol{\sigma v}$ = 376.086 N/mm^2

3.4 Shaft Under Flywheel :

$ds =$ diameter of crankshaft = 76 mm

The bending moment on the shaft is,

$M_S = \pi /32 \times (d_s)^3 \times \sigma_{allow}$
$= \pi/32 \times (76)^3 \times 814.23$
$= 35090.355 \times 10^3$ N-mm

Twisting moment on the shaft

$T_S = F_T \times r = 120.983 \times 10^3 \times 57$
$= 6896.031 \times 10^3$ N-mm

Equivalent twisting moment on the shaft

$$Te = \sqrt{\boldsymbol{Ms^2 + Ts^2}}$$
$$= \sqrt{(35090.355 \times 10^3)^2 + (6896.031 \times 10^3)^2} = 35761.5472 \times 10^3 \text{ N-mm}$$
$= 35.7615 \times 10^6$ N-mm

$$Te = \frac{\pi}{16} \times {d_s}^3 \times \tau$$
$$35.7615 \times 106 = \frac{\pi}{16} \times 76^3 \times \tau$$

Shear stress: $\boldsymbol{\tau}$ = 414.902 N/mm^2

IV. Modeling of crankshaft

In this project work, stress analysis on crankshaft of four cylinder four stroke cycle engine is done. The model is created in Pro-E is shown in Fig 4.1 and Fig. 4.2. The load and boundary conditions were then applied to the model in ANSYS.

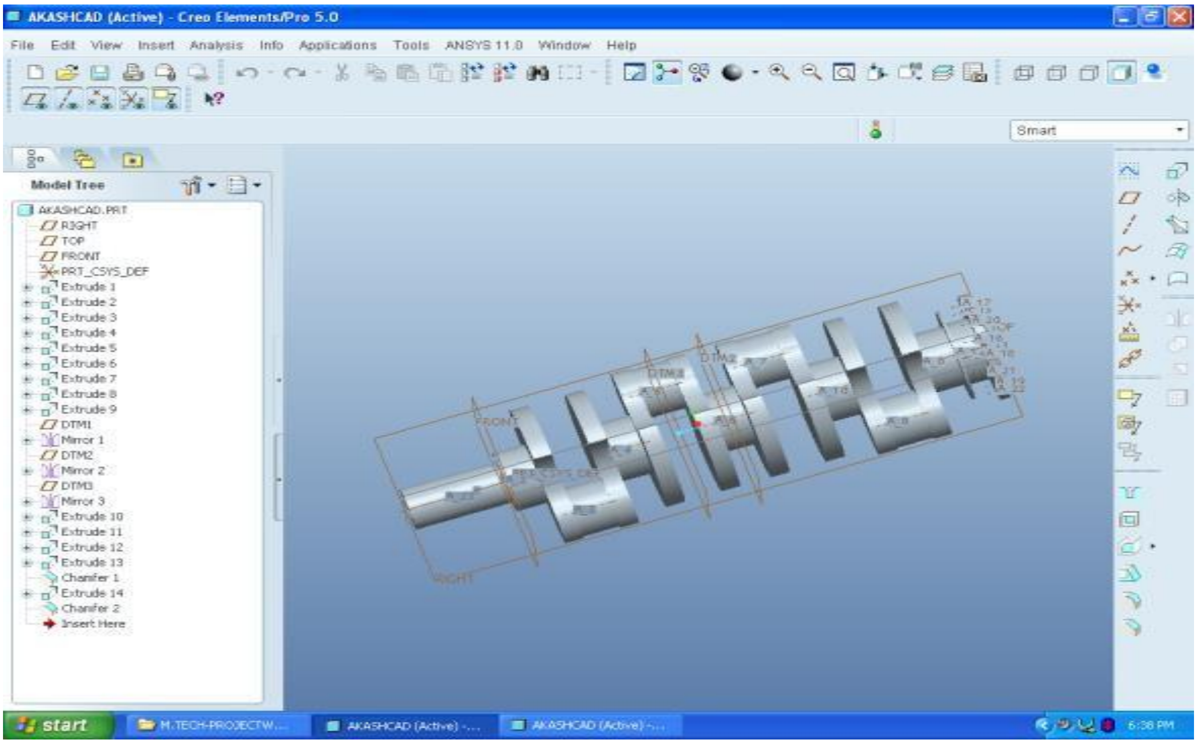

Fig 4.1 : Model of Crankshaft Created in Pro-E

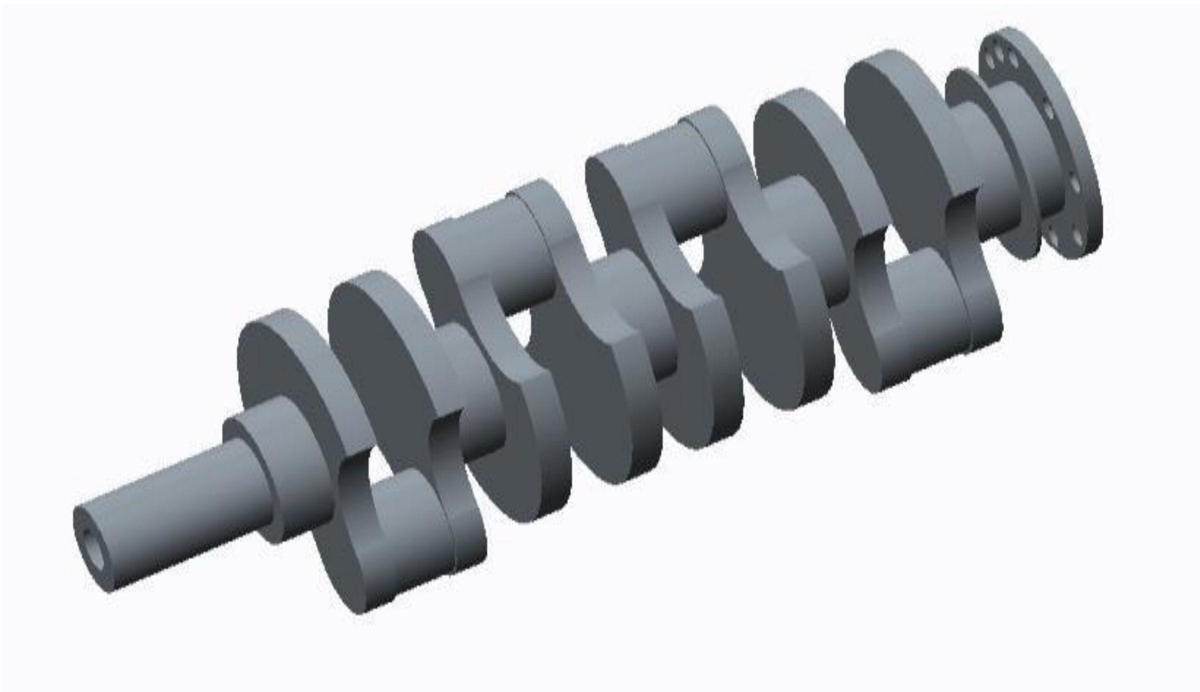

Fig 4.2 : 3-D model of Crankshaft

V. Stress analysis of crankshaft

The stress analysis of Crankshaft is carried out in ANSYS. The mesh model, equivalent stress, loading and boundary conditions, Equivalent (Von-Mises) stress, shear stress, maximum principal stress and minimum principal stress are shown in following figures.

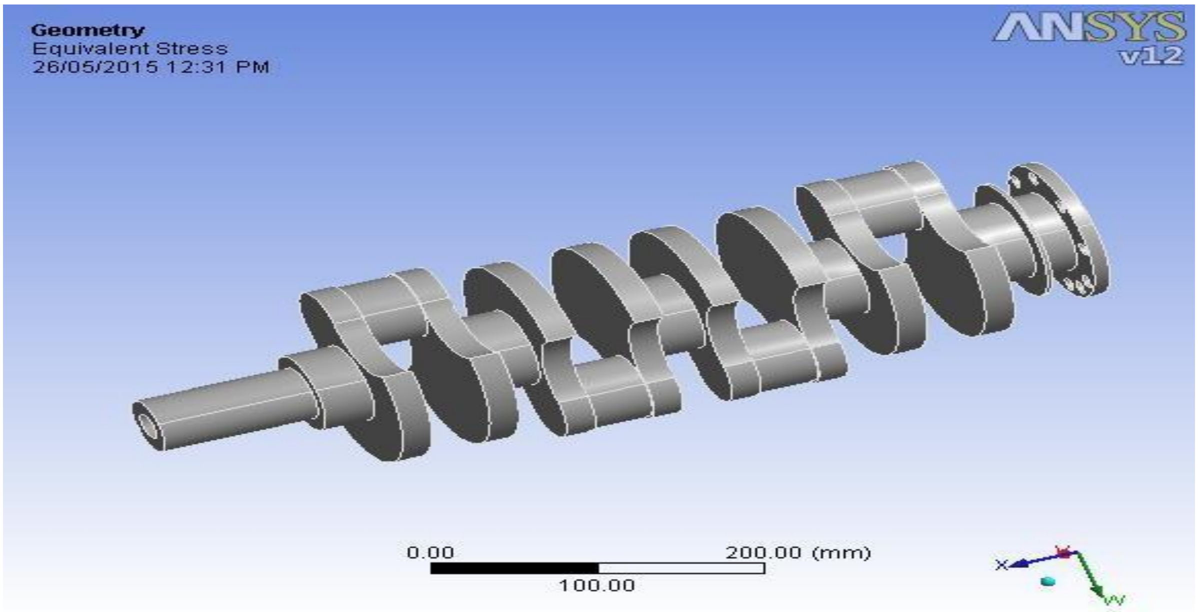

Fig 5.1:- Crankshaft in ANSYS

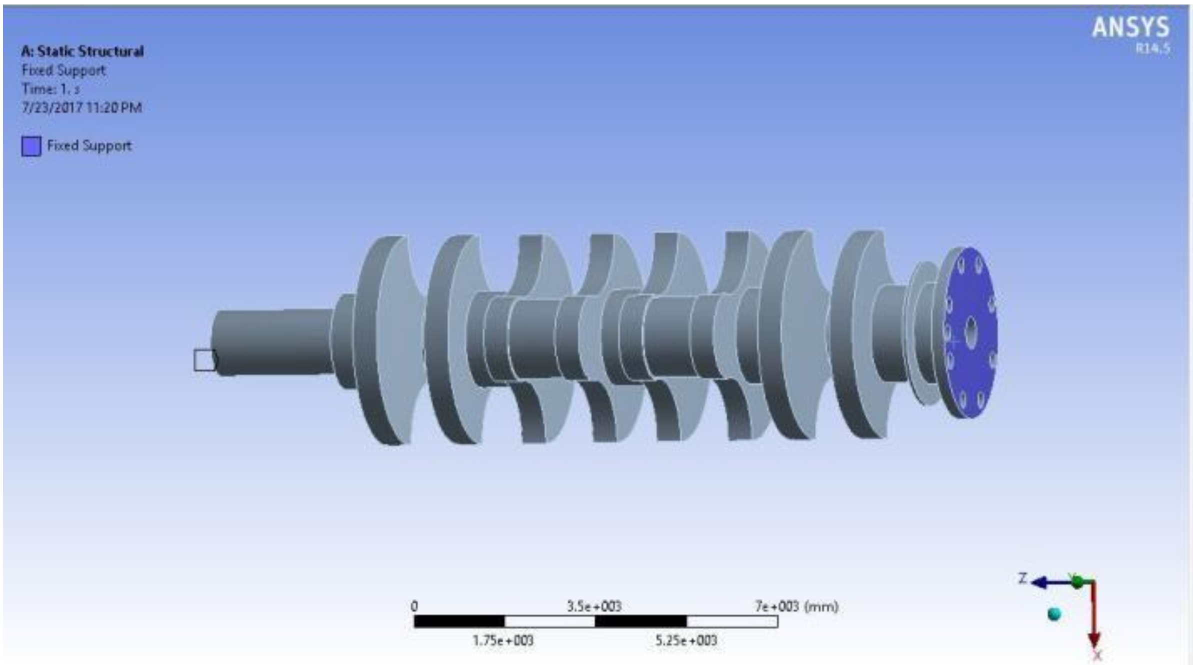

Fig 5.2:- Fixed support to the Crankshaft

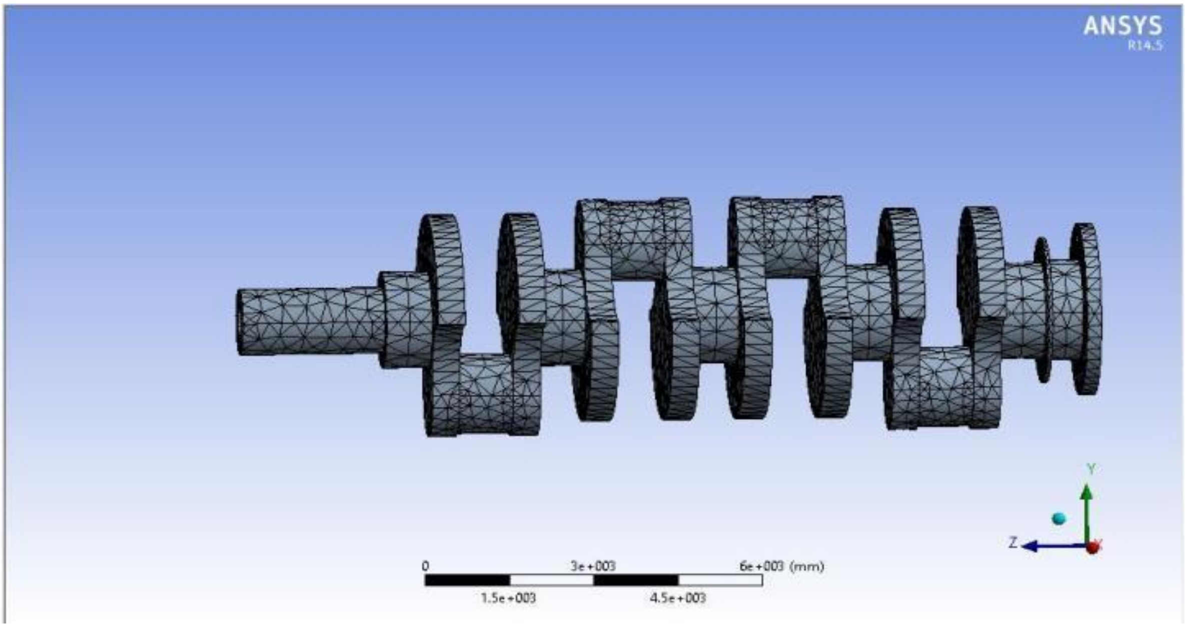

Fig 5.3:- Mesh Model of Crankshaft in ANSYS

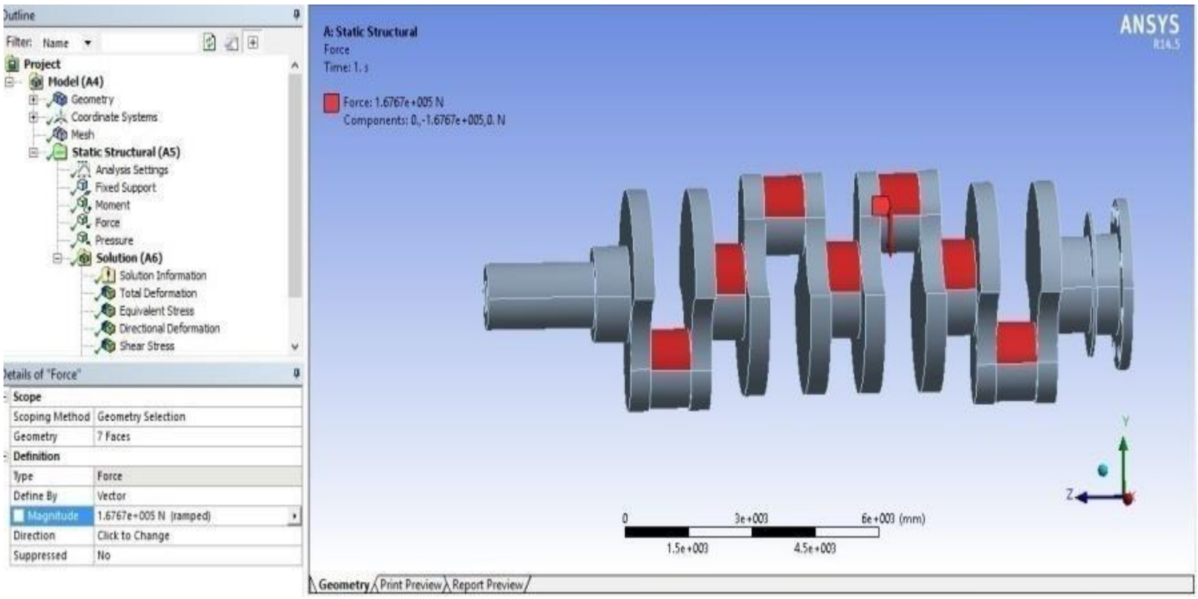

Fig 5.4:- Apply Boundary Condition on Crankshaft in ANSYS

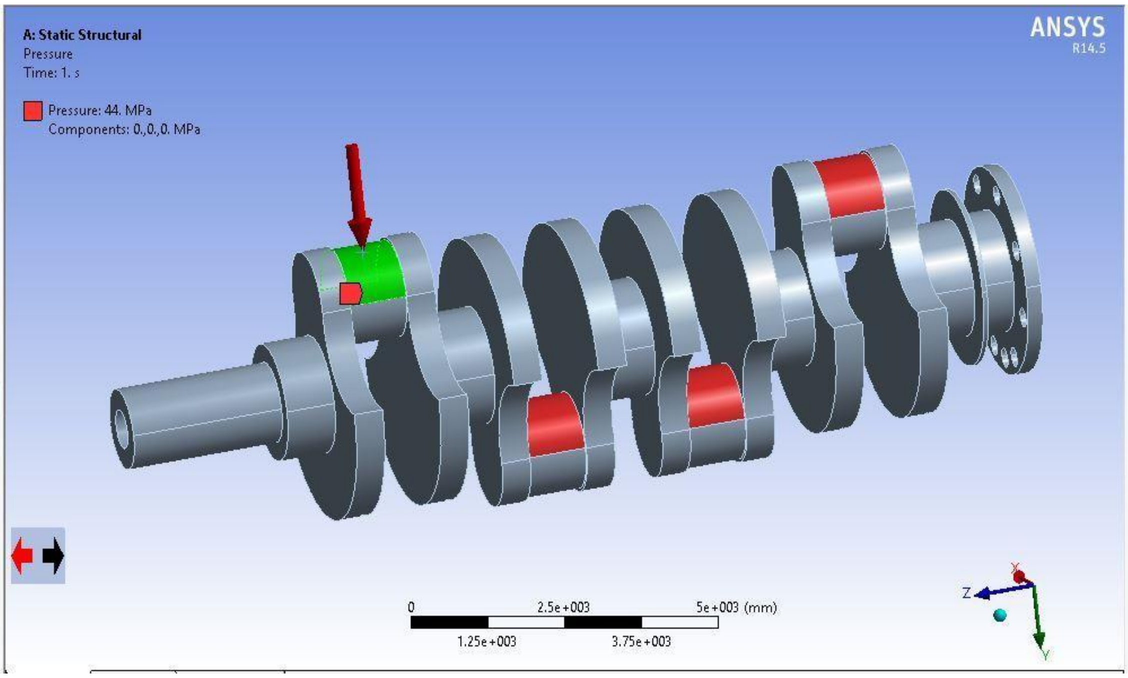

Fig 5.6:- Loading Conditions on Crankshaft

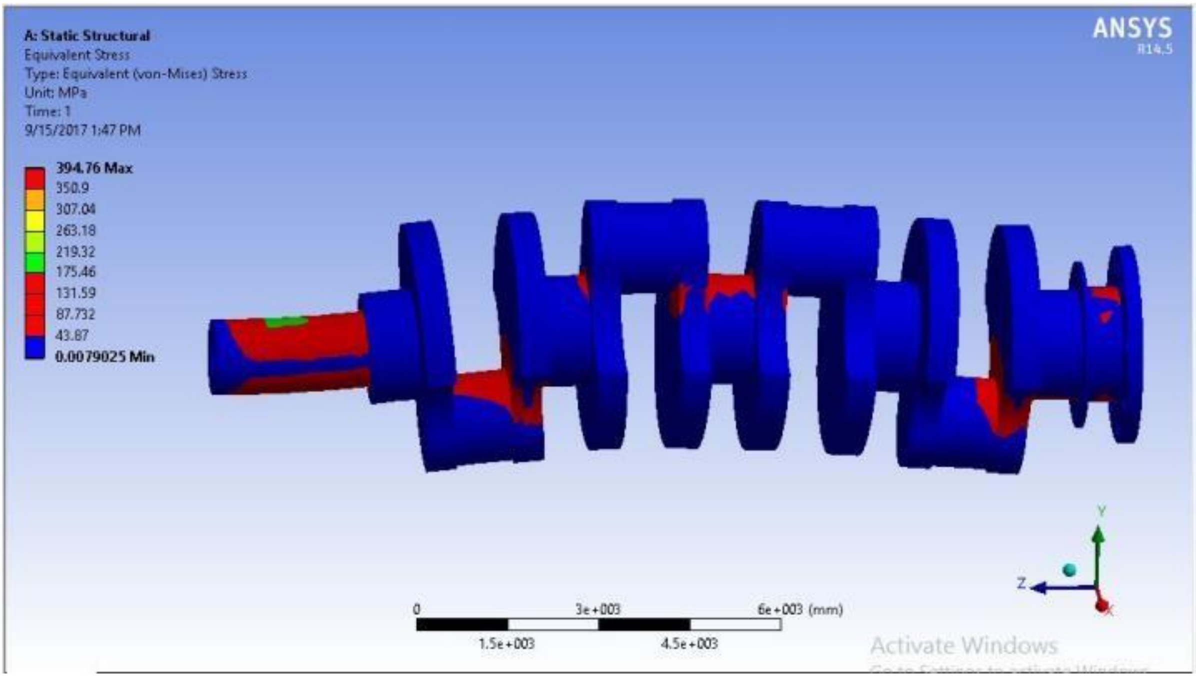

Fig 5.7:- Equivalent (Von-Mises) Stress of Crankshaft

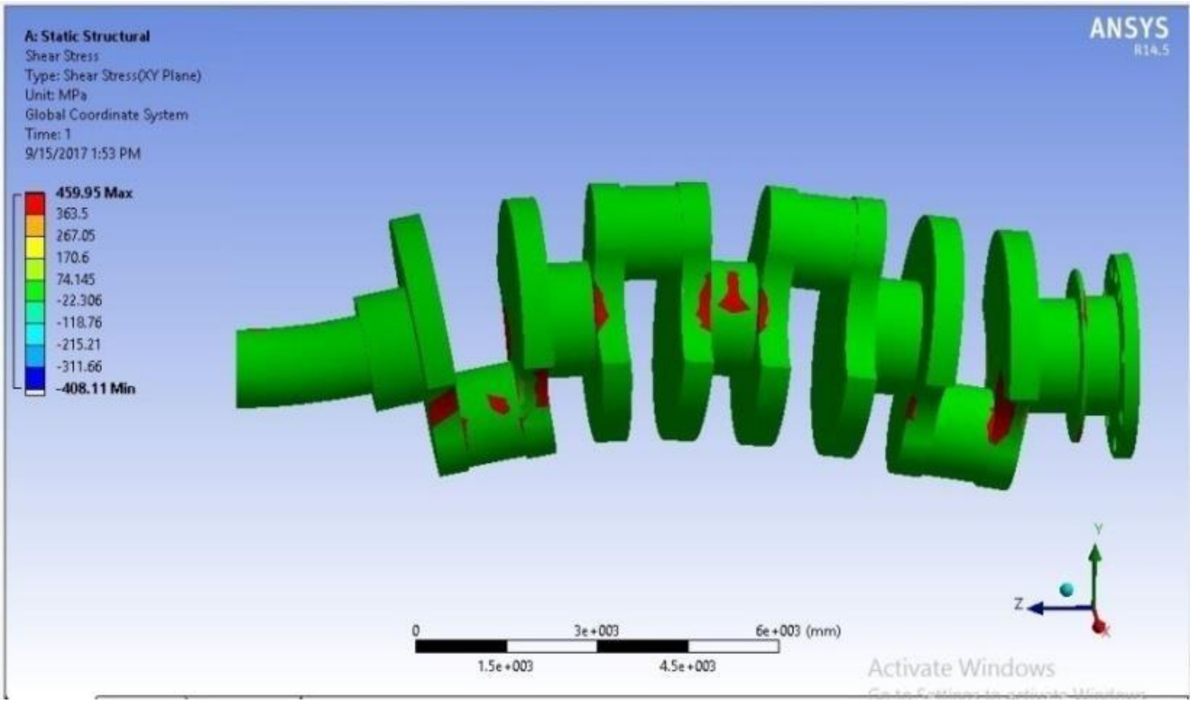

Fig 5.8:- Shear Stress on Crankshaft

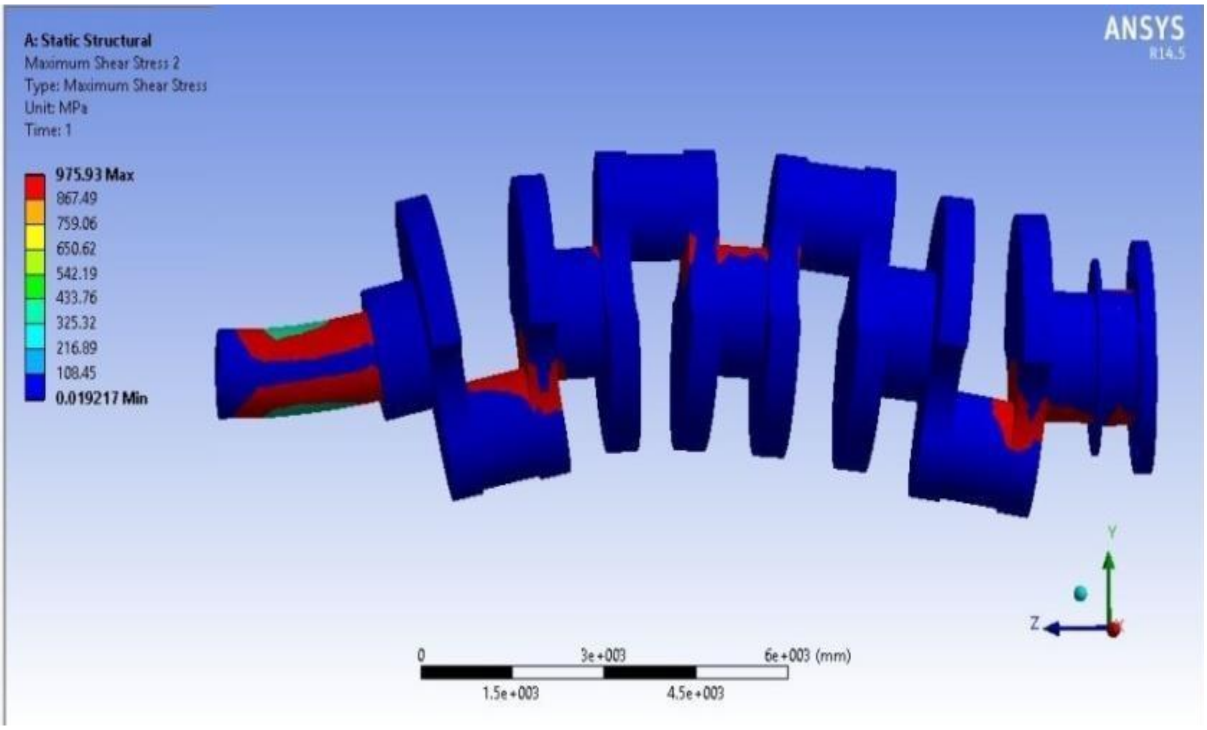

Fig 5.9:- Maximum Shear Stress on Crankshaft

VI. Result and Discussion :

The analytical and FEA analysis is carried out for the crankshaft, the comparison of stresses determined using FEA analysis and analytical calculations is given in table 6.1.

Table 6.1: Comparison between the Theoretical and practical results.

Sr. No.	Types of Stresses	Theoretical	FEA Analysis(ANSYS)
1	Von-Mises Stress (N/mm^2)	$\boldsymbol{\sigma_V}$ = 376.086 N/mm^2	394.76 Mpa
2	Shear Stress (N/mm^2)	$\boldsymbol{\tau}$ = 414.902 N/mm^2	459.95 Mpa

Conclusion:
Hence,
The Von-Mises Stress (N/mm^2) in theoretical analysis is
σ_V = 376.086 N/mm^2
and, FEA Analysis is FEA σ_V = 394.76 Mpa

The Shear Stress (N/mm^2) in theoretical analysis is
$\boldsymbol{\tau}$ = 414.902 N/mm^2
and, FEA Analysis is FEA $\boldsymbol{\tau}$ = 459.95 Mpa

References:

[1]. Farzin H. Montazersadgh and Ali Fatemi, “Dynamic Load and Stress Analysis of a Crankshaft,” SAE Paper No. 2007-01-0258, 2007 SAE International
[2]. K. Thriveni, Dr.B.JayaChandraiah, “Modeling and Analysis of the Crankshaft Using Ansys Software”, International Journal of Computational Engineering Research, Vol, 03, Issue, 5, Pages:84-89, May 2013
[3]. Jian Meng, Yongqi Liu, Ruixiang Liu, “Finite Element Analysis of 4-Cylinder Diesel Crankshaft”, I.J. Image, Graphics and Signal Processing, 2011, 5, 22-29 Published Online August 2011 in MECS.
[4]. Sanjay B. Chikalthankar, V. M. Nandedkar, Surender Kumar Kaundal, “Finite Element Analysis Approach for Stress Analysis of Crankshaft under Dynamic Loading,” International Journal of Scientific & Engineering Research, VOLUME 4, ISSUE 2, FEBRUARY-2013 1 ISSN 2229-5518 IJSER
[5]. Momin Muhammad Zia Muhammad Idris, “Crankshaft Strength Analysis Using Finite Element Method” International Journal of Engineering Research and Applications (IJERA) ISSN: 2248-9622 Vol. 3, Issue 1, January -February 2013, pp.1694-1698.
[6]. Rinkle Garg, Sunil Baghla, “Finite Element Analysis and Optimization of Crankshaft Design”, International Journal of Engineering and Management Research, Vol.-2, Issue-6, December 2012, ISSN No.: 2250-0758, Pages: 26-31
[7]. Mr.B.Varun, “Stress Analysis and Optimization of Crankshafts Subject to Static Loading”, International Journal Of Engineering And Computer Science ISSN:2319-7242 Volume 3 Issue 5 May, 2014 Page No. 5579-5587
[8]. V. Mallikarjuna Reddy, T. Vijaya Devi, “Design, Analysis and Optimization of a 6 cylinder Engine Crank shaft”, International Journal Of Modern Engineering Research (IJMER), ISSN: 2249–6645, Vol. 4, Iss.8, Aug. 2014, Pages: 30-37.
[9]. K. Sandya, M. Keerthi, K. Srinivas, “Modeling and Stress Analysis of Crankshaft Using FEM Package ANSYS” International Research Journal of Engineering and Technology(IRJET), Volume:3, Issue-01, Jan-2016, ISSN No.: 2395-0072, Pages: 687-693.
[10]. Jaimin Brahmbhatt, Prof. Abhishek choubey “DESIGN AND ANALYSIS OF CRANKSHAFT FOR SINGLE CYLINDER 4-STROKE DEISEL ENGINE”, International Journal of Advanced Engineering Research and Studies, E-ISSN2249–8974, IJAERS/Vol. I/ Issue IV/July-Sept., 2012/88-90
[11]. Amit Solanki, Jaydeepsinh Dodiya, ”Design and Stress Analysis of Crankshaft for Single Cylinder 4- Stroke Diesel Engine”, International Journal for Research in Applied Science and Engineering Technology (IJRASET), ISSN: 2321-9653, Vol. 2 Issue V, May 2014, Pages:320-324
[12]. R. J. Deshbhratar, Y. R. Suple, “Analysis & Optimization of Crankshaft Using FEM”, International Journal of Modern Engineering Research (IJMER), Vol. 2, Issue. 5, Sep.-Oct. 2012, pp-3086-3088 ISSN: 2249-6645.
[13]. Sakhare, V. P., Mishra, P., & Tonpe, K. K. (2022). Investigation and Simulation of Coconut Fibrous Husk Extracting Tool Geometry Using Finite Element Analysis. Mathematical Statistician and Engineering Applications, 71(4), 961–981.
[14]. Sakhare, V. P., Mishra, P., Babu, P. S., & Reddy, M. V. (2024). A study and

experimental analysis of coconut husk extracting tool using generative design and topology optimization technique. International Journal on Interactive Design and Manufacturing (IJIDeM), 18(5), 2639–2651. https://doi.org/10.1007/s12008-023-01240-9

[15]. Waghmare, S., Shelare, S., Mungle, N., Sakhare, V., & Dhande, M. (2024). Development of Paddy Transplanter Machine Using Low-Cost Materials. In A. Sachdeva, K. K. Goyal, R. K. Garg, & J. P. Davim (Eds.), Recent Advances in Operations Management and Optimization (pp. 59–73). Springer Nature Singapore.

[16]. KTonpe, K., Sakhare, V. P., & Sakhale, C. N. (2014). Design & Performances of Coconut De-Shelling Machine. Journal of Engineering Research and Applications Www.Ijera.Com, 4(7), 39–44. www.ijera.com.

[17]. Sakhare, M. V. P., Sakhale, C. N., & Tonpe, M. K. K. (2014). Design and development of coconut De-husking machine. International Journal of Engineering Research & Technology (IJERT), ISSN, 2278-0181. www.ijert.org

[18]. Hailemariam Nigus, "Kinematics and Load Formulation of Engine Crank Mechanism", Mechanics, Materials Science & Engineering, October 2015 – ISSN 2412-5954.

[19]. T. Anbu, S. Govindaraji, C. Kannadhasan, S. Selvakumar, "Stress Analysis of Crankshaft Subjected To Dynamic Loading", IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE), e-ISSN: 2278- 1684, p-ISSN: 2320-334X, Volume 11, Issue 4 Ver. II (Jul-Aug. 2014), PP 35-39.

[20]. Peeyush Chauhan, Om Prakash Umrao, "Stress Analysis in Crankshaft Through FEM", International Journal on Recent and Innovation Trends in Computing and Communication, Volume: 4, Issue: 1, January 2016

[21]. Guagliano, M., Terranova, A., and Vergani, L., 1993,"Theoretical and Experimental Study of the Stress Concentration Factor in Diesel Engine Crankshafts, "Journal of Mechanical Design, Vol. 115, pp. 47-52.

[22]. Baxter, W. J., 1993, "Detection of Fatigue Damage in Crankshafts with the Gel Electrode," SAETechnical Paper No. 930409, Society of Automotive Engineers, Warrendale, PA, USA

[23]. Borges, A. C., Oliveira, L. C., and Neto, P. S., 2002, "Stress Distribution in a Crankshaft Crank Using a Geometrically Restricted Finite Element Model," SAE Technical Paper No. 2002-01-2183, Society of Automotive Engineers, Warrendale, PA, USA.

[24]. Sakhare, V. P., Tonpe, K. K., & Sakhale, C. N. (2016). Performance analysis of hydraulically operated coconut de-husking machine. J Emerging Technol Innov Res, 1(2), 151-156.