Journal of Advances in Science and Technology
Vol. 13, Issue No. 1, (Special Issue) March-2017, ISSN 2230-9659

Design of Crankshaft of Micro-alloy Steel by Using MATLAB for Fatigue Failure

Ranjeet Ghule[1]*, M. V. Kavade[2], G. S. Uppar[3]

[1]Mechanical Department, Rajarambapu Institute of Technology, Sakhrale, Sangli, India

[2]Mechanical Department, Rajarambapu Institute of Technology, Sakhrale, Sangli, India

[3]Bharat Forge Limited, Pune, India

Abstract – Crankshaft is one of the most critical components of an IC engine, failure of which may result in perversion and makes engine useless unless costly repairs are performed. The dynamic load and rotating system exerts repeated bending and shear stress due to torsion, which are common stresses acting on the crankshaft and mostly responsible for crankshaft fatigue failure. The fatigue strength determination plays an important role in crankshaft development considering its safety and reliable operation. Fractographic studies indicate that fatigue is the dominant mechanism of failure of the crankshaft.

Micro-alloying is used in wrought steels to refine grain size during Thermo-Mechanical Controlled Processing (TMCP), with an attendant improvement in mechanical properties. The micro-alloyed steel 38MnVS6 material is used for crankshaft. In this work MATLAB (Matrix Laboratory) has been used to design 4 cylinder crankshaft of 2596cc diesel engine.

Keywords — Crankshaft; Micro-alloying; High strength steel; Fatigue; MATLAB

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INTRODUCTION

In Internal Combustion engines, the crankshaft experiences complex loading due to motion of the connecting rod, which transforms into two sources of loading to the crankshaft. The loading on the crankpin consists of bending and torsion. Crankshaft must be strong enough to take the downward force of the power stroke without excessive bending so mostly the life and reliability of engine is depend on the strength of crankshaft[5]. In Internal Combustion engines, the transient load of maximum cylinder gas pressure is transmitted to crankshaft through connecting rod, however crankshaft converts reciprocating motion of piston along with connecting rod in the rotating system of components [7].Because of torsion the dynamic load and rotating system swink continues resulting in repeated bending and shear stress on crankshaft, which are common stresses acting on crankshaft and it is most responsible phenomenon in crankshaft development.

In crankshaft crankpin behaves like a built in beam with a distributed load along its overall length that varies with crank position. Each web is like a cantilever beam subjected to bending and twisting. 1. Bending moment which causes tensile and compressive stresses 2. Twisting moment causes shear stress.

Twisting causes shear stress and due to shrinkage of web into journal compressive stresses are set up in journal and tensile hoop stresses in the web. There are many sources of failures in the engine; one of the most common crankshaft failures is fatigue at the fillet areas due to bending load causes by the combustion. At the root of the fillet areas, stress concentrations exist and these high stress range locations are the points where cyclic loads could cause fatigue crank initiation leading to fracture.

MATLAB is an interactive system whose basic data element is an array that does not require dimensioning. This allows you to solve many technical computing problems, especially those with matrix and vector formulations, in a fraction of the time it would take to write a program in a scalar no interactive language such as C or FORTRAN [13].

MICRO-ALLOYING

Micro-alloyed steels are defined as steels containing small amounts of niobium, vanadium, or titanium, micro-alloying can lead to major increases in strength and toughness [4].

Micro alloyed steels have been gaining popularity as an economical alternative to the medium carbon steel. These steels have low or medium carbon content and small additions of alloying elements such as Mn, Nb, Mo, V and Ti. Titanium has a high propensity for forming oxides and sulphides, the lowest solubility and is generally ineffective as a precipitation strengthen in medium to high carbon steels [4].

The rising cost and volatility of metals has led to the development of new alloy systems for PM steels. The high yield strength is achieved by the combined effects of fine grain size developed during controlled hot rolling and precipitation strengthening that is due to the presence of vanadium, niobium, and titanium.

In niobium micro-alloyed steels carbon is the preferred element for precipitation and at higher carbon levels the solubility of Nb(C,N) is reduced and higher temperatures are needed to get niobium in solution. In addition, the carbonitrides of niobium are more stable at higher temperatures than those of vanadium [4]. Niobium has a negative effect on the hardness of martensite because of its ability to effectively remove carbon from solution and is generally not used in low alloy martensitic steels.

Table I Mechanical Properties

Material	Modulus of elasticity (GPa)	Poisson's ratio	Yield strength (MPa)	Ultimate tensile strength (MPa)
38MnVS6	210	0.3	550	940

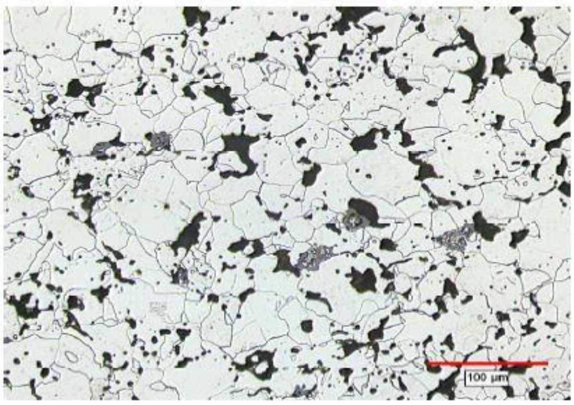

Fig 1 Sintered with Niobium

When vanadium is added, it combines preferentially with nitrogen to form nitrogen-rich vanadium carbonitride [V(C,N)] precipitates. V(C,N) is more soluble in high carbon steels and is less sensitive to carbon level compared with niobium [4].

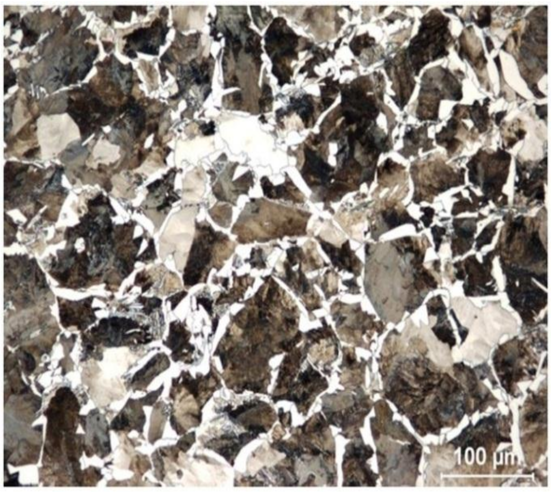

Fig 2 Microstructure of 38MnVS6 Micro-alloy steel

Table II Roles of Micro-Alloying Element [5]

Micro -alloy	Precipitation Strengthening After Hot rolling	Precipitation Strengthening after Normalizing	Influences Recrystalli-zation during Hot Rolling	Refines Grain Size on Normalising	Refines Grain Size during High Temp. Austenitizing	Influences Transformation Characteristics After Hot Rolling
V	VN,VC	VC		VN		
Nb	Nb,CN		Nb,NbCN	NbCN		Nb
Ti	TiC			TiC	Tin	

FATIGUE

Fatigue failures occur due to the application of fluctuating stresses that are much lower than the stress required to cause failure during a single application of stress. It has been estimated that fatigue contributes to approximately 90% of all mechanical service failures [9]. Nowadays in strongly developed probabilistic approaches to fatigue assessments, deterministic approach is however still in use and therefore important, especially in the design phase of the machine parts, structural components and joints. Such components are frequently subjected to the high cycle fatigue (HCF) stress cycling Process and therefore the stress approach to fatigue design is suitable.

Goodman line is based on a huge number of testing and it is more or less unquestionable and generally accepted in design community. Today, structural fatigue has assumed an even greater importance as a result of the ever-increasing use of high-strength materials and the desire for higher performance from these materials.

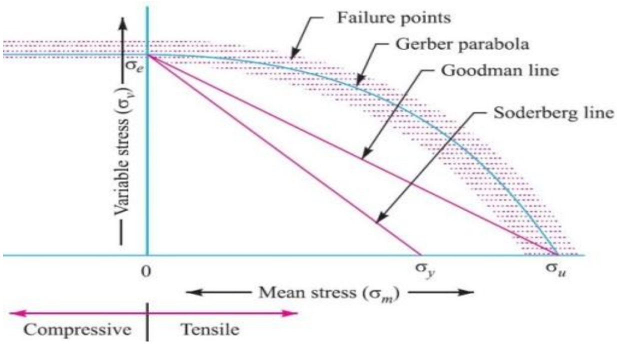

Fig 3Combined mean and variable stress [12]

There are several ways in which problems involving this combination of stresses may be solved, but the following are important from the subject point of view:

Goodman Relation:

$$\sigma_a = \sigma_{fat}(1 - \frac{\sigma_m}{\sigma_{ts}})$$

Gerber Relation:

$$\sigma_a = \sigma_{fat}[1 - (\frac{\sigma_m}{\sigma_{ts}})^2]$$

Soderberg Relation:

$$\sigma_a = \sigma_{fat}(1 - \frac{\sigma_m}{\sigma_{ys}})$$

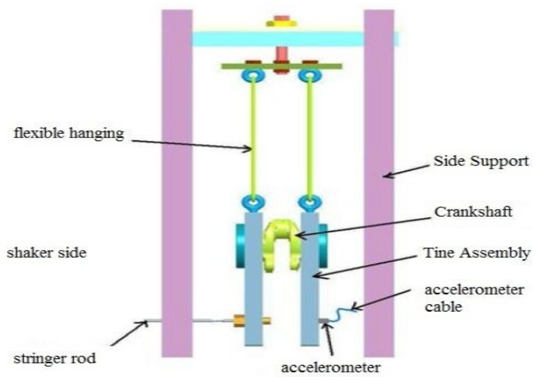

Fig 4 A testing arrangement of the resonance tester for bending loading

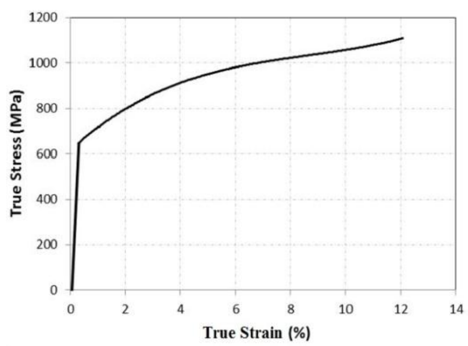

Fig 5 True stress-strain curve of 38MnVS6 steel

A. High-Cycle Fatigue

High-cycle fatigue involves a large number of cycles (N >10e5 cycles) and an elastically applied stress. High-cycle fatigue tests are usually carried out for 10e7 cycles and sometimes 5x10e8 cycles for nonferrous metals. High-cycle fatigue data are usually presented as a plot of stress, S, versus the number of cycles to failure, N. The S-N relationship is usually determined for a specified value of the mean stress, sm, or one of the two ratios, R or A [9]. The fatigue life is the number of cycles to failure at a specified stress level, while the fatigue strength (also referred to as the endurance limit) is the stress below which failure does not occur.

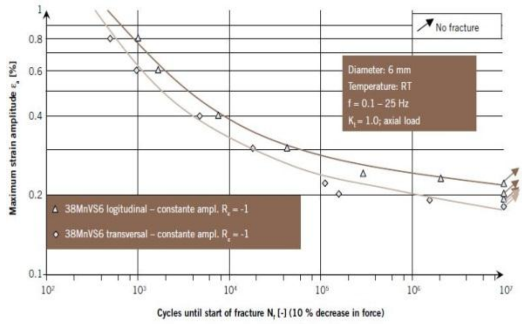

Fig 6 The dependence of high-cycle fatigue strength on the orientation of the fibres (Wöhlerdiagram) [11]

As the applied stress level is decreased, the number of cycles to failure increases. Normally, the fatigue strength increases as the static tensile strength increases.

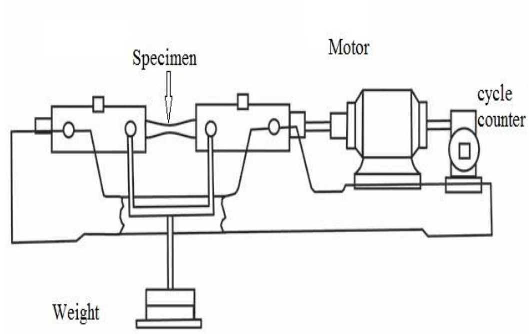

Fig 7 Schematic of R.R. Moore reversed-bending fatigue machine [9]

Fatigue cracking can occur quite early in the service life of the component by the formation of a small crack, generally at some point on the external surface. The crack then propagates slowly through the material in a direction roughly perpendicular to the main tensile axis.

Goodman developed a linear model, while Gerber used a parabolic model. Test data for ductile metals usually fall closer to the Gerber parabolic curve; however, because of the scatter in fatigue data and the fact that notched data fall closer to the Goodman line, the more conservative Goodman relationship is often used in practice. If the component design is based on yield rather than ultimate strength, as most are, then the even more conservative soderberg relationship can be used.

B. Low-Cycle Fatigue

During cyclic loading within the elastic regime, stress and strain are directly related through the elastic modulus. In cyclic strain-controlled fatigue, the strain amplitude is held constant during cycling. Since plastic deformation is not completely reversible, the stress-strain response during cycling can change, largely depending on the initial condition of the metal. When the metal is highly work hardened and the dislocation density is high, cyclic strain allows the rearrangement of dislocations into more stable networks, thereby reducing the stress at which plastic deformation occurs. Conversely, when the initial dislocation density is low, the cyclic strain increases the dislocation density, increasing the amount of elastic strain and stress on the material.

DESIGN OF CRANKSHAFT

To start crankshaft design first all the basic engine details are required. Basic engine details like bore, stroke, engine configuration, compression ratio, bore centre distance and connecting rod length will decide the basic shape of the crankshaft and one can be able to decide the crankpin and main journal locations.For the theoretical calculation of crankshaft we consider the configuration 4 cylinder diesel engine to calculate the theoretical static result:

Table III Specifications of 4 Cylinder Diesel Engine

SR. NO.	Type	4 Cylinder Diesel engine
1	Capacity of engine	2596 cc
2	Number of cylinder	4
3	Bore x Stroke	90.9 x 100 (mm)
4	Compression Ratio	18:1
5	Maximum power	80 HP (60kW) @ 3600 rpm
6	Maximum torque	230 Nm @ 1800-2200 rpm
7	Maximum gas pressure	110 Bar (11 N/mm^2)

The design of crankshafts is based on an evaluation of safety against fatigue in the highly stressed areas. To increase the fatigue life of the shaft, the fillet radius between journals and webs should be as large as possible but not less than 5% of the journal diameter. The calculation is also based on the assumption that the areas exposed to highest stresses are:

- fillet transitions between the crankpin and web as well as between the journal and web
- outlets of crankpin oil bores
- outlets of journal oil bores

The methods of crankshaft fatigue are based on:

- A combination of bending and torsional stresses where bending fatigue strength and torsional fatigue strength are separately assessed, i.e., also valid for anisotropic material behaviour
- Measured stress concentration factors, or analysed with FEM or by using empirical formulae with one standard deviation above 50% reliability.
- Fatigue strength assessment based on full scale testing or small scale, combined with a calculation procedure considering the influence of material strength, forging method, mean stress, and when applicable also fillet surface hardening, shot penning or cold rolling including the influence of the depth of this surface treatment.

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 an angle of 25° to 30º from the dead centre for a constant volume combustion engines (*i.e.,* petrol engines) and 30º to 40º for constant pressure combustion engines (*i.e.,* diesel engines).

A. Crankshaft at dead centre

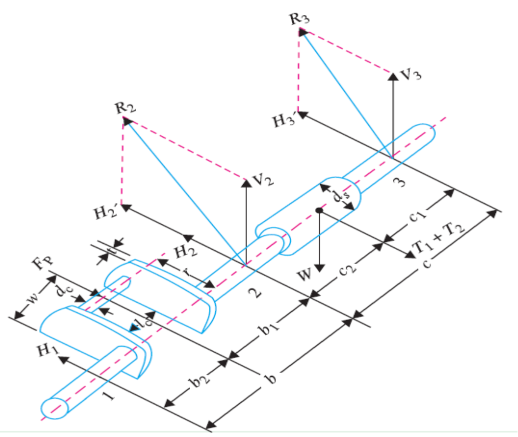

Fig 9 Centre crankshaft at dead centre [12]

B. Crank at an Angle of Max. Twisting Moment

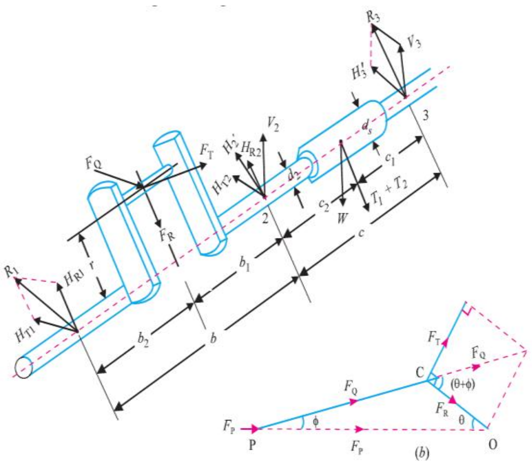

Fig 10 (*a*) Crank at an angle of maximum twisting moment (*b*) Forces acting on the crank [12]

Mathematical Model for Crankshaft:

➢ Piston gas load : The P' is the intensity of pressure (8.5 N/mm^2) on the piston at this instant, then the piston gas load at this position of crank,

$$F_p = \frac{\pi}{4}(D^2) * P'$$

$$= \frac{\pi}{4}(90.9^2) \times 8.5$$

= 55.16 KN

$$l/r = \frac{\sin\theta}{\sin\phi} \rightarrow \sin\phi = \frac{\sin\theta}{4}$$

$$\sin\phi = \frac{\sin 35}{4} = 0.1433$$

$$\phi = 8.244°$$

Thrust on connecting rod,

$$F_Q = \frac{F_p}{\cos\phi} = \frac{55.15}{\cos 8.244} = 54.5\ KN$$

The thrust in the connecting rod (*F*Q) may be divided into two components, one perpendicular to the crank and the other along the crank. The component of *F*Q perpendicular to the crank is the tangential force (*F*T) and the component of *F*Q along the crank is the radial force (*F*R) which produces thrust on the crankshaft bearings

Here, $F_T = F_Q * \sin(\theta + \phi)$

$= 55.4 \times \sin(35 + 8.244)$

= 37.95 KN

Due to the tangential force (F_T), there will be two reactions at brg. 1 and 2,

$$H_{T1} = \frac{F_T * b_1}{b} = \frac{37.95 \times 90.9}{181.8} = 18.97\ KN$$

$$H_{T2} = \frac{F_T * b_2}{b} = \frac{37.95 \times 90.9}{181.8} = 18.97\ KN$$

Here, $F_R = F_Q \times \cos(\theta + \phi)$

$= 55.4 \times \cos(35 + 8.244)$

= 40.35 KN

Due to the radial force (F_R), there will be two reactions at the brg. 1 and 2,

$$H_{R1} = \frac{F_R * b_1}{b} = \frac{40.35 \times 90.9}{181.8} = 20.17\ KN$$

$$H_{R2} = \frac{F_R * b_2}{b} = \frac{40.35 \times 90.9}{181.8} = 20.17\ KN$$

➢ Design of crankpin : (d_c) :

dc= Diameter of the crankpin in mm.

Bending moment,

$$B.M_{centre\ of\ crankpin} = H_{R1} \times b_2 = 20.17 \times 90.9$$

$= 1.83 \times 10^6\ Nmm$

Twisting moment,

$$T.M._{crankpin} = H_{T1} \times r = 18.97 \times 50$$

$= 9.50 \times 10^5\ Nmm$

Ranjeet Ghule[1]*, M. V. Kavade[2], G. S. Uppar[3]

Equivalent twisting moment,

$$\mathrm{T.M.}_{equiv.}, T_e = \sqrt{(M_c{}^2) + (T_c{}^2)}$$

$= 2.06 \times 10^6\ Nmm$

The equivalent twisting moment,

$$T_e = \frac{\pi}{16}(d_c)^3 * \tau$$

$$2.06 \times 10^6 = \frac{\pi}{16}(d_c)^3 * 75$$

$d_c = 51.8 = 52$ mm

Since this value of crankpin diameter (*i.e. dc* = 50 mm) is less than the newly calculated value of *dc* = 52 mm, therefore, we shall take *dc* = 52 mm.

➢ Design of Crank against Fatigue Loading

According to distortion energy theory,

$$M_{ev} = \sqrt{(\mathrm{k}_b * \mathrm{M}_c)^2 + (3/4)(\mathrm{k}_t * \mathrm{T}_c)^2}$$

$$M_{ev} = 3.86 \times 10^6 \text{ Nmm}$$

$$M_{ev} = \frac{\pi}{16}(d_c)^3 \times \sigma_{von}$$

$$\sigma_{von} = 279.79\ \ N/mm^2$$

Where,

Kb=Combine Shock and fatigue factor for bending= 2

Kt=Combine Shock and fatigue factor for torsion = 1.5

Now, Shear Stress

$$T_{ev} = \sqrt{(\mathrm{k}_b * \mathrm{M}_c)^2 + (\mathrm{k}_t * \mathrm{T}_c)^2}$$

$$T_{ev} = 3.9264 \times 10^6 \text{ Nmm}$$

$$T_{ev} = \frac{\pi}{16}(d_c)^3 * \tau$$

$$\tau = 142.25\ N/mm^2$$

RESULTS

Diameter of the crankpin = 52 mm

Length of the crankpin = 35 mm

Diameter of the shaft = 70 mm

Web thickness (both LH and RH) = 20 mm

Web width (both LH and RH)= 70 mm

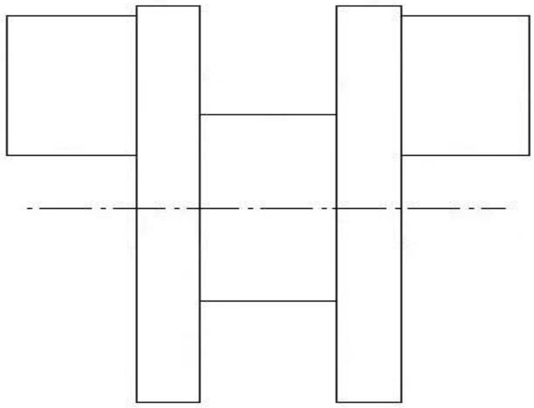

Fig 11 2-D view of crankshaft

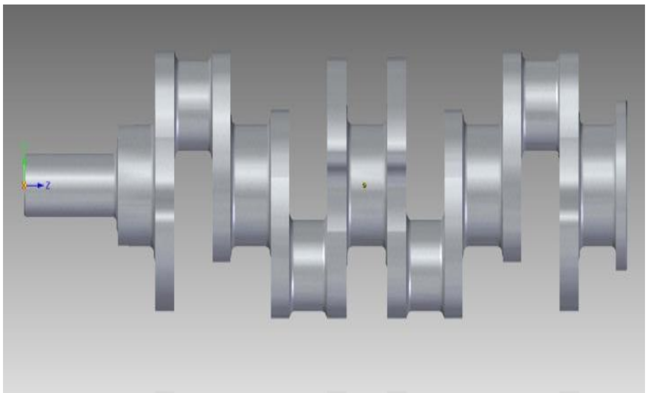

Fig 12 Model of 4 cylinder crankshaft (Solid Edge ST7)

MATLAB

The name MATLAB stands for matrix laboratory. MATLAB is a high-performance language for technical computing. It integrates computation, visualization, and programming in an easy-to-use environment where problems and solutions are expressed in familiar mathematical notation.

Typical uses include:

i. Math and computation
ii. Algorithm development
iii. Modeling, simulation, and prototyping
iv. Data analysis, exploration, and visualization
v. Scientific and engineering graphics

vi. Application development, including graphical User Interface building

The MATLAB system consists of five main parts:

i. The MATLAB language.

ii. The MATLAB working environment.

iii. Handle Graphics.

iv. The MATLAB mathematical function library

v. The MATLAB Application Program Interface (API).

MATLAB features a family of application-specific solutions called toolboxes. Very important to most users of MATLAB, toolboxes allow you to *learn* and *apply* specialized technology. Toolboxes are comprehensive collections of MATLAB functions (M-files) that extend the MATLAB environment to solve particular classes of problems. Areas in which toolboxes are available include signal processing, control systems, neural networks, fuzzy logic, wavelets, simulation, and many others [13].

RESULT AND DISCUSSION

The MATLAB gives the result as same as that of numerically calculated values of the design. There is % error only because of the approximation of the values which is very less. The time required for the calculations of results is very less so the design time is reduced.

CONCLUSION

In this work, a crankshaft is designed using theoretical calculations for the given engine specifications. The designed part is modeled in 3D modeling software Solid Edge ST7. The answers generated by MATLAB code are verified with the analytical answers and are proved to be correct. The design of any component can be done using MATLAB and the results can be obtained with relative ease.

FUTURE WORK

Analysis of crankshaft will be done in CAE software. Comparison of the MATLAB results will be done with the analysis results obtained from CAE software.

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Corresponding Author

Ranjeet Ghule*

Mechanical Department, Rajarambapu Institute of Technology, Sakhrale, Sangli, India

E-Mail – ranjeetghule92@gmail.com