Chapter 16
The Stress Analysis of the Jack Rod Crankshaft on a Single-Cylinder Engine: A Study Using the Finite Element Method

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Abstract The crankshaft is an important component in an engine that converts the reciprocating displacement of the piston into a rotary motion. The maximum possibility of crankshaft failure on the crankpin occurs because the load of pistons and connecting rods are indirectly induced onto the crankshaft. Therefore, crankshaft failure will occur due to the decrease in fatigue strength. The main purpose of this research is to investigate the stress performance on single-cylinder engine crankshaft by setting up differing heights of the crankpin. By using finite element analysis, the analytical data on the stress and the effect of the crankshaft in the occurrence of geometry changes has shown that the high stress value is the main factor that contributes towards the reduction of the crankshaft lifecycle in the engine. A comprehensive study on the material types is recommended for future improvement on the study.

Keywords Crankshaft · Single-cylinder engine · Jack rod · Stress analysis

16.1 Introduction

The main function of the crankshaft is to convert the mechanical energy on the conversion of the piston driven by pneumatic energy to move back and forth and it is due to pressure changes that occur as a result of the combustion reaction. The crankshaft is also an engine component that involves an internal combustion process for high

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volume production and it is usually used in vehicle engines. In engine operation, the reciprocating motion of the piston is a linear motion and is converted to rotary motion by passing through the crankshaft (Mathapati and Dhamejani 2015). The maximum gas pressure on the piston will release the maximum force on the crankpin in the plane of the crankshaft which will cause only the bending of the shaft to occur (Joshi and Patel 2017). In addition, the crankpin and crank web of the crankshaft will experience a bending moment. The maximum possibility of crankshaft failure on the crankpin occurs because the load of pistons and connecting rods are indirectly induced on the crankshaft. Therefore, crankshaft failure will occur due to a decrease in fatigue strength.

The pressure on the shaft is subjected to a force variable; however, in general, it should be divided into two positions. First, at the maximum bending position, it is likely to allow failure in the central region of the crank or at both ends ((Amit and Jaydeepsinh 2014; Durga Prasad et al. 2018)). The pressure in this position is a maximum pressure but it is a small fraction of the maximum. Furthermore, there is a possibility of a curved failure on the wrench that makes the interfacing bar to be affected by the shear in the maximum bending position (Aatish et al. 2020). Crankshafts are generally subject to torsional stresses and bending stresses arising from the weight of the crankshaft itself or the possibility of misalignment between journal bearings or the weight of components on the crankshaft (Kumar et al. 2014). The reliability and service life of the engine are influenced by the crankshaft having stiffness and intensity or not. This is because the design on the crankshaft and its load is quite complex which uses classical mechanical methods to analyse structures that often have limited characteristics (Yingkui and Zhibo 2011; Ding and Li 2011).

The finite element analysis (FEA) by using the ANSYS software is to analyse the stresses, models and harmonics on the crankshaft. Hence, the results of the stress analysis on the crankshaft found the values of the maximum stress point, maximum deformation and critical points areas on the crankshaft (Mathapati and Dhamejani 2015). The force that has been applied through the crankshaft will make it very stressed on the crank pin area with the web crank and the junction of the journal with the web crank. Therefore, it will provide a response on the fillet radius area to reduce stress and this stress has a high level of achievement on material strength and fatigue life (Karthick et al. 2021; Thriveni and Chandraiah 2013; Metkar et al. 2013). The location at the failure critical on the crankshaft is in the critical fillet area that connects the crankpin with the web crank and the web crank with the shaft (Mathapati and Dhamejani 2015; Sorte and Sheikh 2013; Pandiyan et al. 2018). The reduction in crankshaft weight will provide cost reduction on the crankshaft as well as improvements upon engine performance in the internal combustion engine (Karthick et al. 2021). Maximum deformation often occurs in the mid-crankpin region and maximum stresses appear in the area between the crank cheeks and the journal but the design against the crankshaft is safe or strong because the von Mises stress value is less than the yield stress value of the material (Joshi and Patel 2017; Amit and Jaydeepsinh 2014). The edge of the main journal is an area of high stresses (Solanki and Dodiya 2014; Singh et al. 2014).

The performance single-cylinder engine crankshaft has various heights on the crankshaft in other terms it is called as "Jack rod". In this paper, the values emphasized in crankpin heights start from 1.15, 2, 3 and to 4 mm. The main purpose of this paper is to observe the change in resistance resulting from these differences made in crankpin height and whether it provides variable changes. In addition, observations in the changes that occur at each height difference on the crankpin involve a variable of force applied to the crankshaft whether it allows the occurrence of bending moments and in the area most affected. Thus, the crankshaft performance for the 135 cc 4-stroke motorcycle model is made the reference in this paper. Hence, the analysis is performed after performing the modelling process on the crankshaft. In addition, static analysis was chosen in order to determine von Mises stress, strain and deformation. A force variable will be applied to each crankpin height difference to observe how the stress builds up and the resistance that will result from each crankpin height difference on the crankshaft. Thus, the results released by ANSYS software will be the main output of this research.

16.2   Methodology

The research is focused to determine the stress on the crankpin height difference on the performance of a single-cylinder crankshaft. Therefore, CATIA V5 software and ANSYS workbench are used in this research. Performance single-cylinder crankshaft on the Yamaha 135 LC model is used as references. Measurements on the performance of the crankshaft must be made accurately and without errors to ensure that every result that comes out is a reliable source. Furthermore, the single-cylinder performance crankshaft was modelled with four different crankpin heights, namely 1.15, 2, 3 and 4 mm. Therefore, there are four crankshafts to be modelled in this software. Once the design is completed, the performance single cylinder of this crankshaft will be simulated using ANSYS. All data will be collected and analysed.

16.2.1   Analytical Calculation

The design calculation of a single-cylinder petrol engine crankshaft with specific two-wheeler Yamaha 135LC regular model is as per description. The dimension of the engine is given in Table 16.1.

- Moment on pin:

$$M = \frac{P_\text{p}}{2} \times \frac{l_\text{c}}{2} = \frac{20000}{2} \times \frac{45}{2} = 225 \times 10^3 \, \text{Nmm} \qquad (16.1)$$

Table 16.1 Yamaha 135LC engine specification

Engine type	4-stroke, 1 cylinder. Liquid-cooled
Valve system	SOHC, 4-valve
Cylinder bore	54 mm
Stroke	58.7 mm
Displacement	134.4 cm^3
Compressor ratio	10.9:1
Maximum power	8.45 kW/8500 r/min
Maximum torque	11.65 Nm/5500 r/min

- Section module of crankpin:

$$Z = \frac{\pi}{32} \times (d_c)^3 = \frac{\pi}{32} \times (27.8^3) = 2109.82 \, \text{mm}^3. \tag{16.2}$$

- Torque obtained at a maximum power of Yamaha 135LC engine:

$$P = \frac{2\pi N T}{60} \quad \text{where} \quad 8.45 \times 10^3 = \frac{2 \times \pi \times 8500 \times T}{60}. \tag{16.3}$$

$$T = \frac{8.45 \times 10^3 \times 60}{2 \times \pi \times 8500} = \frac{507000}{53407.0751} = 9.4931 \times 10^3 \, \text{Nmm} \tag{16.4}$$

- Von Mises stress:

$$\sigma_{\text{von}} = \frac{M_{\text{eq}}}{Z} \tag{16.5}$$

where

$$M_{\text{eq}} = \text{equivalent bending moment}$$

So, the Equivalent bending moment

$$M_{\text{ev}} = \sqrt{(k_b \times M)^2 + \frac{3}{4}(k_c \times T)^2} \tag{16.6}$$

where

k_b = Combined shock and fatigue for bending = 1
k_t = Combined shock and fatigue for bending = 1
M_{max} = Bending moment

$$= \sqrt{\left(1 \times 225 \times 10^3\right)^2 + \frac{3}{4}(1 \times 9.4931 \times 10^3)^2} = 225.15 \times 10^3 \, \text{Nmm}$$

Now,

$$\sigma_{\text{von}} = 106.72\,\text{MPa}$$

- Stain

$$\varepsilon = \frac{\sigma_{\text{von}}}{\varepsilon} = \frac{106.72}{21000} = 0.0005082 \tag{16.7}$$

Analytical calculation at force 40,000 N

Description	Value
Moment on pin, M	450×10^3 Nmm
Section module of crankpin, Z	2109.82 mm^3
Torque at maximum power, P	9.4931×10^3 Nmm
Von Mises stress, σ_{von}	213.32 MPa
Strain, ε	0.001524

Analytical calculation at force 80,000 N

Description	Value
Moment on pin, M	900×10^3 Nmm
Section module of crankpin, Z	2109.82 mm^3
Torque at maximum power, P	9.4931×10^3 Nmm
Von Mises stress, σ_{von}	426.59 MPa
Strain, ε	0.002031

Analytical calculation at force 100,000 N

Description	Value
Moment on pin, M	1125×10^3 Nmm
Section module of crankpin, Z	2109.82 mm^3
Torque at maximum power, P	9.4931×10^3 Nmm
Von Mises stress, σ_{von}	533.24 MPa
Strain, ε	0.002539

(a) Design 1.15mm height of crankpin

(b) Design 2mm height of crankpin

(c) Design 3mm height of crankpin

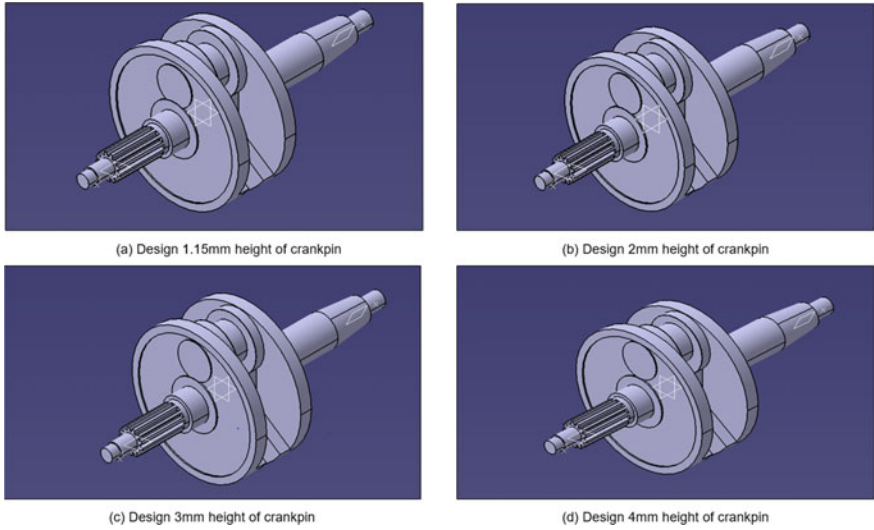(d) Design 4mm height of crankpin

Fig. 16.1 Models of single-cylinder engine crankshaft

16.2.2 Models of Single-Cylinder Engine Crankshaft

Four differences crankpin height of single-cylinder performance crankshaft model have been modelled in the CATIA V5 software and exported into ANSYS workbench software which is saved as STP file format to perform further analysis. The models are shown in Fig. 16.1.

16.2.3 Material Properties of Crankshaft

The material properties of the crankshaft are given in Table 16.2.

Table 16.2 Material properties of forged steel crankshaft	Material type	Forged steel
	Density (kg/m^3) 1.15 mm 2 mm 3 mm 4 mm	: 8395 8420 8444 8469
	Young's modulus (GPa)	210
	Poisson's ratio	0.3
	Yield strength (MPa)	540
	Ultimate tensile strength (MPa)	845

16.2.4 Crankshaft Meshing

The finite element model for the crankshaft geometry will be meshed with tetrahedral elements. Therefore, mesh refinement will be done on the crankpin fillet area and journal fillet in order to obtain a fine mesh on the fillet area. Elements of tetrahedral shapes are used to connect complex geometries that have been imported into the ANSYS workbench software. Mesh optimization is carried out until results from FEA and analytical solutions are obtained and similar to each other. Table 16.3 gives the details of the crankshaft meshing and Fig. 16.2 shows the meshing of the single-cylinder engine crankshaft.

Table 16.3 Details of crankshaft meshing

Height of crankpin (mm)	Type of meshing	Sizing (mm)	Nodes	Elements
1.15	Tetrahedral shapes	1.0	334,847	196,513
2	Tetrahedral shapes	1.0	334,976	196,561
3	Tetrahedral shapes	1.0	338,660	199,109
4	Tetrahedral shapes	1.0	335,912	197,180

(a) Meshing of 1.15mm height of crank pin

(b) Meshing of 2mm height of crank pin

(c) Meshing of 3mm height of crank pin

(c) Meshing of 4mm height of crank pin
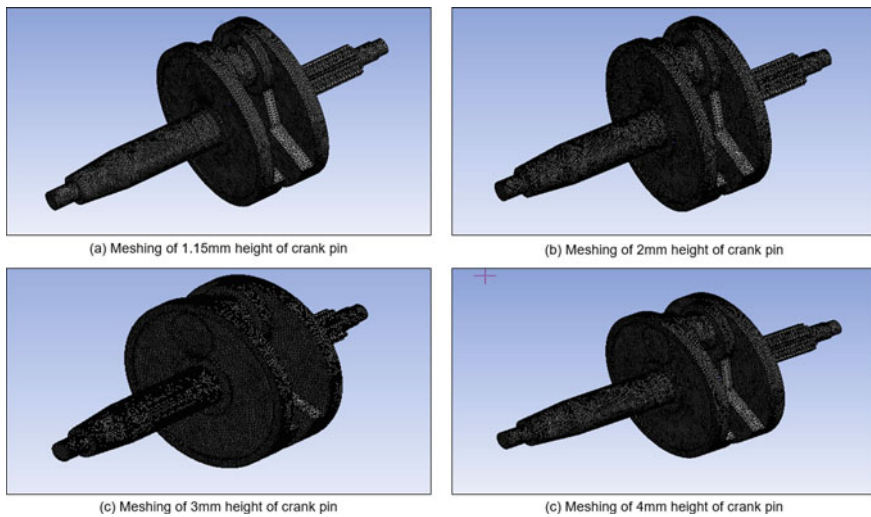
Fig. 16.2 Meshing of single-cylinder engine crankshaft

(a) Fixed support of crankshaft   (b) Applied force at centre of crankpin
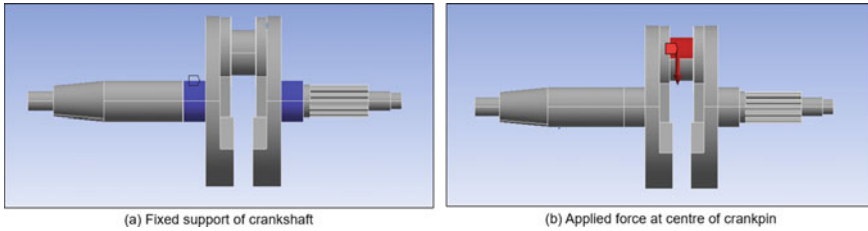
Fig. 16.3 Applied load and boundary conditions

16.2.5 *Apply Load and Boundary Conditions*

The component is charged with a load to observe whether the crankshaft is in the maximum bending moment position or is in the dead centre. In this study, the force will be variable with the gas load on the piston obtained is 20000, 40,000, 60,000, 80,000 and 100,000 N, which is applied at the centre of the crankpin. Therefore, load data was taken from the crankshaft journal. The boundary condition is based on the supporting condition of the crankshaft, which is that crankshaft journals are fixed on both sides of the crank web. Figure 16.3 shows the applied load and boundary conditions.

16.3 Results and Discussion

16.3.1 *Deformation*

Figure 16.4 shows the critical deformation area involved in this single-cylinder crankshaft located in the crank pin area, i.e. on the surface of the crank pin. From the analysis, the highest deformation for 1.15 mm value is at 100,000 N which is placed with a total of 0.034184 mm and while the lowest deformation value is at 20,000 N which has been placed with a total of 0.006837 mm. The highest deformation for 2 mm value is at 100,000 N which is placed with a total of 0.034408 mm and while the lowest deformation value is at 20,000 N which has been placed with a total of 0.006882 mm. The highest deformation for 3 mm value was at 100,000 N which was placed with a total of 0.034743 mm and while the lowest deformation value was at 20,000 N which was placed with a total of 0.006949 mm. The highest deformation for 4 mm value is at 100,000 N which is placed with a total of 0.0350990 mm and while the lowest deformation value is at 20,000 N which has been placed with a total of 0.0070199 mm. Hence, the higher the force applied, the higher the deformation value.
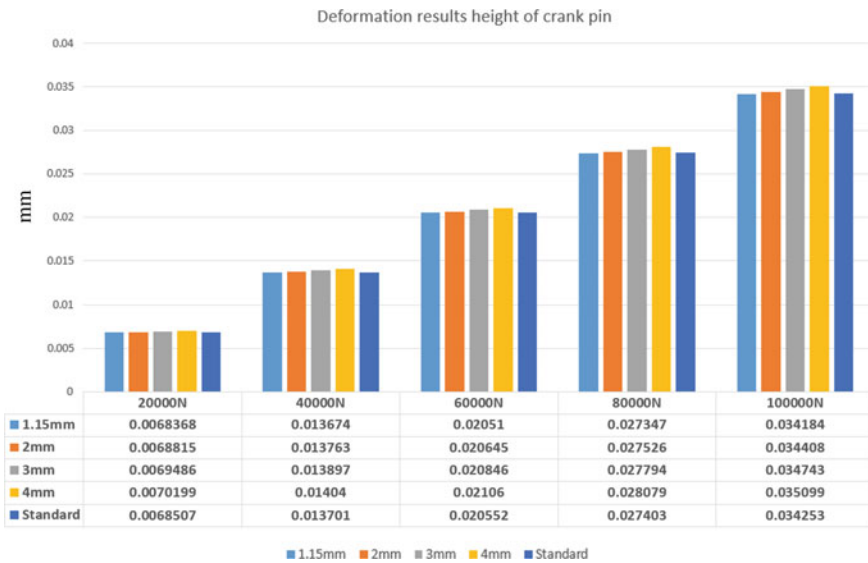
	20000N	40000N	60000N	80000N	100000N
■ 1.15mm	0.0068368	0.013674	0.02051	0.027347	0.034184
■ 2mm	0.0068815	0.013763	0.020645	0.027526	0.034408
▨ 3mm	0.0069486	0.013897	0.020846	0.027794	0.034743
▨ 4mm	0.0070199	0.01404	0.02106	0.028079	0.035099
■ Standard	0.0068507	0.013701	0.020552	0.027403	0.034253

Fig. 16.4 Graphs and data for deformation on height of crank pin

16.3.2   Strain

Based on the strain results obtained, there are two results from the strain, namely the analytical result and the computational result. These results analytical results and software results which produce different data outputs. Figure 16.5 shows that the critical strain area involved in this single-cylinder engine crankshaft is located in the area between the fillet crank web and the shaft. The strain value of the highest strain analytical result is at 100,000 N which is placed with a total of 0.0005082 and while the lowest strain value is at 20,000 N which has been placed with a total of 0.0025390. The highest strain for 1.15 mm value on the result software is at 100,000 N which is placed with a total of 0.0046062 and while the lowest strain value is at 20,000 N which has been placed with a total of 0.0009212. The highest strain value for 2 mm on the result software is at 100,000 N which is placed with a total of 0.0040417 and while the lowest strain value is at 20,000 N which has been placed with a total of 0.0008083. The highest deformation for 3 mm value was at 100,000 N which was placed with a total of 0.034743 mm and while the lowest deformation value was at 20,000 N which was placed with a total of 0.006949 mm. The highest strain for 4 mm value on the result software is at 100,000 N which is placed with a total of 0.0048726 and while the lowest strain value is at 20,000 N which has been placed with a total of 0.0009745. Hence, the higher the force applied, the higher the strain value.

Referring to Fig. 16.5, the higher the value of force applied resulted higher value of strain obtained. The crankshaft is also expanded based on the amount of force that has been applied. The value of stress and strain will increase together if it is
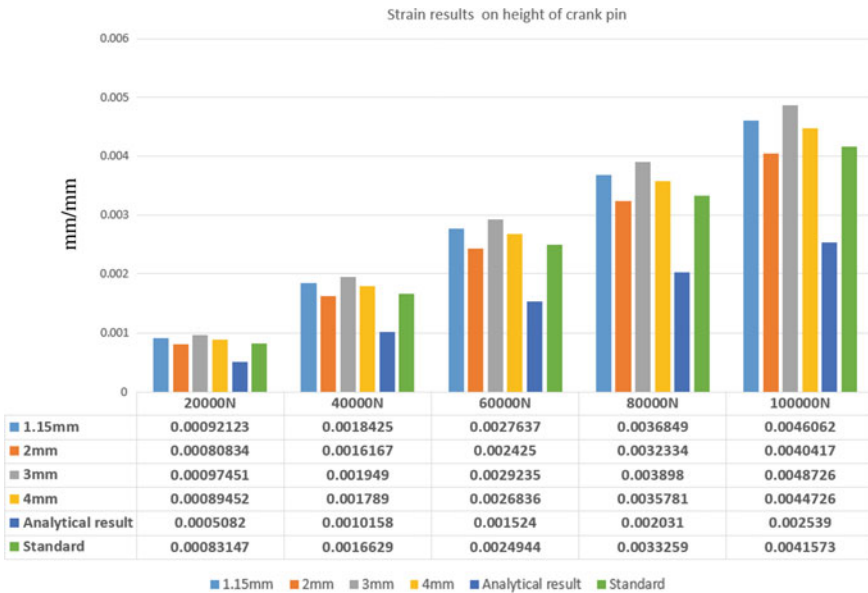
Fig. 16.5 Graphs and data for strain on height of crank pin

in the elastic limit phase. Moreover, the percentage rate difference values at all the crank pin heights have shown that the strain rate percentage values at 3 mm crankpin heights are the highest compared to the crank pin heights at 1.15, 2 and 4 mm. This can be translated because the value of stress experienced at the 3 mm crankpin height is high which has been influenced by the location and height of the crank pin.

16.3.3 Stress

Figure 16.6 shows that the critical stress area involved in this single-cylinder engine crankshaft is located in the area between the fillet crank web and the crank pin. The highest stress value on the analytical result is at 100,000 N which is placed with a total of 533.24 MPa and while the lowest stress value is at 20,000 N which has been placed with a total of 106.72 MPa. The highest stress for 1.15 mm value on the result software is at 100,000 N which is placed with a total of 563.47 MPa and while the lowest stress value is at 20,000 N which has been placed with a total of 112.69 MPa. The highest stress for 2 mm value on the result software is at 100,000 N which is placed with a total of 575.49 MPa and while the lowest stress value is at 20,000 N which has been placed with a total of 115.10 MPa. The highest stress for 3 mm value on the software result is at 100,000 N which is placed with a total of 684.91 MPa and while the lowest stress value is at 20,000 N which has been placed with a total of 136.98 MPa. The highest stress for 4 mm value on the result software is at 100,000 N
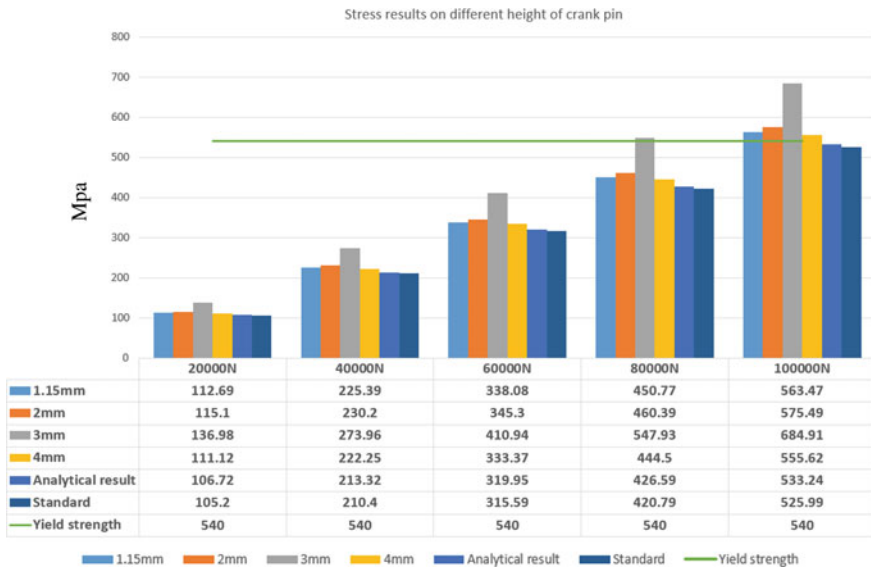
Fig. 16.6   Graphs and data for stress on height of crank pin

The data table for the figure:

	20000N	40000N	60000N	80000N	100000N
1.15mm	112.69	225.39	338.08	450.77	563.47
2mm	115.1	230.2	345.3	460.39	575.49
3mm	136.98	273.96	410.94	547.93	684.91
4mm	111.12	222.25	333.37	444.5	555.62
Analytical result	106.72	213.32	319.95	426.59	533.24
Standard	105.2	210.4	315.59	420.79	525.99
Yield strength	540	540	540	540	540

which is placed with a total of 555.62 MPa and while the lowest stress value is at 20,000 N which has been placed with a total of 111.12 MPa.

This shows the crankshaft at 100,000 N for all crankshaft heights, i.e., 1.15, 2, 3 and 4 mm in phase to fracture or cracking because the stress value obtained has exceeded the yield strength value of 540 MPa. However, for a height of 3 mm, the phase to fracture or crack is experienced earlier at a force value of 80000 N because the resulting stress value exceeds the yield stress value.

The critical stress area for 3 mm height of crank involved in this single-cylinder engine crankshaft is located in the area between the fillet crank web and the crank pin. The higher the value of stress placed, the higher the value of stress obtained. As the stress of the crankshaft is increased, this will decrease fatigue life of the crankshaft. The critical strain area involved in the crankshaft of this single-cylinder engine is located in the area between the fillet crank web and the shaft. The higher the value of force applied, the higher the value of strain obtained. This is because the higher the force that will make the crankshaft to expand according to the amount of force that has been applied. Thus at the critical deformation area for 3 mm height of crankpin involved in this single-cylinder crankshaft located in the crank pin area, i.e., on the surface of the crank pin. Normal deformation may occur due to the maximum load of the crankshaft that needs to be taken for confirmation of work with von Mises stresses.

The stress at 3 mm height of crankpin is higher than the height of other crankpin location. This means fatigue life at 3 mm height of crankpin will decrease. This is because the location of the 3 mm height of the crankpin is not practical because it will put pressure on the web crank to accommodate the crankpin when it experiences any
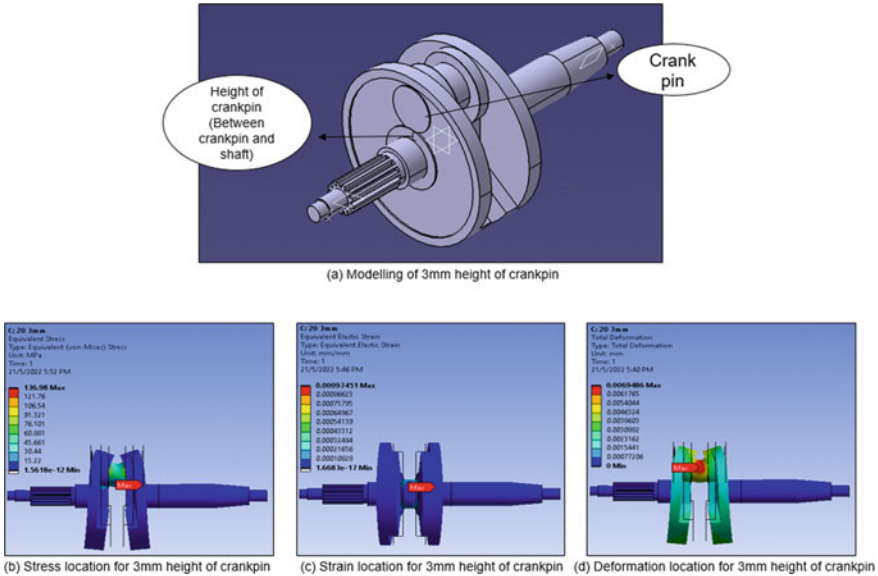
(a) Modelling of 3mm height of crankpin

(b) Stress location for 3mm height of crankpin (c) Strain location for 3mm height of crankpin (d) Deformation location for 3mm height of crankpin

Fig. 16.7   Results and detail for 3 mm height of crankpin

pressure. However, the deformation value for 3 mm height of crankpin is lower than the deformation value at 4 mm. This is because the geometry shape at the crankpin height at 3 mm is lower than 4 mm. If it is getting higher, the crankpin position will be at the end of the web crank which will cause the deformation value to increase. In addition, for the Yamaha 135 LC, crankshaft model at 3 mm height of crankpin is not practiced in the field of modification and it is difficult to find in the market. After performing this simulation analysis, it is proven that this 3 mm height of crankpin has a high stress value compared to the crankpin height at 1.15, 2 and 4 mm. Figure 16.7 shows the results and detail for 3 mm height of crankpin.

16.4   Conclusion

As part of this study, the main purpose is about stress effect on the performance of a single-cylinder four-stroke cycle engine. There are four different crank pin heights on the crankshaft starting from 1.15, 2, 3 and 4 mm from similar engines which were studied in this research. The single-cylinder engine crankshaft models with different crank pin heights, i.e. 1.15, 2, 3 and 4 mm were created by CATIA software. Then, the model that has been designed in CATIA software has been imported into ANSYS software. Thus, finite element analysis using ANSYS software and analytical methods has been performed. The maximum stress results in all differences in crankshaft height in the area between the crank pin and the crank web where the edge of the

crank pin is the main area for maximum stress in the crankshaft. Mesh refinement is performed on the fillet crank pin and fillet shaft, so that a fine mesh is obtained on the fillet area, which is generally a critical location on the crankshaft. The failure in the crankshaft is initiated at the crank pin area and strain is initiated at the area between the crank web and the shaft. The results of comparison at all crankpin heights with different force values show the effect of stress on the crankshaft. Finally, the difference between the analytical result and the software result is dependent on the height of the crank pin. This is because the data output produced by the analytical result is the same even though the geometry on the crankshaft has changed. Therefore, this can be summarized that the software has produced data based on geometry and parameters, while the analytic result is not sensitive with the changes that occur in the crankshaft geometry.

References

Aatish C, Abhishek M, Devendra R et al (2020) Design and Analysis of Crankshaft using Forged Steel & Composite Materials. Int J Res Appl Sci Eng Tech 8(5):1107–1112

Amit S, Jaydeepsinh D (2014) Design and Stress Analysis of Crankshaft for Single Cylinder 4-Stroke Diesel Engine. Int J Res Appl Sci Eng Tech 2(5):320–324

Ding Y, Li X (2011) Crankshaft Strength Analysis of a Diesel Engine using Finite Element Method. 2011 Asia-Pacific Power and Energy Engineering Conference. IEEE, p 1

Durga Prasad K, Jagath Narayana K, Kiranmayee N (2018) Design and Stress Analysis of Crankshaft for Single Cylinder 4 Stroke Diesel Engine. Int J Eng Res Tech 7(11):154–159

Joshi JJ, Patel DM (2017) Design and Failure Analysis of Single Cylinder Petrol Engine Crankshaft using ANSYS Software. Int J Eng Sci Comput 7(4):10549–10555

Karthick L, Mallireddy N, Yogaraja J et al (2021) Modelling and Analysis of an EN8 crankshaft material in comparison with Forged steel crankshaft. Mater Today 47(17):6168–6172

Kumar MS, Ragunathan S, Suresh M (2014) Analysis of Crankpin Failure in a Single Cylinder Engine. Int J Mech Eng Robot Res 3(4):260

Mathapati NC, Dhamejani CL (2015) FEA of A Crankshaft in Crank-pin Web Fillet Region for Improving Fatigue Life. Int J Inno Eng Res Tech 2(6):1–8

Metkar RM, Sunnapwar VK, Hiwase SD et al (2013) Evaluation of FEM based fracture mechanics technique to estimate life of an automotive forged steel crankshaft of a single cylinder diesel engine. Procedia Eng 51:567–572

Pandiyan A, Arunkumar G, Patel A et al (2018) Design and optimization of crankshaft for single cylinder 4-Stroke spark ignition engine using coupled steady-state thermal structural analysis. Int J Mech Eng Techn 9(7):135–145

Singh AK, Praveen KS, Tripathi AK et al (2014) FEA of the crankshafts design by using Ansys workbench for nickel chrome steel and structural steel. Int J Sci Eng Res 5(4):1249–1253

Solanki A, Dodiya J (2014) Design and stress analysis of crankshaft for single cylinder 4-stroke diesel engine. Int J Res Appl Sci Eng Techn 2(5):320–324

Sorte SM, Sheikh SM (2013) Stress Analysis and Design Optimization of Crankpin. Int J Sci Mod Eng 1(4):18–20

Thriveni K, Chandraiah BJ (2013) Modeling and analysis of the crankshaft using ANSYS software. Int J Comp Eng Res 3(5):84–89

Yingkui G, Zhibo Z (2011) Strength Analysis of Diesel Engine Crankshaft based on PRO/E and ANSYS. 3rd International Conference on Measuring Technology and Mechatronics Automation, vol 3. IEEE, p 362